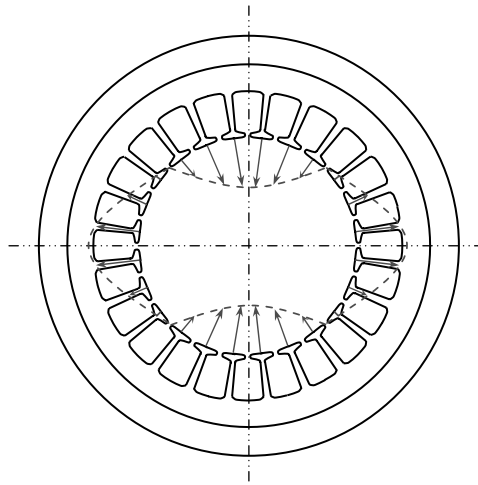


**FIGURE 9.11**

Stator distortion due to traveling forces as a result of rotating magnetic flux.

2. Torsional vibration is the dynamics of the rotor in the angular/rotational direction. Excessive torsional vibration and resonance lead to damaged motor components such as rotor, bearings, couplings, gears, and auxiliary equipment.
3. Axial vibration is the dynamic response of the rotor in the axial direction under external thrust forces and internal electromagnetic forces due to the misalignment of the rotor relative to the stator windings. There are indications that rotor-dynamic vibration and axial vibration influence each other through variations in roller bearing stiffness due to the time-dependent axial load [9.13].

In practice, a multidirectional vibration may occur to act on a motor. The interactions between different directional vibrations make the vibration problems considerably complicated.

9.3 Electromagnetic Vibrations

Magnetic field present in an air gap is the primary source for the electromagnetic vibrations due to the variation in electromagnetic forces over time. The harmonics in the electromagnetic forces induce structural vibration and acoustic noise. The following sections describe the various causes of electromagnetic vibrations.

9.3.1 Unbalanced Forces/Torques Caused by Electric Supply

Stator-related faults account for the second largest number of faults, up to 38% in IMs. Stator winding faults occur when the stator windings are shorted due to the insulation issues related to manufacturing defects, contaminations, overheating and wear, collision of the rotor and stator, and voltage stress imposed by the fast switching of inverters. The unbalanced electromagnetic flux in air gap occurs on the twice line frequency.

9.3.2 Broken Rotor Bar and Cracked End Ring

For squirrel cage IMs, broken rotor bars and cracked end rings are the commonly encountered motor faults, which may be due to manufacturing defects, pulsating loads, frequent direct online starting, and thermal and mechanical stresses from heavy-duty cycle operation [9.14]. The broken rotor bar accounts for approximately 9% of all faults of IM. Due to the bar/end ring breakages, which primarily occur at the joints between bars and end rings, the inducing current in the rotor will be redistributed, leading to overheating of the adjacent bars, torque pulsation, speed fluctuation, changes in magnetic fields, noise, and vibration.

When a broken rotor bar fault occurs in an IM, sideband frequencies f_{sb} (where $f_{sb} > 0$) will appear in stator current around the supplied current fundamental frequency f [9.15]:

$$f_{sb} = (1 \pm 2ks)f, \quad k = 1, 2, 3, \dots \quad (9.61)$$

where s is the slip of motor.

Figure 9.12 illustrates the distribution of sideband frequencies at the sides of the fundamental line frequency. Thus, these sideband harmonics can be used to identify and detect the broken rotor bar fault.

As proposed by Thomason and Rankin [9.16], the number of broken bars n_b for full-load operation (also known as broken bar factor) can be estimated from the following equation:

$$n_b = \frac{2N_{slot}}{10^{N/20} + p} \quad (9.62)$$

where

N_{slot} is the number of rotor slots

p is the pole pair

N is the average decibel difference between upper and lower sidebands

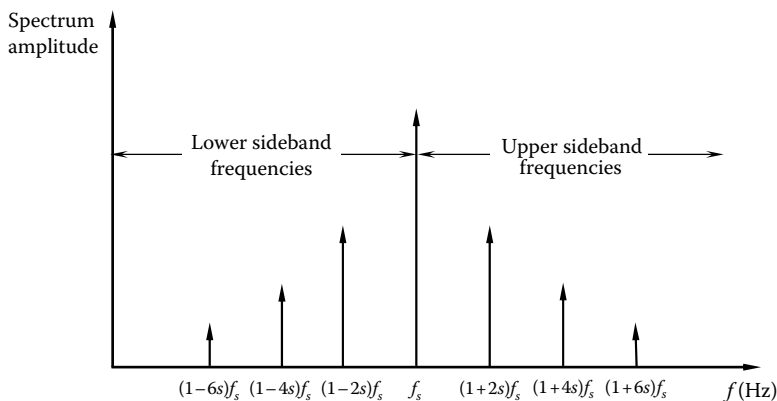


FIGURE 9.12

Sideband frequencies around the fundamental line frequency f_s .

The most important harmonics caused by the broken rotor bars f_b are as follows [9.17]:

$$f_b = (1 \pm 2ks)f, \quad k = 1, 2, 3, \dots \quad (9.63)$$

where

s is the slip of motor

f is the supply frequency

9.3.3 Unbalanced Magnetic Pull due to Asymmetric Air Gap

As electric current flows through the stator windings, a magnetic field is generated in the air gap. The magnetic flux in the air gap thus produces an attractive force between the stator and rotor in the direction of the flux. In a system with a uniform air gap, the rotor is subjected to a uniform attraction force from all directions around it. Therefore, there is no resultant force acting on it due to symmetry. However, when an air gap asymmetry is present, a resultant traction force is produced on both the rotor and stator, occurring at the minimum air gap.

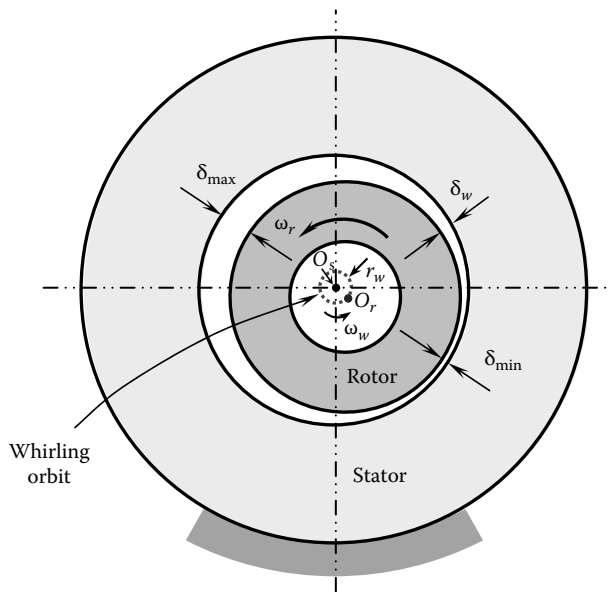
Air gap asymmetries are often encountered in almost all types of electric motors. These asymmetries distort the distribution of the magnetic flux density in the air gap and create unbalanced electromagnetic forces that act on both the rotor and stator. This phenomenon is known as UMP. The presence of UMP always tends to further increase the eccentricity magnitude, and in the worst case, the rotor may touch the stator to cause serious damage to the electric motor.

Asymmetric air gap is called eccentricity in electric motors. A general eccentricity form is cylindrical circular rotor whirling. In this form, the rotor axis always remains parallel to the stator axis and travels around it in a circular orbit with a certain radius and a certain angular velocity (or frequency). This motion is defined as the whirling motion and the corresponding orbit radius and angular velocity (or frequency) are defined as the whirling radius and whirling angular velocity (or whirling frequency), respectively. In dynamic whirling cases, the whirling velocity is equal to the rotor speed. An electric motor operated with the cylindrical circular rotor eccentricity is shown in Figure 9.13. The rotor rotates at a constant angular velocity ω_r . During the rotor rotating, its center O_r also rotates with respect to the stator center O_s at an angular velocity ω_w in a circular orbit.

There are two types of air gap eccentricity associated with electric motors: static eccentricity and dynamic eccentricity, depending on the magnitude of the rotor whirling angular velocity. In the case of a static air gap eccentricity, the positions of the maximum and minimum air gap are fixed in space, that is, the eccentric rotor displacement remains stationary with respect to the stator. This indicates that in this case, the rotor whirling angular velocity is zero (i.e., $\omega_w = 0$). In contrast, in the case of a dynamic air gap eccentricity, the positions of the maximum and minimum air gap rotate with the rotor, implying that the whirling angular velocity ω_w is the same as the mechanical angular velocity of the rotor ω_r . In either static or dynamic air gap eccentricities, the resultant force on the rotor always points to the direction of the minimum air gap, so does the resultant force on the stator but in the opposite direction.

9.3.3.1 Electromagnetic Force at Air Gap

UMP has been discussed for more than a century. Many researchers have performed intensive studies to calculate the UMP for predicting vibration characteristics of rotor

**FIGURE 9.13**

A general form of cylindrical circular rotor whirling.

systems due to UMP. Früchtenicht et al. [9.18] developed an analytical model for predicting the electromagnetic forces between the rotor and stator for a rotor in circular whirling motion. With their model, the stiffness and damping coefficients induced by the electromagnetic field can be determined. It was found that the electromechanical interaction can cause rotordynamic instability. Arkkio et al. [9.19] presented a simple parametric model for calculating the electromagnetic forces at the air gap field when the rotor performs cylindrical circular whirling motion with respect to the stator. They determined the model parameters of a motor by time stepping FEA including the nonlinear saturation of magnetic materials. In this study, the computed and measured forces show good agreement.

The responses due to certain deviations of shape in the rotor and the stator have been investigated numerically and analytically by Lundström and Aidanpää [9.20]. In their study, the perturbation on the rotor is considered to be of oval character, and the perturbations of the stator are considered triangular. In this case, the rotor whirling motion becomes much more complicated.

Regardless of the mechanism of air gap eccentricity, any eccentricity in the air gap results in variations of the distribution of the magnetic flux, magnetic flux density, and reluctance. The magnetic flux density B is defined as the magnetic flux Φ per unit area:

$$B = \frac{\Phi}{A} \quad (9.64)$$

Because the magnetic flux Φ is related to the total MMF F_m and the reluctance of the magnetic flux path R_m ,

$$\Phi = \frac{F_m}{R_m} \quad (9.65)$$

the reluctance can be expressed as

$$R_m = \frac{\delta}{\mu A} \quad (9.66)$$

where μ is the permeability of air. Therefore, it gives that

$$\Phi = \frac{\mu A F_m}{\delta} \quad (9.67)$$

$$B = \frac{\mu F_m}{\delta} \quad (9.68)$$

This indicates that both the magnetic flux Φ and flux density B are inversely proportional to the length of the air gap.

There are two basic methods used to calculate the forces acting between the rotor and stator: the methods that are based on Maxwell's stress and the methods that are based on the principle of the virtual work. Derived from Maxwell's equations, the electromagnetic force at air gap can be expressed as

$$\mathbf{F} = \int_0^{2\pi} \left[\frac{1}{\mu_0} B_r B_\phi \mathbf{e}_\phi + \frac{1}{2\mu_0} (B_r^2 - B_\phi^2) \mathbf{e}_r \right] r d\phi \quad (9.69)$$

where B_r and B_ϕ are the radial and tangential components of the magnetic flux density, respectively. More reliable results are obtained if the line integral in Equation 9.69 is transformed to a surface integral over the cross section of the air gap [9.21]:

$$\mathbf{F} = \frac{1}{\delta(\phi)} \int_{S_{ag}} \left[\frac{1}{\mu_0} B_r B_\phi \mathbf{e}_\phi + \frac{1}{2\mu_0} (B_r^2 - B_\phi^2) \mathbf{e}_r \right] dS \quad (9.70)$$

where

S_{ag} is the cross-sectional area of the air gap

$\delta(\phi)$ is the air gap thickness at the angular position ϕ

$$\delta(\phi) = r_s(\phi) - r_r(\phi) \quad (9.71)$$

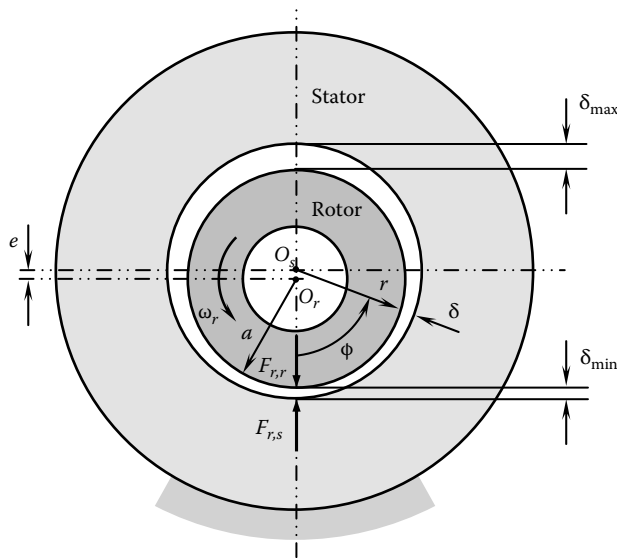
where $r_s(\phi)$ and $r_r(\phi)$ are the outer and inner radii of the air gap. It must be noted that this equation is valid only for small eccentricities [9.22].

It can be seen from Equation 9.70 that the electromagnetic force at the air gap is inversely proportional to the thickness of the air gap. This indicates that even a small change in air gap may result in a large change in electromagnetic force.

To better understand the effects of asymmetric air gaps on UMP, several causes of asymmetric air gap are addressed separately in the following sections.

9.3.3.2 Asymmetric Air Gap due to Nonconcentric Rotor and Stator

There are a variety of causes that can result in air gap asymmetries. As demonstrated in Figure 9.14, an asymmetric air gap resulted from the nonconcentric rotor and stator,

**FIGURE 9.14**

Asymmetrical air gap due to nonconcentric stator and rotor.

which may be due to the accumulation of tolerances during manufacturing, wear of bearings and bearing bores, poor maintenance, and many other factors. This is a special case of the general eccentricity form in Figure 9.13, with $\omega_w = 0$ and $r_w = e$. In this case, the rotor and stator axes are parallel but not coincident. The air gap profile maintains a steady state, that is, the positions of the maximum and minimum air gap are independent of the rotor rotation.

The eccentricity between the rotor and stator is defined as

$$e = \frac{1}{2}(\delta_{\max} - \delta_{\min}) \quad (9.72)$$

where $\delta_{\max}$ and $\delta_{\min}$ are the maximum and minimum air gaps, respectively. As the mean air gap is given as

$$\delta_m = \frac{1}{2}(\delta_{\max} + \delta_{\min}) \quad (9.73)$$

it follows that

$$e = \delta_{\max} - \delta_m = \delta_m - \delta_{\min} \quad (9.74)$$

In a polar coordinate system, the general equation of a circle with the center at $(e, 0)$ and the radius of c can be expressed as

$$r_r^2 - (2e \cos \phi) r_r + e^2 = c^2 \quad (9.75)$$

Solving this equation, it yields

$$r_r(\phi) = e \cos \phi + \sqrt{e^2 (\cos^2 \phi - 1) + c^2} \quad (9.76)$$

It is easy to confirm that at $\phi=0$, $r_r=c+e$ and at $\phi=\pi$, $r_r=c-e$.

Thus, the air gap thickness for a nonconcentric rotor–stator system can be obtained by subtracting the rotor radius from the stator radius in the polar coordinate:

$$\delta_{nc}(\phi) = r_s - r_r(\phi) \quad (9.77)$$

where the stator radius is a constant, expressed as

$$r_s = \delta_m + c \quad (9.78)$$

Therefore, the air gap for nonconcentric rotor and stator is a function of the angular spatial position and can be expressed as

$$\delta_{nc}(\phi) = \delta_m + c - e \cos \phi - \sqrt{e^2 (\cos^2 \phi - 1) + c^2} \quad (9.79)$$

In this case, the resultant forces acting on the rotor $F_{r,r}$ and stator $F_{r,s}$ point each other at the minimum air gap.

9.3.3.3 Asymmetric Air Gap due to Elliptic Stator

An asymmetric air gap can result from a number of manufacturing flaws, such as a non-circular stator core or a noncircular rotor.

The number of poles can significantly impact the vibration patterns. For two-pole motors, electromagnetic forces produced by fundamental flux attempt to deflect the stator into an elliptical shape. Since the stator core is mounted inside the motor housing, the restraining forces to the stator deformation primarily come from the motor housing. Unlike in the previous cases, the air gap profile is independent of the rotor rotation (Figure 9.15). The equation of the elliptic stator bore can be written as

$$\frac{r_{es}^2 \cos^2 \phi}{a^2} + \frac{r_{es}^2 \sin^2 \phi}{b^2} = 1 \quad (9.80)$$

Solving Equation 9.80 for r_{es} , it yields

$$r_{es}(\phi) = \frac{ab}{\sqrt{a^2 \sin^2 \phi + b^2 \cos^2 \phi}} \quad (9.81)$$

The rotor radius is a constant:

$$r_r = a - \delta_{\max} = b - \delta_{\min} \quad (9.82)$$

Similarly, the eccentricity of the elliptic stator is defined as

$$e_{es} = \sqrt{1 - \frac{b^2}{a^2}} \quad (9.83)$$

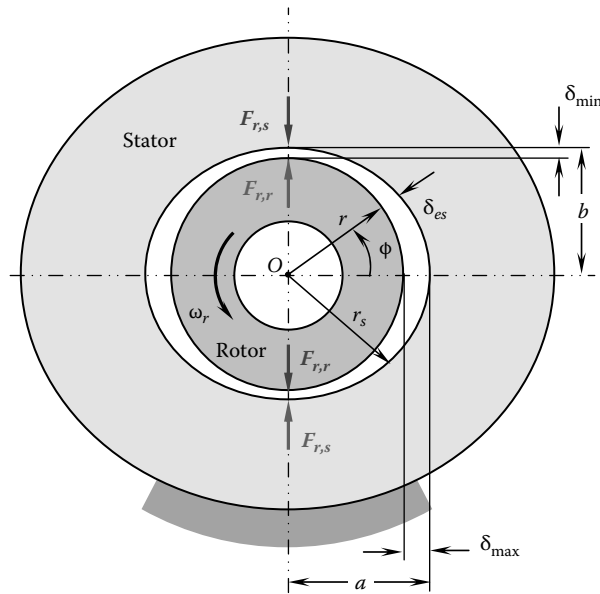


FIGURE 9.15
Asymmetrical air gap due to elliptic stator.

The difference between the maximum and minimum air gaps can be expressed as

$$\Delta\delta_{es} = \delta_{\max} - \delta_{\min} = a \left(1 - \sqrt{1 - e_{es}^2} \right) = b \left(\frac{1}{\sqrt{1 - e_{es}^2}} - 1 \right) \quad (9.84)$$

Hence, the air gap is obtained by subtracting r_r from r_{es} :

$$\delta_{es}(\phi) = r_{es}(\phi) - r_r = \delta_{\max} + a \left(\frac{b}{\sqrt{a^2 \sin^2 \phi + b^2 \cos^2 \phi}} - 1 \right) \quad (9.85)$$

For vibrations caused by unbalanced electromagnetic flux at the air gap, the frequency of vibration f equals to two times the frequency of the power source, called twice line frequency vibration:

$$f = 2f_s \quad (9.86)$$

For instance, the 60 Hz power supply shows a 120 Hz vibration at the outer surface of the stator. Finally, this twice line frequency vibration is transmitted through the motor frame to the environment.

9.3.3.4 Asymmetric Air Gap due to Elliptic Rotor

Figure 9.16 presents the asymmetric air gap due to the elliptic rotor. It is noted that the air gap profile in this case is no longer steady in the stationary coordinate system. It only maintains the steady state in the rotating frame of reference that has the same

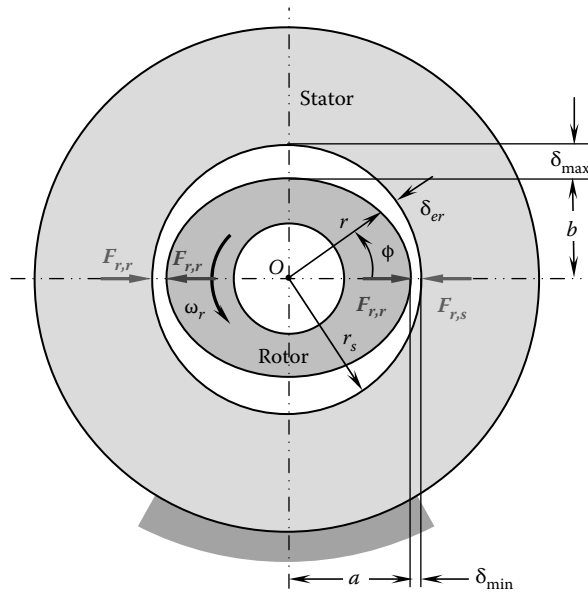


FIGURE 9.16
Asymmetrical air gap due to elliptic rotor.

rotating speed and direction of the rotor. In fact, this is a typical example of dynamic air gap eccentricity.

In the rotating frame of reference (x', y') , a rotating ellipse can be expressed as

$$\frac{x'^2}{a^2} + \frac{y'^2}{b^2} = 1 \quad (9.87)$$

where a and b are the length of the semimajor axis and semiminor axis, respectively. In order to obtain the ellipse equation in the stationary frame of reference, the rotating frame of reference must be converted into a stationary frame via the conversion equation as the following:

$$\begin{bmatrix} x' \\ y' \end{bmatrix} = \begin{bmatrix} \cos(\omega_r t) & \sin(\omega_r t) \\ -\sin(\omega_r t) & \cos(\omega_r t) \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} \quad (9.88)$$

Thus, the rotating ellipse can be expressed in the stationary frame of reference (x, y) :

$$\frac{[x \cos(\omega_r t) + y \sin(\omega_r t)]^2}{a^2} + \frac{[-x \sin(\omega_r t) + y \cos(\omega_r t)]^2}{b^2} = 1 \quad (9.89)$$

The polar coordinate system uses r as the radial coordinate and ϕ as the angular coordinate. Thus, the Cartesian coordinate system (x, y) can be related to the polar coordinate system (r, ϕ) as $x = r \cos \phi$ and $y = r \sin \phi$. Consequently, the equation of the rotating elliptic rotor can be rewritten as

$$\frac{r_{es}^2 \cos^2(\phi - \omega_r t)}{a^2} + \frac{r_{er}^2 \sin^2(\phi - \omega_r t)}{b^2} = 1 \quad (9.90)$$

The eccentricity of the elliptic rotor is defined as

$$e_{er} = \sqrt{1 - \frac{b^2}{a^2}} \quad (9.91)$$

From the equation of the elliptic rotor, it can be derived that the radius of the elliptic rotor is expressed as a function of ϕ :

$$r_{er}(\phi - \omega_r t) = \frac{ab}{\sqrt{a^2 \sin^2(\phi - \omega_r t) + b^2 \cos^2(\phi - \omega_r t)}} \quad (9.92)$$

This equation can be alternatively expressed in terms of eccentricity e [9.23]:

$$r_{er}(\phi - \omega_r t) = a \sqrt{\frac{1 - e_{er}^2}{1 - e_{er}^2 \cos^2(\phi - \omega_r t)}} = \frac{b}{\sqrt{1 - e_{er}^2 \cos^2(\phi - \omega_r t)}} \quad (9.93)$$

The radius of the stator is a constant, expressed as

$$r_s = a + \delta_{\min} = b + \delta_{\max} \quad (9.94)$$

From this equation, the relationship of the difference between the maximum and minimum air gaps and the ellipse parameter a and b can be derived as

$$\Delta\delta_{er} = \delta_{\max} - \delta_{\min} = a - b \quad (9.95)$$

that is,

$$\Delta\delta_{er} = a \left(1 - \sqrt{1 - e_{er}^2} \right) = b \left(\frac{1}{\sqrt{1 - e_{er}^2}} - 1 \right) \quad (9.96)$$

The air gap with the elliptic rotor is thus determined by subtracting the radius of the elliptic rotor from the inner radius of the stator r_s , as shown in the following three equivalent equations:

$$\delta_{er}(\phi - \omega_r t) = r_s - r_{er}(\phi - \omega_r t) = r_s - \frac{ab}{\sqrt{a^2 \sin^2(\phi - \omega_r t) + b^2 \cos^2(\phi - \omega_r t)}} \quad (9.97a)$$

$$\delta_{er}(\phi - \omega_r t) = r_s - r_{er}(\phi - \omega_r t) = \delta_{\min} + a \left(1 - \sqrt{\frac{1 - e^2}{1 - e^2 \cos^2(\phi - \omega_r t)}} \right) \quad (9.97b)$$

$$\delta_{er}(\phi - \omega_r t) = r_s - r_{er}(\phi - \omega_r t) = \delta_{\max} + b \left(1 - \frac{1}{\sqrt{1 - e^2 \cos^2(\phi - \omega_r t)}} \right) \quad (9.97c)$$

For the case of dynamic air gap eccentricity, the resultant forces acting on the rotor $F_{r,r}$ and stator $F_{r,sr}$ occurring at the minimum air gaps, rotate at the same rotating velocity of

the rotor. In the present situation, because of two identical minimum air gaps occurring in the system, a pair of resultant forces $F_{r,r}$ and $F_{r,s}$ acts at each of the minimum air gap, increasing the degree of the rotor eccentricity.

9.3.3.5 Asymmetric Air Gap due to Rotor Misalignment

The rotor misalignments always occur to some extent due to the need to allow some tolerances during the manufacturing process. As a misaligned rotor rotating with respect to its axis that has a small angle with the stator axis (Figure 9.17), the air gap varies along the angular position. However, these cases are more complicated than those discussed previously, usually requiring 3D analyses.

The magnetic force is inversely proportional to the air gap. For a nonuniform air gap, the smaller the air gap, the larger the magnetic force. Regardless of the causes of air gap asymmetry, an uneven air gap always results in unbalance in magnetic forces between the stator and rotor and further results in motor vibration [9.24].

9.3.3.6 Asymmetric Air Gap Resulting from Shaft Deflection

An asymmetric air gap may be induced from the rotor shaft deflection. The causes of shaft deflection may be static or dynamic in origin. The static shaft deflection is defined as the deflection that is measurable when the motor is not in operation, and the dynamic deflection is defined as the deflection that is detectable only during motor operation. A common static shaft deflection is due to the loads acting on the shaft. As a motor operates at high speeds, the centrifugal force acting on the rotor may cause the rotor shaft to bend out as a dynamic deflection (Figure 9.18).

9.3.4 Nonuniform Air Gap due to Stator Slots

The electric current that goes through the stator winding generates a rotating magnetic field and induces magnetic fluxes circulating around the air gap. However, the existence of the stator slots breaks up the uniformity of the air gap. Thus, the air gap will change periodically as the relative position of the stator and rotor changes.

As proposed by Lipo [9.25], due to the stator slot opening, the air gap variation is modeled considering the distribution of flux lines on the slots (Figure 9.19). The air gap rises linearly to the slot center in a slot and then drops up to its nominal value.

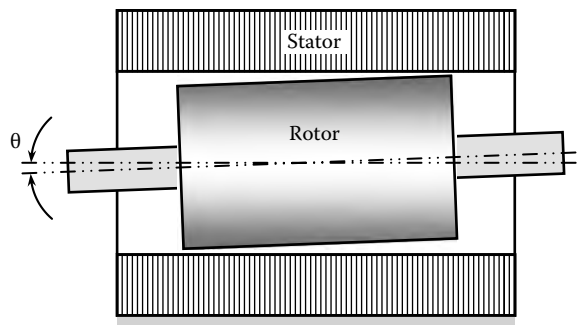


FIGURE 9.17

Asymmetric air gap resulted from the misalignment of rotor.

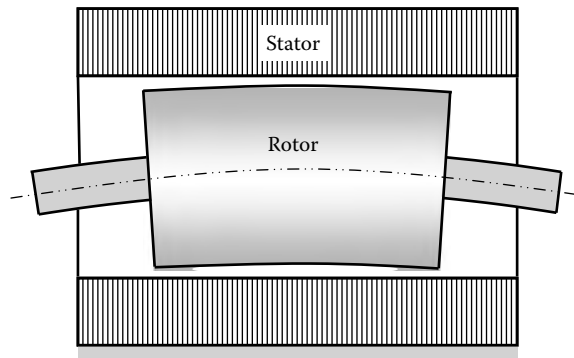


FIGURE 9.18
Asymmetric air gap resulted from the deflected motor shaft.

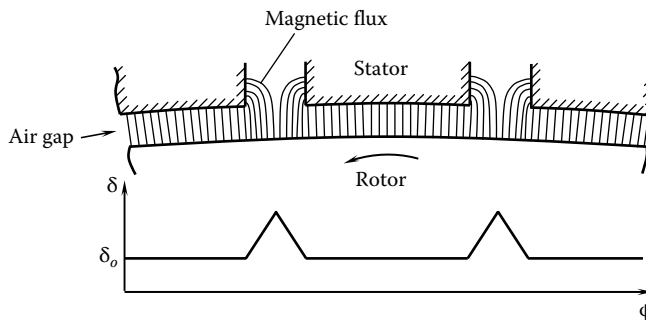


FIGURE 9.19
The impact of stator slots on magnetic flux distribution and air gap length.

The equivalent air gap of an IM is induced by replacing the actual slotted surface with an equivalent unslotted surface, which has the same cross section but with a modified *equivalent* gap. Equating the permeance of the equivalent unslotted surface to the actual permeance, the equivalent gap δ_e and the real gap δ can be related via a so-called Carter factor k_c :

$$\delta_e = \delta k_c \quad (9.98)$$

where k_c is a function of the slot geometries and the real air gap thickness. Lipo [9.25] reported that for most practical machines, δ_e is generally found to be 15%–25% larger than δ .

In practice, several situations discussed previously may occur simultaneously to cause asymmetric air gaps. For such complicated cases, FEMs may be used for predicting the unbalanced magnetic forces.

9.3.5 Mutual Action Forces between Currents of Stator and Rotor

Vibration may come from an irregular magnetic reluctance torque existing between the teeth of the stator and rotor. When the rotor rotates, the electromagnetic force on the stator varies periodically with high frequency because of the teeth structure of salient geometry. This periodic force of high frequency brings vibration and acoustic noise [9.26].

9.3.6 Vibration due to Unbalanced Voltage Operation

According to Muljadi et al. [9.27], in a weak power system network, an unbalanced load at the distribution lines can cause unbalanced voltage conditions. As an induction machine operates under unbalanced voltage conditions, it will result in unbalanced stator current, which creates torque pulsation on the shaft resulting in speed pulsation, mechanical vibration, and, consequently, audible acoustic noise and extra mechanical stress.

9.4 Mechanical Vibrations

Mechanical vibrations can be caused by a variety of factors, including but not limited to unbalanced mass in rotating parts, friction between motor components, misalignment, frame distortion, rotating speed at or near the critical speed, shocks caused by internal or external forces, looseness of parts, bearing deterioration, and various support and coupling effect problems.

9.4.1 Misaligned Shaft and Distorted Coupling

In industry, about 30% of the machines' downtime is due to the poorly aligned machine. Rotor shaft misalignment is the common problem in operation rotating machinery. It is strongly influenced by operation speed and stiffness of the coupling. Flexible couplings tend to provide less amount of vibration levels. Unlike rotor unbalance, misalignment effects on motor vibration are more complex. The vibration induced by the misalignment between a motor shaft and a driven machine shaft varies in different directions.

Misalignment is the most common cause of machine vibration. Experimental studies were performed by Hariharan et al. [9.28] on a rotordynamic test apparatus to obtain the vibration spectrum for shaft misalignment. The vibration spectra show that misalignment can be characterized primarily by second harmonics (2×) of the shaft running speed.

9.4.2 Defective Bearing

Motor bearings could be the main sources of vibration. The primary causes of bearing failures include overloads, nonuniform wear, corrosion, manufacturing errors, bearing fatigue, overheating, current flowing through bearings, high friction sintered bearing material, and improper lubrication and installation.

Most small- and middle-sized motors use rolling element bearings. A rolling element bearing basically consists of an outer and inner race, rolling balls, and a cage for holding the rolling balls. Each damaged component on a bearing can lead to vibration at specific frequencies [9.29,9.30]. Assuming that balls perform only pure rolling motions between the inner and outer races, the characteristic vibrating frequencies that are based upon the bearing dimensions can be determined for each bearing component:

$$\text{Inner race } f_{inner} = \frac{N_b f_r}{2} \left(1 + \frac{d_b}{d} \cos \beta \right) \quad (9.99)$$

$$\text{Outer race} \quad f_{outer} = \frac{N_b f_r}{2} \left(1 - \frac{d_b}{d_p} \cos \beta \right) \quad (9.100)$$

$$\text{Ball} \quad f_b = \frac{d_p f_r}{d_b} \left[1 - \left(\frac{d_b}{d_p} \cos \beta \right)^2 \right] \quad (9.101)$$

$$\text{Cage} \quad f_{cage} = \frac{f_r}{2} \left(1 - \frac{d_b}{d_p} \cos \beta \right) \quad (9.102)$$

where

β is the contact angle

N_b is the number of balls or rollers

f_r is the frequency of rotor rotation ($f_r = n/60$, in rps)

d_b and d_p are the diameter of ball and pitch, respectively

By omitting the clearances between rolling balls and the inner and outer raceways, the pitch diameter can be expressed as

$$d_{pitch} = d_{inner} + d_{ball} = \frac{d_{inner} + d_{outer}}{2} \quad (9.103)$$

Faults of bearing components can be detected using a cepstrum (the word is derived from spectrum by reversing the first four letters of spectrum) that reveals the periodicity of a spectrum.

9.4.3 Self-Excited Vibration

Self-excited vibration system is a system for which the exciting force is a function of the motion parameter, such as displacement, velocity, or acceleration. The motion diverges and the system becomes unstable if energy is fed into the system through self-excitation. As the motion stops, the exciting force vanishes. By contrast, in a forced vibration, the external exciting force always exists regardless of whether the motion stops or continues. Friction-induced vibration (such as motor brakes), flow-induced vibration (such as motor ventilation flows and fluid-conveying pipelines), and aerodynamically induced motion of bridge are typical examples of self-excited vibration.

A self-excited vibration can be also excited from the internal damping of the shaft material. This happens basically for the case of assembled shafts because the friction between two parts in contact is equivalent to the internal damping of the material.

For rotating electric machines, self-excitations often produce vibrations in the rotating shaft in the lateral or flexural directions, rather than in the torsional or longitudinal directions.

9.4.4 Torsional Vibration

Torsional vibration is one of the main concerns in some motor applications. Many rotating machineries such as motors, generators, compressors, and similar machines can be

modeled as two-inertia systems, connected by a shaft that functions as a torsional spring. One inertia can be taken to represent the motor's rotor and another corresponds to the driven load such as fan and pump impeller. By comparing lateral vibration, torsional failures are even more sudden and serious without obvious signs. When the system is started and approaches to steady speed, the driving and load torque can twist the shaft from its free equilibrium position by a twist angle. If both the driving and load torque are suddenly removed from the system, the twisted shaft will uncoil and drive the two-inertia system oscillating in the opposite circumferential directions with respect to the shaft [9.31]. For an undamped system, the torsional vibration continues indefinitely as the system energy converts back and forth between the kinetic and potential energy. Unlike lateral vibration, torsional natural frequencies are unaffected by machine operation speeds.

In addition, because electric motors have discrete poles, their output torques do not develop smoothly but have periodic torque pulsations or torsional vibrations. These torque variations can produce periodic velocity variations or accelerations, causing torsional vibration.

9.5 Vibration Measurements

Vibration can be primarily measured in three detection schemes for displacement, velocity, and acceleration: (1) peak-to-peak, (2) zero-to-peak, and (3) RMS. The relationship of these parameters is displayed in Figure 9.20. Generally, acceleration emphasizes high frequencies, displacement emphasizes low frequencies, and velocity gives equal emphasis to all frequencies.

The selection of proper detectors depends on the characteristics of signals. For instance, RMS detectors are typically suitable for rotating machinery such as motors, gears, and turbines. Transient signals have irregular properties and usually have wide frequency band and low energy. Thus, adopting a peak detector is a good choice.

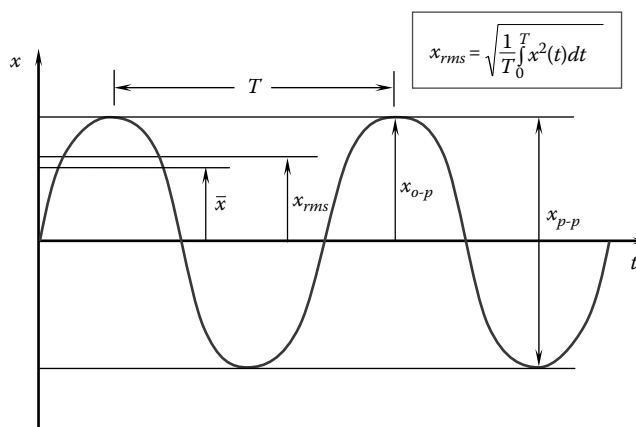


FIGURE 9.20

Comparison of average, RMS, zero-to-peak, and peak-to-peak displacements for a continuous sinusoidal signal.

Displacement is the most easily understood vibration parameter. There are several methods for measuring vibrational displacement depending upon applications. Accelerometers have long been used to measure displacement. During the measurement, the vibration levels are converted to electrical signals for data measurement by accelerometers. The benefits of using accelerometers to sense machine vibration through casing measurement include the following.

An eddy-current probe creates an alternating electromagnetic field with induced small current in the target object. Simultaneously, the eddy currents create an opposing magnetic field that resists the field being generated by the probe. The interaction of the magnetic fields is sensitive to the distance between the probe and the target object. If the distance changes, the magnetic coupling between the probe and the object is altered.

For rotating machinery, eddy-current probes are usually mounted in a casing to measure the location of the shaft relative to the casing and the amount of $1\times$ rotational vibration. Based on the measurement, it can be determined if the shaft vibration is within acceptable limits. Eddy-current probes can be also used for monitoring shaft eccentricity and thermal expansion in the axial direction. Eddy-current probes are useful for applications requiring noncontact measurements and not sensitive to environments and materials. However, this type of probes is not suitable for some conditions that require extremely high resolutions and large gap between probe and target object.

In recent years, there has been increasing interest in loop-powered vibration sensors for their simplicity and cost-effectiveness in continuous vibration monitoring. The sensor basically measures overall vibration in terms of velocity. The vibration signals taken by a loop-powered sensor are the same as an accelerometer.

Laser Doppler vibrometer (LDV) is one of the most efficient devices used in noncontact vibration measurements. This noncontact measuring technology is particularly useful for some applications where contacting vibration surfaces is difficult or even impossible. When a laser beam is directed to the measure surface from an LDV, the vibration velocity, frequency, and displacement are extracted by using the Doppler effect. For some small objects, the use of an LDV is especially desirable because there is no mass loading to the measure objects, as in other contact measurement methods.

Vibration and sound signals indicate the condition or quality of operating machines. In analyzing vibration in complicated mechanical systems, the Fourier transform is a powerful tool that takes a signal as a function of time (time domain) and decomposes it into a number of harmonic components as a function of frequency (frequency domain). This indicates that a signal can be viewed either in the time domain or in the frequency domain. The frequency domain representation is also called the spectrum of the signal, which can be used to determine the source of the vibration. Similarly, the spectrum of acoustic noise from rotating machinery can be used to design noise abatement systems. With the advanced computational techniques, the fast Fourier transform (FFT), as a digital implementation of Fourier transfer, has been developed. The FFT is the basic operation in frequency analysis, which is the most commonly used analysis method in many applications. Today, FFT-based signal-processing algorithms are widely used in signal processing.

The RMS method is often used to process vibration data. The RMS value of vibration signal is a time analysis feature that is the measure of the power content in the vibration signature. It can be very effective in detecting a major out-of-balance in rotating systems. A formula for the calculation of RMS value of a series of data is given as

$$x_{rms} = \sqrt{\frac{1}{N} \sum_{i=1}^N x_i^2} \quad (9.104)$$

where N is the number of data. For a continuous signal $x(t)$, RMS values can be calculated as

$$x_{rms} = \sqrt{\frac{1}{T} \int_0^T x^2(t) dt} \quad (9.105)$$

where T is the period.

9.6 Vibration Control

Vibration can be reduced by increasing either the damping capacity or the stiffness. The loss modulus is the product of these two quantities and thus can be considered a figure of merit for the vibration reduction ability [9.32].

There are many classical forms of damping, including hysteresis (structural) damping, fluid viscosity damping, Coulomb friction damping, air damping, particle damping, and magnetic and piezoelectric damping, to name a few. Each form of damping exhibits its own damping characteristics and is suitable to certain applications. In industrial applications, the most common form of damping is passive-based viscoelastic damping, used for solving a variety of noise- and vibration-related problems.

9.6.1 Damping Materials

In practice, damping can be characterized by two damping parameters: (1) loss modulus, which is defined as the product of the damping capacity and the stiffness and (2) loss tangent, which is defined as the ratio of the imaginary part to the real part of the complex shear modulus.

Damping is inherent in all resilient and viscoelastic materials. Viscoelastic materials encompass a broad range of materials, including pressure-sensitive adhesives, epoxies, rubbers, foams, thermoplastics, enamels, and mastics. The common characteristics of these materials are that their modulus is represented by a complex quantity, possessing both stored and dissipative energy components [9.33].

As presented by Chung [9.34], damping materials fall into four categories: (1) materials exhibiting high loss modulus but low loss tangent (such as cast iron and shape-memory alloys); (2) materials exhibiting low loss modulus but high loss tangent (such as rubber and other polymers); (3) materials exhibiting low values of both loss tangent and loss modulus; and (4) materials exhibiting high values of both loss tangent and loss modulus (such as graphite networks and cement-matrix composite). High damping materials must have high values of both loss tangent and loss modulus, that is, high damping capacity and high stiffness. For metallic materials, magnesium alloys have comparatively high damping capacity, up to about 3 times that of cast iron and up to about 30 times higher than that

of aluminum, and a high strength-to-density ratio. The combination of high damping, good strength, and low mass makes magnesium alloys an excellent choice for vibration damping materials [9.35].

A number of investigators have studied the vibration damping ability for a variety of materials. Based on the review of 34 reference papers, Birchak [9.36] compared specific damping capacity of 17 metals and 2 plastics, including cast iron, steel, brasses, and magnesium alloys. Schetsky [9.37] presented the values of specific damping capacity for 20 metals, from 49% of magnesium to 0.2% of aluminum, nickel, and titanium alloys. In his article, specific damping capacity is based on decay rate of strain energy.

Generally, damping is inversely proportional to temperature due to the crystallinity and viscosity in elastomers.

9.6.2 Vibration Isolation

Many devices and systems exist that incorporate features to counteract the effect of vibration on electric machines. A common approach to mitigate vibration of rotating machinery is to employ vibration isolation systems, which are used to filter out undesirable vibrations by modifying the vibration transmissibility. Good vibration isolation can be achieved by supporting the vibration source on a flexible low-frequency mounting. Thus, though exciting forces act on the system, only a small proportion of the forces can be transmitted to the support structure.

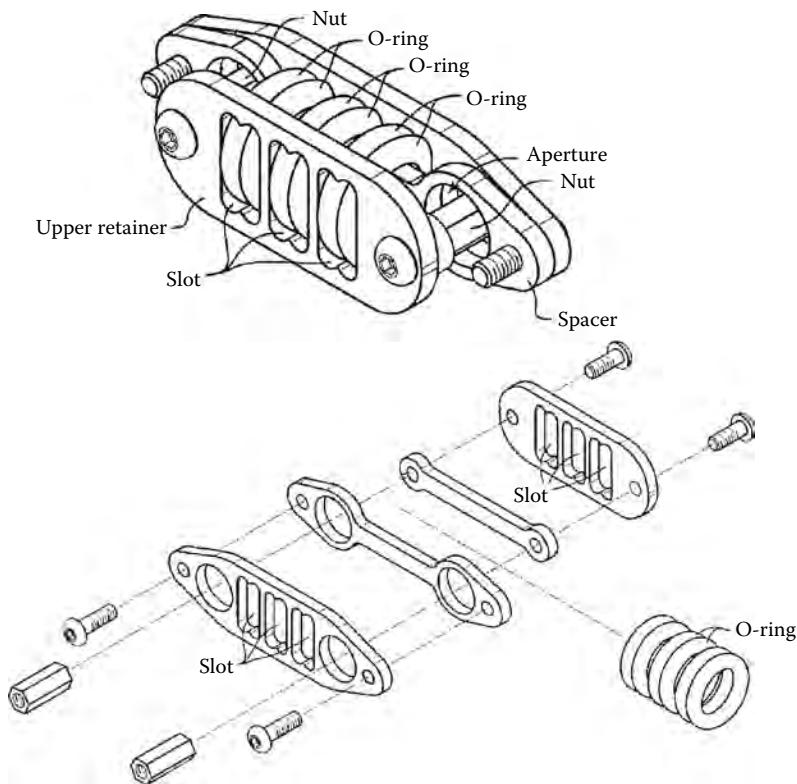
According to control schemes, vibration isolation systems can be categorized in passive, active, or semiactive type. Among them, passive isolating is extensively used in industries.

A typical example of passive isolation system is a suspension system (as the vibration isolator and shock absorber) employed in a vehicle for isolating the vehicle body from the wheel axels. Incorporating vibration isolation into motor design follows the same principles. To reduce the vibration transmitted from a motor to the ground, the motor may be directly mounted on isolators. Various isolators are available today, from isolation pads to coil spring isolators with integral vibration damping materials.

As the example shown in Figure 9.21, a vibration reduction device is revealed recently in a US patent application. This device consists of a number of O-rings as the resilient vibration dampeners and an upper and lower assembly. The removable upper and lower assemblies are coupled to one another, and the O-rings are positioned between these two assemblies to provide the capability to *tune* the device to a desired stabilization condition. This feature offers the ability to reduce or isolate vibration caused by motors [9.38].

An innovative idea is to use a multilevel magnetic system for vibration isolation [9.39]. The system includes upper and lower magnetic structures and transitions between an attract mode and a repel mode when the upper and lower magnetic structures are separated by an equilibrium separating distance. The multilevel magnetic system is placed between two objects and configured to oscillate about the equilibrium separating distance in response to a vibration from a motion source. The oscillation about the equilibrium separating distance causes the magnetic system to function as a low-pass filter that substantially attenuates vibration above the cutoff frequency, thus limiting the conduction of the vibration between the two objects.

Figure 9.22 depicts a disk-shaped repel-snap multilevel magnetic system. As shown in the figure, this system consists of upper and lower magnetic structures. Each magnetic structure has a region of coded maxels and a region of polarity. Each maxel has a code or

**FIGURE 9.21**

Vibration isolation system used to attenuate machinery vibration (U.S. Patent Application 2013/0026689) [9-38]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

pattern of positive or negative polarity. The maxels are in seven rows for each structure, and each maxel for each row in one structure has polarity that is opposite a corresponding maxel in corresponding row in another structure. When a maxel is magnetized entirely through a magnetizable material the opposite polarities occur at the two sides of the maxel because each maxel is essentially a dipole magnetic source. Similarly, the polarity regions of the two structures have opposite polarities since they are also a dipole magnetic source. When the attractive force equals the repelling force, the system is configured to oscillate about the equilibrium separation distance in response to a vibration.

As demonstrated in Figure 9.23, a mechanical isolation system comprises four mechanical isolators comprising the contactless attachment multilevel magnetic system. These mechanical isolators are posited between the first and second objects to control mechanical impedance between the two objects.

Viscoelastic material damping pads have been designed for passive vibration damping. As shown in Figure 9.24, a laminated damping pad consists of thin viscoelastic films (0.05–0.3 mm in thickness) sandwiched between stainless steel plates (about 1.8 mm in thickness). Among various viscoelastic materials, polymers are high-energy dissipative materials that have been widely used for many years in automotive, aerospace, and electronic industries for solving complex vibration and noise problems.

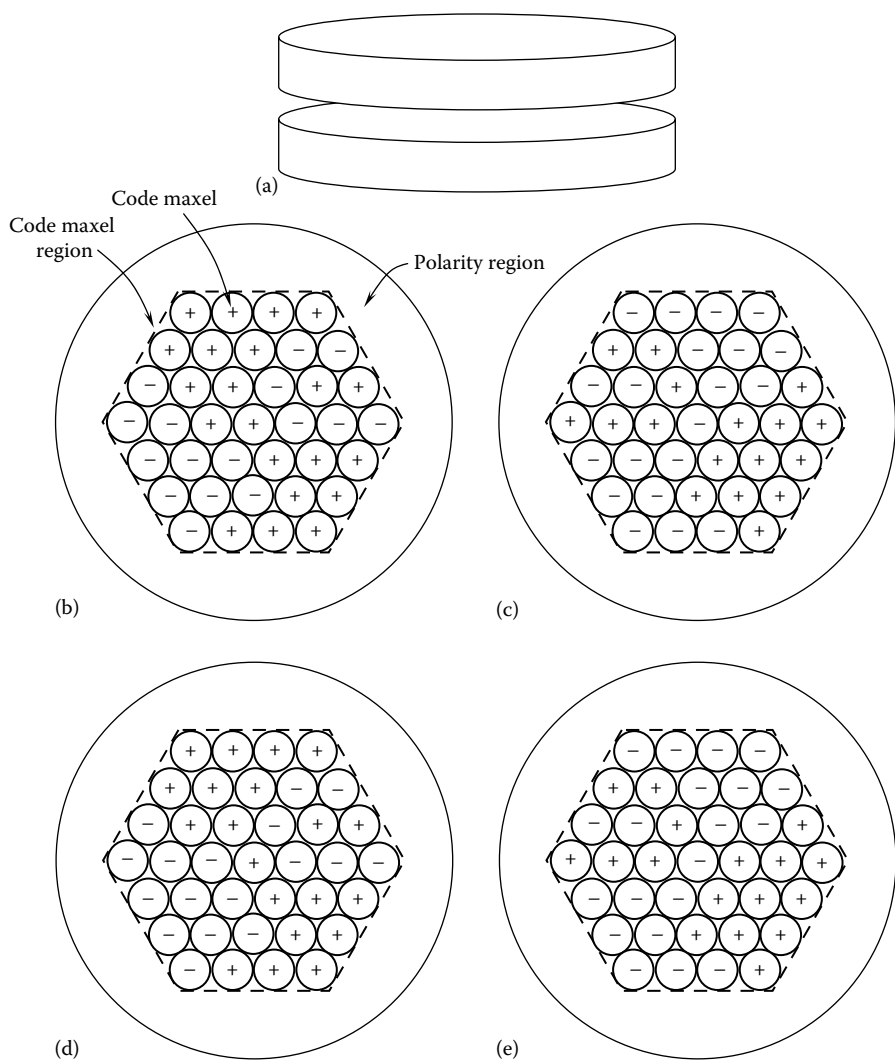
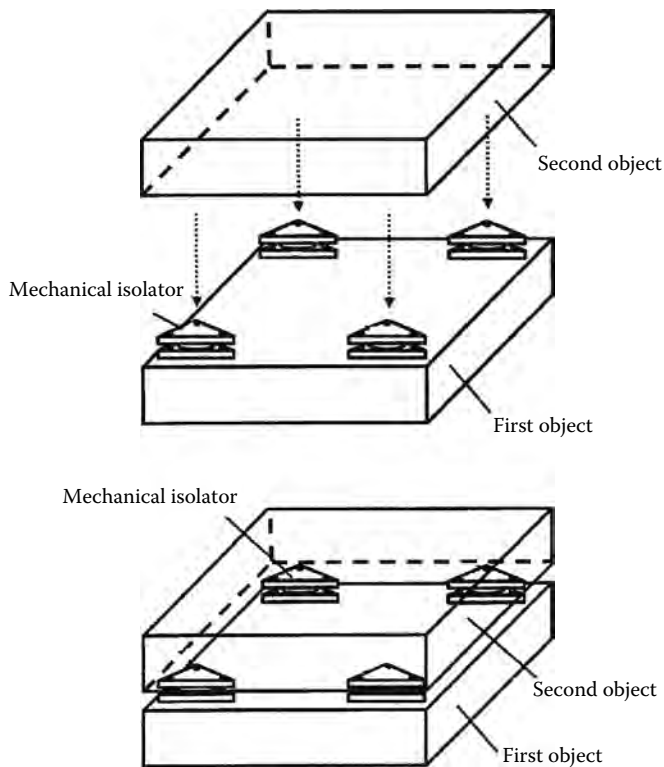


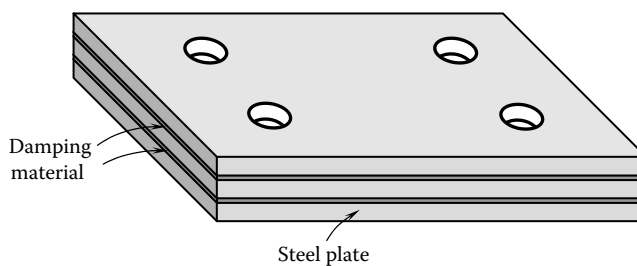
FIGURE 9.22 Multilevel magnetic system: (a) disk-shaped repel-snap magnetic system separated by an equilibrium separating distance, (b) a top view of the upper magnetic structure, (c) a bottom view of the upper magnetic structure, (d) a top view of the lower magnetic structure, and (e) a bottom view of the lower magnetic structure.

Viscoelastic materials are characterized by their shear modulus G , and loss factor η , which is defined as the ratio of vibration energy dissipated per cycle to the energy stored in all structural elements. It has been found that the performance of damping pads depends not only on the material properties (such as G and η) but also on the machine's complex interaction with the structural system to be damped. Results from Magra et al. [9.40] show that damping pads can reduce the vibration amplification factor (which is the reciprocal of η) from over 100 to 8 for an advanced photon source machine.

Molded and bonded rubber is often used as machinery mounts to absorb shock and attenuate vibration.

**FIGURE 9.23**

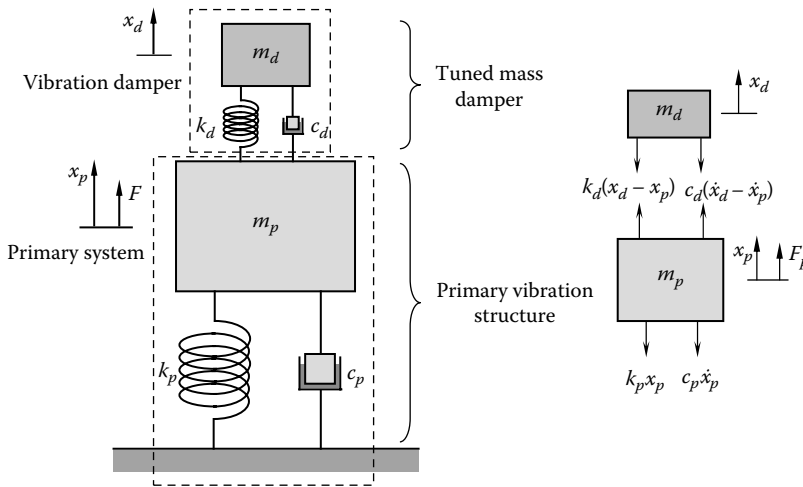
Mechanical isolation systems assembled into the multilayer magnetic system (U.S. Patent 8,279,031) [9.39]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

**FIGURE 9.24**

Laminated damping pad for providing effective damping of electric machine.

9.6.3 Tuned Mass Damper

Rotating machines such as motors and engines often incite vibration due to rotational imbalances. A tuned mass damper (TMD) is a passive damping device that utilizes the secondary mass attached to the primary structure to reduce the dynamic response of the primary structure. Figure 9.25 depicts the schematic of a two-degree-of-freedom TMD

**FIGURE 9.25**

Two-degree-of-freedom TMD system; TMD is attached to the primary system for reducing dynamic response of the primary system.

system. It can be seen that a primary system (i.e., a motor) consists of a mass m_p , a spring (with spring coefficient k_p), and a viscous damper (with damping coefficient c_p). A TMD, or harmonic absorber, consists of a mass m_d , a spring (with the spring factor k_d), and a viscous damper (with damping coefficient c_d) and is attached to the primary system to reduce or eliminate the undesirable vibration of the primary system. Usually, m_d is chosen approximately 5%–10% of the system mass m_p . To obtain the best damping effect, TMDs set their natural frequencies substantially equal to the natural frequencies of the primary structures. It has been proven that the TMD is effective in reducing the response of structures to harmonic or wind excitations [9.41].

From the free-body diagram of the mass m_p and m_d , the equations of motion of the system can be written as

$$m_p \frac{d^2 x_p(t)}{dt^2} + (c_p + c_d) \frac{dx_p(t)}{dt} - c_d \frac{dx_d(t)}{dt} + (k_p + k_d)x_p - k_d x_d = F \quad (9.106)$$

$$m_d \frac{d^2 x_d(t)}{dt^2} + c_d \frac{dx_d(t)}{dt} - c_d \frac{dx_p(t)}{dt} + k_d x_d - k_d x_p = 0 \quad (9.107)$$

These two equations of motion can be expressed in the matrix form

$$\begin{bmatrix} m_p & 0 \\ 0 & m_d \end{bmatrix} \begin{Bmatrix} \ddot{x}_p \\ \ddot{x}_d \end{Bmatrix} + \begin{bmatrix} c_p + c_d & -c_d \\ -c_d & c_d \end{bmatrix} \begin{Bmatrix} \dot{x}_p \\ \dot{x}_d \end{Bmatrix} + \begin{bmatrix} k_p + k_d & -k_d \\ -k_d & k_d \end{bmatrix} \begin{Bmatrix} x_p \\ x_d \end{Bmatrix} = \begin{Bmatrix} F \\ 0 \end{Bmatrix} \quad (9.108)$$

It can be seen that $[m]$, $[c]$, and $[k]$ are all 2×2 matrices whose elements are the known masses, damping coefficients, and stiffnesses of the system, respectively. Because each of these matrices is symmetric, the transpose of the matrix is equal to its original matrix, that is, $[m]^T = [m]$, $[c]^T = [c]$, and $[k]^T = [k]$.

The external exciting force F is considered to vary harmonically with the frequency ω :

$$F = F_o e^{i\omega t} \quad (9.109)$$

The solutions for the equations of motion may take the form

$$\begin{cases} x_p(t) = X_p e^{i\omega t} \\ x_d(t) = X_d e^{i\omega t} \end{cases} \quad (9.110)$$

Substituting Equations 9.109 and 9.110 into the set of governing equation results in

$$\begin{bmatrix} -m_p \omega^2 + (c_p + c_d)i\omega + (k_p + k_d) & -c_d i\omega - k_d \\ -c_d i\omega - k_d & -m_d \omega^2 + c_d i\omega + k_d \end{bmatrix} \begin{Bmatrix} X_p \\ X_d \end{Bmatrix} = \begin{Bmatrix} F_o \\ 0 \end{Bmatrix} \quad (9.111)$$

Introducing the natural frequency ω and damping ratio ζ for the primary system and tuned mass system, respectively,

$$\omega_p = \sqrt{\frac{k_p}{m_p}} \quad (9.112)$$

$$\zeta_p = \frac{c_p}{2\sqrt{m_p k_p}} \quad (9.113)$$

$$\omega_d = \sqrt{\frac{k_d}{m_d}} \quad (9.114)$$

$$\zeta_d = \frac{c_d}{2\sqrt{m_d k_d}} \quad (9.115)$$

and defining $\bar{m}$ as the mass ratio,

$$\bar{m} = \frac{m_d}{m_p} = \frac{k_d}{k_p} \frac{\omega_p^2}{\omega_d^2} \quad (9.116)$$

the equations of motion can be rewritten in the matrix form as

$$\begin{bmatrix} -\omega^2 + 2(\zeta_p \omega_p + \zeta_d \omega_d \bar{m})i\omega + (\omega_p^2 + \omega_d^2 \bar{m}) & (-2\zeta_d \omega_d i\omega - \omega_d^2) \bar{m} \\ -2\zeta_d \omega_d i\omega - \omega_d^2 & -\omega^2 + 2\zeta_d \omega_d i\omega + \omega_d^2 \end{bmatrix} \begin{Bmatrix} X_p \\ X_d \end{Bmatrix} = \begin{Bmatrix} F_o/m_p \\ 0 \end{Bmatrix} \quad (9.117)$$

As measured previously, in order to obtain the best damping effect, the natural frequency of the mass damper ω_d is equal to the natural frequency of the primary system ω_p :

$$\omega_p = \omega_d = \omega \quad (9.118)$$

Substituting Equation 9.118 into 9.117, it yields

$$\begin{bmatrix} 2(\zeta_p \omega^2 + \zeta_d \omega^2 \bar{m})i + \omega^2 \bar{m} & (-2\zeta_d \omega^2 i - \omega^2) \bar{m} \\ -2\zeta_d \omega^2 i - \omega^2 & 2\zeta_d \omega^2 i \end{bmatrix} \begin{Bmatrix} X_p \\ X_d \end{Bmatrix} = \begin{bmatrix} F_o/m_p \\ 0 \end{bmatrix} \quad (9.119)$$

Hence, X_p and X_d are determined as

$$X_p = \frac{F_o}{m_p} \frac{C}{AC - B^2 \bar{m}} \quad (9.120)$$

$$X_d = -\frac{F_o}{m_p} \frac{B}{AC - B^2 \bar{m}} \quad (9.121)$$

where

$$A = 2(\zeta_p \omega^2 + \zeta_d \omega^2 \bar{m}) + \omega^2 \bar{m}$$

$$B = -2\zeta_d \omega^2 i - \omega^2$$

$$C = 2\zeta_d \omega^2 i$$

The solutions of the motion equations are thus obtained by substituting Equations 9.120 and 9.121 into 9.110.

TMDs commonly use metal coil springs and viscous dampers. They can also use other types of dampers, such as viscoelastic, air-suspended, sloshing water, and liquid column dampers. When designed, fabricated, installed, and tuned properly, TMDs can effectively absorb the vibrating energy of the structure and dissipate energy internally and hence reduce structural vibrations of machines or buildings. The 509 m (11,617 ft) supertall Taipei 101 building installed a 730 ton (0.26% of the building mass) TMD, which is the largest TMD in the world. And it is also the first ever constructed as a key architectural and visible element in the building [9.42].

TMDs can be essentially viewed as energy sinks. It was reported that with the use of the dynamic vibration absorber on a lightly damped system that have steplike trajectories or disturbances, the settling time has been greatly improved by 82% [9.43].

9.6.4 Double Mounting Isolation System

Greater attenuation of the exciting force at high frequencies can be achieved by using a double mounting, also known as two-stage mounting. In this arrangement, the machine is set on flexible mountings on an inertia block, which is supported by flexible mountings [9.44].

Figure 9.26 demonstrates the schematic of a two-degree-of-freedom double mounting system. The system consists of a primary mass m_p (machine mass) and an auxiliary mass m_a (damper mass). Dampers are placed between the primary m_p and auxiliary mass m_a and between the auxiliary mass m_a and the base. The purpose of using the system is to isolate either the base from the vibration of the machine or the machine from the vibration of the base (e.g., during an earthquake). The system is also used where there is a demand for high structure-borne noise attenuation.

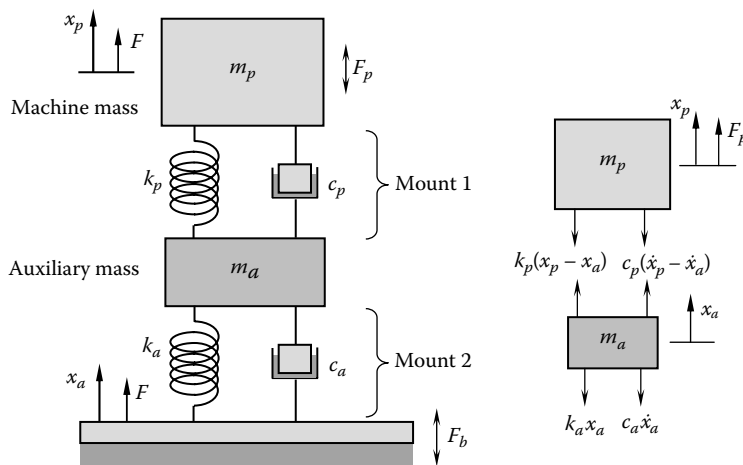


FIGURE 9.26
Two-degree-of-freedom double mounting damping system.

While the passive isolating systems are very effective in lowering the vibration transmission at high frequencies, they suffer from some drawbacks including design complexity, weight penalty, large volume, and high cost. Without any constraints, a double mounted system has totally 12 degrees of freedom, twice of the unconstrained single mounted system. Accordingly, it has 12 resonant frequencies. As a matter of fact, keeping all resonant frequencies away from forcing frequencies of a machine is a major design challenge for design engineers.

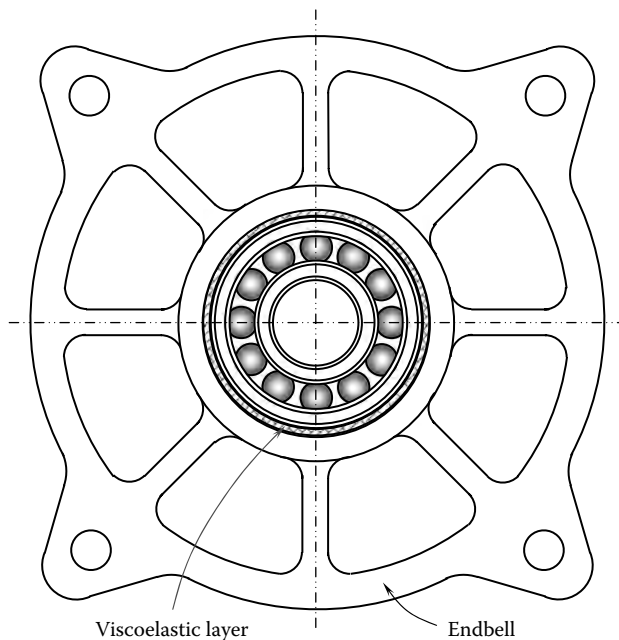
9.6.5 Viscoelastic Bearing Support

In many cases, the rotor vibratory loads are the dominant source of motor vibration. The mechanism of rotor vibration has been extensively studied and is well understood. As discussed previously, a variety of factors can impact the vibration amplitude and frequency such as rotor mass imbalance, broken rotor bar, and UMP, to name a few. Rotor structural vibration is transmitted through the bearings and then to the stator, motor housing, and finally to the motor base.

In order to interrupt the transmission path of rotor vibration, it is highly desired to design a viscoelastic bearing support that has a viscoelastic layer between the outer ring of a rolling bearing and the motor endbell [9.45], as demonstrated in Figure 9.27. In this way, vibration of the rotor is isolated from the surrounding structure. Moreover, the viscoelastic layer provides additional damping to the vibrating rotor. However, a relative soft viscoelastic bearing support can reduce the static stiffness of the machine to influence the machine operation accuracy.

9.6.6 Active Vibration Isolation and Damping

Although passive vibration isolation systems are still extensively used in industries today, they are inadequate for some applications that require high accurate control of vibration, such as optical devices, sonars, piezoelectric crystal flowmeters, medical detection devices, space structures, and other vibration-sensitive equipment.

**FIGURE 9.27**

Viscoelastic layer positioned between the bearing outer ring and motor endbell for isolating rotor vibration from the motor and its base.

Active vibration isolation systems use external energy to directly cancel energy in systems. The system involves the use of actuators along with sensors and controllers to create actuation with the goal of reducing the transmission of vibration from the vibration source to the rest of the components in the system. Sensors are used to continuously monitor the dynamic motion of the target system and send the relevant motion data (relative displacement, velocity, acceleration, etc.) to controllers. The controllers calculate the required external forces or displacements and send the signals to actuators, which provide the desired forces or displacements to the target system, as illustrated in Figure 9.28. This method can work effectively for low vibrating frequencies. However, it is very difficult to apply it to high-frequency applications.

A semiactive vibration isolation system is presented in Figure 9.29. It combines the features of a passive and an active vibration isolation system. Unlike in an active system, the actuator in the semiactive system is treated as a passive element. The properties of the semiactive actuator such as damping ratio and stiffness can vary so that the control can be implemented without adding external energy into the system, except a small amount of energy required to change the properties of the actuator. In the semiactive system, the controller is used to determine the desired properties of the actuator, and the sensor functions the same in an active system to detect the motion of the target system [9.46].

Generally, the low-frequency isolation in the horizontal direction can be easily achieved by a passive system since it requires less concern about the horizontal static load by having a low stiffness. A vertical isolation is well implemented by an active system because it can provide a high stiffness to support static loads. Platus [9.47] has developed a passive vibration isolation system that outperforms conventional passive systems by adopting a technique called negative stiffness mechanism (NSM). The use of this system in a vertical isolation reduces the vertical stiffness and hence natural frequencies while maintaining

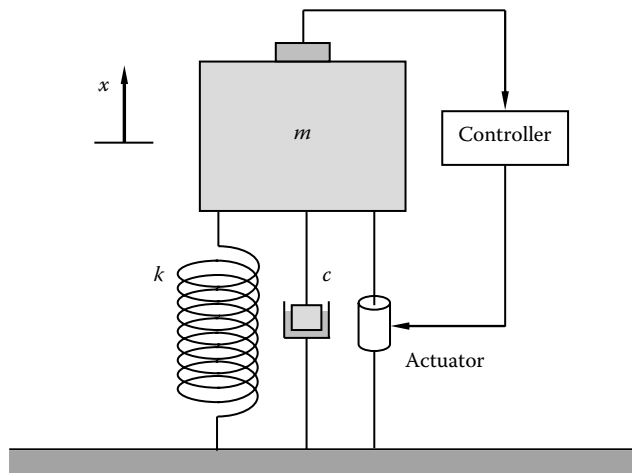


FIGURE 9.28
Active vibration isolation system.

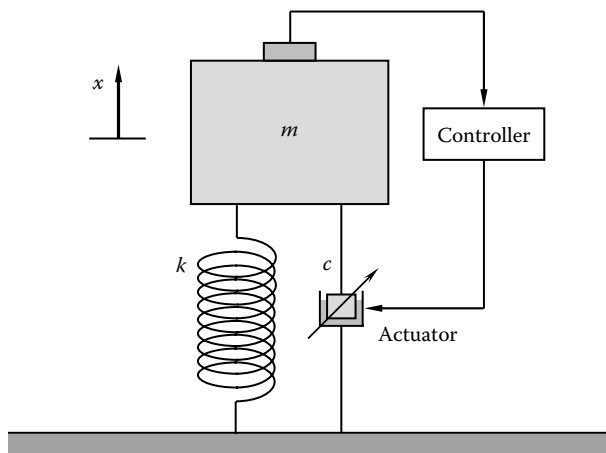


FIGURE 9.29
A semi-active vibration isolation system. The controller is used to adjust actuator damping characteristics.

static load-supporting capability. It was reported that the vertical natural frequencies can be tuned to as low as 0.2 Hz.

Active vibration damping is to actively reduce the response amplitude of the system within a limited bandwidth near the natural frequencies of the system. It offers a promise of high efficiency without the restrictions of passive methods. Active vibration control involves monitoring vibrations of a structure and utilizing the vibration signal to generate a force with the proper phase and amplitude to attenuate the vibration. An additional advantage of an active approach is the ability to supply a vibration signal that can be used independently for monitoring the vibration environment.

In some high-precision industrial applications (e.g., optical tables), vibration isolation methods are usually used for relatively low-frequency vibration problems. For high-frequency vibrations, passive damping, either broadband or tuned, provides varying levels

of structural damping performance, but both are affected by applied loads. By means of vibration sensors, actuators, and controllers, active vibration damping can be autotuned to account for varying loads [9.48], making it more attractive for industrial and medical end users.

An alternative method for active vibration isolation is to use noncontacting electromagnetic voice coil actuators (VCAs), which are used for some special applications like large telescopes and robots. In this method, electric current passing through the voice coil generates a magnetic field that interacts with the magnetic field generated by PMs to produce desired forces [9.49]:

$$F = n\pi diB \sin \theta \quad (9.122)$$

where

n is the total number of coils

d is the average diameter of the coil

i is the current through the wire

B is the magnetic field strength

θ is the angle between the magnetic flux lines and the direction of the current

Newell et al. [9.50] have used VCAs to construct an active vibration isolation system for earth-based interferometric gravitational wave detectors. This system provides vibration isolation in all 6 degrees of freedom by at least 40 dB at 0.5 Hz.

9.6.7 Measurements of Motor Vibration

A comprehensive assessment of motor vibration requires measurements of vibration acceleration in the unit of m/s^2 . A typical vibration measurement system comprises a number of vibration sensors (e.g., accelerometers, velocity transducers, and proximity probes) for sensing vibration velocity or acceleration; a vibration analyzer for data acquisition, bearing condition diagnosis, and spectrum diagnosis; and a computer for processing the measurements and displaying the results. The vibration sensors are typically attached to the vibrating motor housing at different locations and on the baseplate. Since motor bearings are the load-carrying components of the mechanical drive train, the vibration sensors should be placed close to bearings to measure both motor radial and axial vibrations (Figure 9.30). The vibration sensors (used accelerometers here) produce

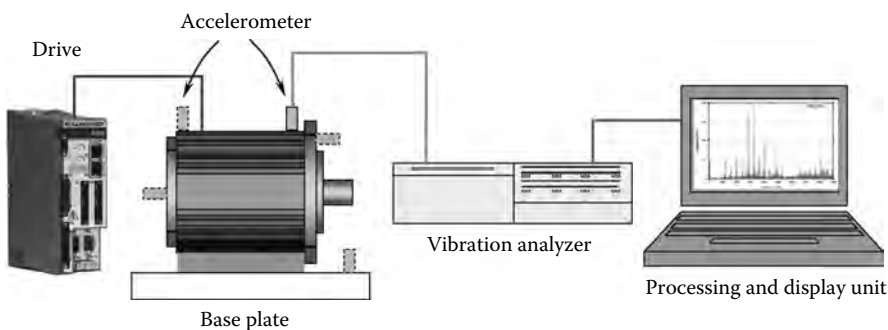
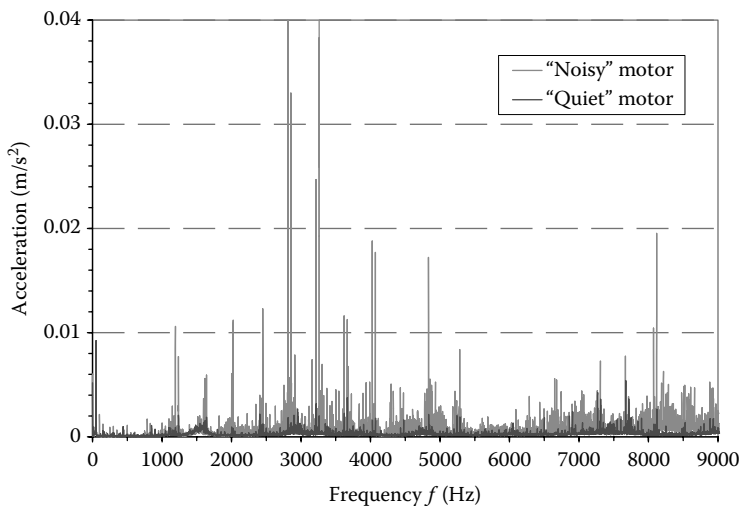


FIGURE 9.30

Motor vibration measurement using accelerometers for assessing motor vibration and diagnosing vibration root causes.

**FIGURE 9.31**

Comparison of measured vibration data from two PM servomotors with the same brand and same model, showing distinct differences in harmonic components of the acceleration between two motors. Courtesy of Kollmorgen Corporation.

electrical signals that are proportional to the vibrating acceleration on the measure surface. Then, the measure data are transmitted to a vibration analyzer for data processing and data recording. Finally, the measured vibration results are displayed by a computer to provide engineers for the further analysis. In such a way, mechanical and electrical defects can be identified through vibration analysis. The useful information detected from the motor vibration measurements includes bearing vibration, rotordynamic unbalance, eccentricity, and electrical faults.

As an example, the results of motor vibration for two identical servomotors are presented in Figure 9.31. For the purpose of comparison, a *noisy* motor and a *quiet* motor were selected for measuring. It can be observed that there are great differences in vibrating accelerations between the two motors for almost all frequencies, especially in the range of 1000–5000 Hz, and $f > 8000$ Hz. The root cause analysis reveals that the motor bearings and assembly quality attribute to high vibration.

A special attention must be put on keyed shafts in vibration measurements. Prior to the test, a *half key* must be placed in the shaft keyway to eliminate the influence of the unbalance that is induced from the keyway.

9.7 Fundamentals of Acoustic Noise

Noise is one of the most common occupational health hazards. In heavy industrial and manufacturing environments, permanent hearing loss is the main health concern. In the United States, growing environmental concerns and recognition that lengthy unprotected exposure to high industrial noise levels is detrimental to man have resulted in government

promulgation of noise level criteria. In many countries, especially European countries, noise reduction is an integral part of machinery safety, and law regulates the allowable noise exposure of workers. Manufacturers must make specific quantitative information on noise emitted under designed operating conditions. In recent years, the noise requirement to ensure quiet working conditions is getting even tighter.

Acoustic noise is the result of the pressure waves produced by vibration sources. Generally, a sinusoidal wave is referred to as a tone, a combination of several tones is called a sound, and an irregular vibration is referred to as noise [9.51]. Noise is defined in terms of frequency spectrum (in Hz), intensity (in dB), and time duration.

9.7.1 Tonal Noise and Broadband Noise

There are two types of noise radiation: pitched or unpitched noise. Pitched noise, also known as tonal noise, is created from vibrations of an object that occur in a fine range around one or very few distinct frequencies. Tonal noise can be extremely irritating to human hearing. An electric motor that emits tonal noise has a characteristic frequency spectrum. Unpitched noise, also known as broadband noise, has a well-distributed energy spectrum. Broadband noise is generally not irritating to human hearing and does not transmit a characteristic frequency spectrum, such as wind noise.

9.7.2 Sound Pressure Level and Sound Power Level

Sound pressure is a measure of air pressure fluctuation a noise source creates. It is usually expressed in a unit of Pascal (Pa). However, the use of Pascal as the sound pressure unit may generate a broad range of sound pressures. A convenient way to compress the scale of numbers into a manageable range is to use decibel (dB). Sound pressure converted to the decibel scale is called sound pressure level (SPL), denoted as L_p :

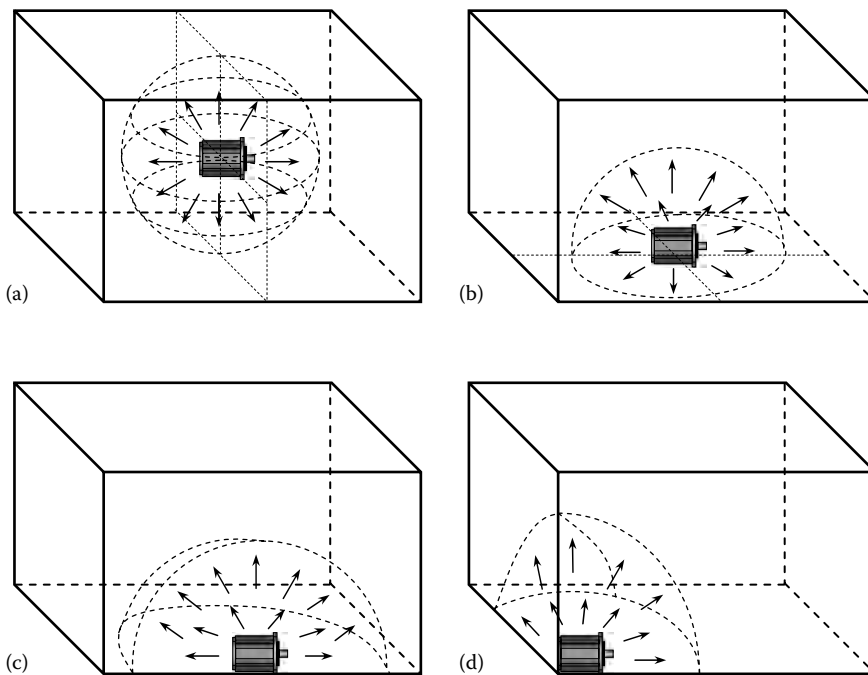
$$L_p = 10 \log_{10} \left(\frac{p_{rms}^2}{p_o^2} \right) = 20 \log_{10} \left(\frac{p_{rms}}{p_o} \right) \quad (9.123)$$

where p_{rms} is the RMS sound pressure being measured and p_o is the reference sound pressure, typically 20 μ Pa.

It is important to distinguish between sound power and sound pressure. Sound power is a measure of the total amount of sound energy emitted by the source over a period of time, usually expressed in watts. Sound pressure depends on the distance of observation location from the source but sound power does not. The sound power level L_w , in unit of dB, is defined by

$$L_w = 10 \log_{10} \left(\frac{W}{W_o} \right) \quad (9.124)$$

where W is the sound power emitted by the source in watts and W_o is the reference power, typically 10^{-12} W. SPL L_p can be related to sound power level L_w as [9.52]

**FIGURE 9.32**

Directivity factor of the sound source for different source positions: (a) at room center, $Q_0=1$; (b) at the center of floor, $Q_0=2$; (c) at the center of floor edge, $Q_0=4$; and (d) at floor corner, $Q_0=8$.

$$L_p = L_w + 10 \log \left(\frac{Q_0}{4\pi r^2} + \frac{4}{R_c} \right) \quad (9.125)$$

where

r is the distance from the source

Q_0 is the directivity factor of the source in the direction of r , which depends on the position of the sound source (see Figure 9.32)

R_c is the room constant, defined as

$$R_c = \frac{A\bar{\alpha}}{1 - \bar{\alpha}} \quad (9.126)$$

where

A is the total surface area of the room

$\bar{\alpha}$ is the average sound absorption coefficient of the room

9.7.3 Octave Frequency Bands

The audible frequency range is usually divided into 10 octave bands having center frequencies at 31.5, 63, 125, 250, 500, 1,000, 2,000, 4,000, 8,000, and 16,000 Hz. The center frequency of each consecutive octave band is twice the center frequency of the previous one. In each octave band, the upper cutoff frequency is twice the lower cutoff frequency. Usually, an electric machine such as motor and generator generates and radiates noise over

the entire audible range of hearing. The amount and frequency distribution of the noise can be obtained using an octave band analyzer. This analyzer equips a set of contiguous filters, and each filter has a bandwidth of an octave. Therefore, ten such filters cover the most frequency range of interest. The measured L_p data at each octave band are very useful for determining the noise root sources.

Each octave band can be further divided into three one-third octave bands. The SPL L_p is thus determined from three measure values $L_{p,i}$ that are taken at each one-third octave band, respectively.

The comparison of full octave band and one-third octave band is presented in Table 9.1. The relationship between the upper cutoff frequency f_{n+1} and lower cutoff frequency f_n is

TABLE 9.1

Comparison of One Octave Band and One-Third Octave Band in Audible Sound Range

Band	One Octave			One-Third Octave		
	Lower Cutoff Frequency (Hz)	Middle Frequency (Hz)	Upper Cutoff Frequency (Hz)	Lower Cutoff Frequency (Hz)	Middle Frequency (Hz)	Upper Cutoff Frequency (Hz)
1	22	31.5	44	22.4	25	28.2
				28.2	31.5	35.5
				35.5	40	44.7
2	44	63	88	44.7	50	56.2
				56.2	63	70.8
				70.8	80	89.1
3	88	125	177	89.1	100	112
				112	125	141
				141	160	178
4	177	250	355	178	200	224
				224	250	282
				282	315	355
5	355	500	710	355	400	447
				447	500	562
				562	630	708
6	710	1,000	1,420	708	800	891
				891	1,000	1,122
				1,122	1,250	1,413
7	1,420	2,000	2,840	1,413	1,600	1,778
				1,778	2,000	2,239
				2,239	2,500	2,818
8	2,840	4,000	5,680	2,818	3,150	3,548
				3,548	4,000	4,467
				4,467	5,000	5,623
9	5,680	8,000	11,360	5,623	6,300	7,079
				7,079	8,000	8,913
				8,913	10,000	11,220
10	11,360	16,000	22,720	11,220	12,220	14,130
				14,130	16,000	17,780
				17,780	20,000	22,390

$$\frac{f_{n+1}}{f_n} = 2^k \quad (9.127)$$

where $k=1$ for one octave band and $k=1/3$ for one-third octave band. The middle frequency f_o in each full octave band is about $\sqrt{2}$ times the lower cutoff frequency in that band. The band width w_b of each band is given as

$$w_b = f_{n+1} - f_n = (2^k - 1)f_n \quad (9.128)$$

9.7.4 Three Sound Weighting Scales

Human hearing varies in sensitivity for different acoustic frequencies. Although the audible range of acoustic frequencies generally ranges from 20 to 20,000 Hz, humans usually are most sensitive to pure tones at frequencies between 2,000 and 6,000 Hz. Hearing sensitivity drops off above 7000 Hz and below about 200 Hz. Thus, to establish a uniform noise measurement that simulates human's perception and annoyance, several frequency weighting methods are developed to account for those frequencies most audible to the human hearing range.

Acoustic frequency weightings refer to different sensitivity scales for noise measurement. In the development of sound level meters over the years, manufacturers have built in the different response curves, that is, A-, B-, and C-weighting filter curves, as defined in the IEC standard 61672: 2003 [9.53] and various national standards, as shown in Figure 9.33. Among these, A- and C-weighting scales are commonly incorporated into commercially marketed sound level meters.

The A-weighting scale is presently the most commonly used weighting scale in measuring industrial noise, because it best predicts the damage risk of human ear. Using the decibel A filter, the sound level meter is less sensitive to very low and very high frequencies.

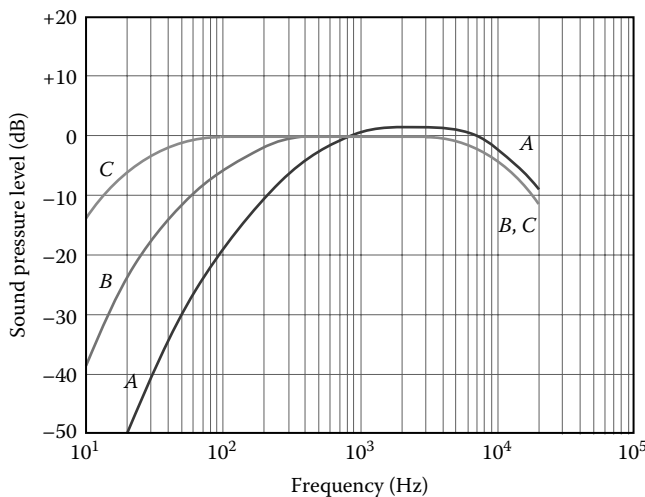


FIGURE 9.33

A-, B-, and C-weighting filter curves, defined up to 20 kHz.

Sound pressure measurements with this scale are expressed as dB(A) or simply dBA. The A-weighting curve in Figure 9.33 suggests that for most listeners, the noise at 100 Hz would sound about 19 dB quieter than that at 1000 Hz at the same SPL. It is worth to note that since the dBA scale is logarithmic, every increase of 10 dBA doubles the perceived loudness of hearing.

The B-weighting scale was used in the early time for predicting performance of loudspeakers and stereos, but today, the B-weighting scale is no longer in any international standard and already has very little practical use.

The C-weighting scale was originally designed to predict the human ear's sensitivity to tones at high noise levels. However, the ear's loudness sensitivity for tones is not the same as the ears' damage risk for noise. Much of the low-frequency noise is actually being filtered out by the ear, making it less likely to cause damage. By comparing with the A-weighting scale, the C-weighting curve is quite flat, indicating it contains much more of the low-frequency range of sounds.

9.7.5 Averaged Sound Pressure Level

The averaged SPL is the logarithm average of the measured SPL around the motor, expressed as

$$\bar{L}_p = 10 \log_{10} \frac{1}{n} \left[\sum_{i=1}^n 10^{(L_{p,i}/10)} \right] \quad (9.129)$$

where

$L_{p,i}$ is the measured SPL at i th point
 n is the total number of the points

It must be noted that $L_{p,i}$ may consist of noise produced by sources in the vicinity of the testing motor in a field test. In order to gain the SPL only produced by the motor, the background noise must be measured and subtracted from the measured noise data. This can be done by disconnecting the test motor and maintaining all other equipment operating. Then, by repeating the measurements around the motor, the background noise pressure level at i th point $L_{p,i,b}$ is obtained. As a result, the noise pressure level at i th point from the tested motor can be calculated as

$$L_{p,i,m} = 10 \log_{10} \left[10^{(L_{p,i}/10)} - 10^{(L_{p,i,b}/10)} \right] \quad (9.130)$$

Thus, the averaged SPL only from the tested motor is obtained:

$$\bar{L}_{p,m} = 10 \log_{10} \frac{1}{n} \left[\sum_{i=1}^n 10^{(L_{p,i,m}/10)} \right] \quad (9.131)$$

9.7.6 Types of Noise

According to the source and pattern of noise production, several types of noise can be identified; each of them has its own characteristics in noise generation and transmission.

9.7.6.1 Structure-Borne Noise

Sound waves can travel through different media such as air, water, wood, rock, soil, or metal. Depending on the media through which it travels, sound is either structure-borne or airborne. Structure-borne noise propagates through the motor structure as vibration and subsequently radiated as sound. The intensity and frequency of structure-borne noise depend on many factors such as the rotational speed of the motor, material and geometry of the vibrating components, and system structure. The structure-borne noise is a result of the action of the aerodynamically, mechanically, and electromagnetically excited forces. When the frequency of the exciting force acting on the motor, as well as the response force on the motor support, approaches to or coincides with any of the motor natural frequencies, resonance occurs accompanied with strong vibration and high noise.

9.7.6.2 Airborne Noise

Airborne noise radiates from a noise source directly into and travels through the air. As the sound waves are being carried by the atmosphere, airborne noise can directly impact human hearing. These sound waves make contact with walls, floors, or ceilings, inducing noise pollution.

Airborne sound transmission loss (STL) is a measure of the degree to which a material can effectively block or reduce transmission of airborne sound. It is usually measured at each one-third octave band frequency from 125 to 4000 Hz, in the unit of dB. While structure-borne noise is attenuated by isolation, airborne noise can be attenuated by treatment with sound-absorbing materials or through the use of barrier materials.

9.7.6.3 Aerodynamic Noise

There are two types of aerodynamic noise: rotational and nonrotational. Rotational aerodynamic noise is associated with the rotation of the rotor and fan in a motor. When a rotor rotates, the wall rotating-driven flows, also referred to as the rotating pumping flow, are generated with high angular momentum to be flung radially outward, leading to flow-induced noise. The great amount of noise from a rotating fan is caused by steady thrust and drag forces that are induced on the blades as they rotate through air, and the air turbulence produced by the cooling fan can create a great amount of noise.

Nonrotational aerodynamic noise is mainly associated with the ventilating flow inside a motor, as a result of fluctuating fluid forces in time and frequency domains. When the motor uses a fan to force cooling air flowing through the motor, the ventilating flows pass through very complicated and irregularly shaped flow passages to cool stator windings and other components. The nonuniform velocity fields, high turbulent flows, the interactions between the rotational flows and ventilation flows, and the interaction between the flows and rigid structure all contribute to the nonrotational aerodynamic noise.

9.8 Noise Types and Measurements for Rotating Electric Machines

Growing environmental sound concerns and recognition that lengthy, unprotected exposure to high industrial noise levels can be detrimental to people have resulted in increased attention to reducing industrial noise. In many countries, particularly those in Europe, the

allowable maximum noise levels that workers should be exposed to are regulated by law, through government promulgation of noise level criteria.

Increasingly strict noise regulations in the United States, European Union, Japan, and China have made the development of new electric machines with low noise emission. Specifically, the European Union has set the tightest harmonized noise limits for various industrial machines, motor vehicles, household appliances, and other noise-generating products. Because of the increased awareness of the harmful effects from high industrial noise levels and government regulations that establish acceptable noise levels in the workplace, noise reduction has become an integral part of machinery safety. Noise abatement is a concern with, for example, generator-steam and gas turbine power plants. As demands for electricity increase, the power industry faces increasing challenges to build and operate efficient and quiet power generators, for example, steam turbines, gas turbines, and electrical generators.

In an industrial environment, various machines often emit noise waves that register at high and potentially harmful decibel levels. In a power plant, noise may come from a variety of machine sources, such as generators, gas or steam turbines, fans, pumps, coolers, and other mechanical and electrical equipment, many of which may be in operation simultaneously. Individuals working in such an environment are often faced with the need to reduce the near- and far field machinery noise levels. In an environment where individuals work in close proximity to the sources of a machine noise, near field sound levels must be controlled in order to comply with noise regulation and avoid hearing damage to the workers. Where machine noise can reach areas that are near an industrial plant, it may be prudent to abate far field machine noise to acceptable levels and to avoid broadcasting neighboring communities.

Regulatory requirements for low noise from electric motors have become increasingly more stringent. Today, environmental and safety agencies throughout the world are requiring even lower noise levels. With the increased demands for high efficiency, high power, and cost-effective electric machines, the motor industry is now facing challenges to build up not only more efficient but also quieter motors.

During motor operation, noise is produced from both rotating and stationary motor components, leading to environmental problems for people nearby. Thus, to protect people's health, motor noise must fall within acceptable limits, which have been strictly regulated in North American and European countries.

Various types of acoustic noise are produced during motor operation, as addressed in the following sections.

9.8.1 Noise Types in Rotating Electric Machines

Rotating electric machines are composed of various noise sources such as stator, rotor, ventilating fan, bearings, and housing, to name a few. The mechanisms of noise generation depend on the rotation of rotor, fan, and bearings, interaction of electromagnetic fields of the rotor and stator, machine structure, geometry of ventilation paths, and operation conditions. All noise generation mechanisms contribute to the overall noise pressure level of the machine. During normal operation, the noise sources create three main types of acoustic noise: mechanical noise, electromagnetic noise, and flow-induced noise.

9.8.1.1 Mechanical Noise

Shaft misalignment in motor causes preload forces to be generated in couplings that are then transmitted to various motor components. As one of the major causes of motor vibration, shaft misalignment produces a mechanical vibration with the frequency f_s :

$$f_s = 120n_s \quad (9.132)$$

where n_s is the rotating speed of the shaft in rpm.

Dynamically unbalanced rotor, bent shaft, eccentricity, rubbing parts, etc., produce a vibrating frequency of once per revolution or multiple of the number of revolution per cycle:

$$f_s = kn_m \quad k = 1, 2, 3, \dots \quad (9.133)$$

The magnitude of the vibration noise due to the unbalanced rotating masses depends on the degree of balancing of the rotating masses and the rotating speed.

Loose stator lamination stack results in the vibration frequency with the frequency sidebands of 1000 Hz:

$$f_{lam} = 2f \quad (9.134)$$

where f is the line frequency.

Rolling bearings are extensively used in modern electric motors. Rolling bearing generates mechanical impulses when the rolling element passes the defective groove, causing a small radial movement of the rotor. Rolling bearings produce noise with discrete frequencies corresponding to the rotating frequency of each rolling element, such as roller, cage, and inner/outer ring. The bearing noise levels depend on the speed of rotation and their manufacturing perfections. Some large-sized motors also use sleeve bearings. The noise due to sleeve bearing is generally lower than that of rolling bearings. The vibration produced by sleeve bearings depends on the roughness of sliding surfaces, lubrication, and stability and whirling of the oil film in the bearing manufacturing process and bearing assembly [9.54].

Sound power radiated from an electric machine is proportional to the vibration area S and the mean vibration velocity square $\bar{u}^2$ [9.55]:

$$W = \rho c S \bar{u}^2 \sigma_{rad} \quad (9.135)$$

where

ρ is the air density

c is the speed of sound

σ_{rad} is the radiation efficiency

This equation indicates that mechanical noise can be reduced by reducing the vibration area and/or the vibration velocity. Reducing the vibration area can be achieved by dividing a large area into a number of small areas with flexible joints between them. Reducing the vibration velocity can be achieved by using damping and vibration-absorbing materials to isolate the vibration source from other components. A reduction of the exciting forces by a factor of two can possibly lower the SPL up to 6 dB.

9.8.1.2 Electromagnetic Noise

Electromagnetic field is not uniform in the air gap and consists of various harmonics. The periodic fluctuations of the magnetic flux in the air gap cause oscillating forces on both the rotor and stator. The noise due to the fluctuating magnetic forces is called nonrotational

noise, which resulted from the asymmetry of the electric and magnetic circuit and saturation of the magnetic field.

There are many normal modes of vibration in the stator core in all directions. While the radial forces cause radial extension of the stator core and consequently vibration and structure-borne noise, the axial forces and tangential forces result in vibration of the stator core in the axial and circumferential directions.

The magnitude of rotational electromagnetic noise strongly depends on the motor rotating speed. Hence, the electromagnetic noise is characteristically tonal in character with a great number of discrete frequency tones, the most important of which are at the frequency [9.56]

$$f_{em} = N_s n \quad (9.136)$$

where

N_s is the number of stator slots

n is the motor rotating speed

Electromagnetic noise is stimulated when the electromagnetic force matches or is close to the natural frequencies of vibration of motors.

9.8.1.3 Aerodynamic Noise

Aerodynamic noise, also referred to as windage noise or flow-induced noise in some references, is generated by the interactions of turbulent ventilating air flow with rotating (e.g., rotor, fan) and stationary components (e.g., stator, endbell) inside a motor. As airborne noise, this type of noise increases with the increasing flow speed.

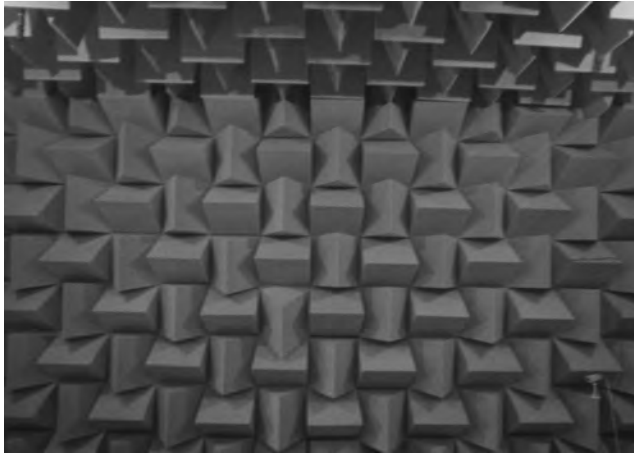
As discussed previously, the aerodynamic sound power generated by turbulent flows is proportional to the 6th to 8th of the flow velocity (i.e., $W \propto u^{6-8}$). Consequently, the increase in the flow velocity can result in the great increase in sound power. Thus, a logical approach to lower the aerodynamic sound power is to reduce the flow velocity and turbulence intensity. This requires that in the design of motor ventilation system, it should avoid the sudden changes in the cross-sectional areas along the flow paths and reduce the right-angle bends as much as possible.

A ventilation fan is usually a dominant noise source in a motor. The fan generates broadband aerodynamic noise as the cooling flow passes through the inlet and outlet of the fan. In practice, fan noise reduction can be achieved by the use of an absorptive silencer or by redesigning the cooling fan [9.56]. The sound power level generated by fans can be predicted in the early design stage using the Graham equation [9.57].

9.8.2 Acoustic Anechoic Chamber

Noise measurement is a necessary step to determine the SPL of an electric machine is below the allowable level. With today's technology, many types of measuring systems can be used depending on the characteristics of sound, the extent of information that is desired about the sound, and cost, from the most popular sound level meters to the professional sound-measuring systems in acoustic laboratories.

An acoustic anechoic chamber is the specially designed room that can completely absorb reflections of sound waves and insulate testing equipment from environmental sources of noise. To absorb sound waves, the interior surfaces of the acoustic anechoic chamber are covered with acoustically absorbent materials. One of the most effective types of absorbent

**FIGURE 9.34**

An anechoic chamber is specially designed to completely absorb reflections of sound waves.

materials comprises arrays of pyramid-shaped pieces, each of which is constructed from a suitably lossy material. In order to get the best results, all internal surfaces of the acoustic anechoic chamber must be entirely covered with the pyramids (Figure 9.34).

The relationship of the sound wavelength λ , phase velocity of wave v , and frequency f is given as

$$\lambda = \frac{v}{f} \quad (9.137)$$

To shield for specific range of wavelengths, the shape and size of the pyramids must be carefully designed for absorbing the range of wavelengths.

9.8.3 Measurement of Motor Noise

Typically, the anechoic chamber is used to measure the directivity of noise radiation from testing machinery or equipment and to determine the transfer function of a loudspeaker.

There are a number of elements in a sound-measuring system, including the following:

- Microphones that respond to sound pressure of a testing motor at specific locations and transform it into an electric signal. The commonly used types of microphones include piezoelectric, condenser, electret, or dynamic microphones.
- Signal amplifier boosts the amplitude of the microphone output signal to a useful level for further processing.
- Active filter that reduces high-frequency signal components, unwanted electrical interference noise, or electronic noise from the signal.
- The data storage, processing, and transportation.

Noise is measured at the rated operation conditions in the anechoic chamber by using microphones at a distance of one meter from the motor in different orientations, as shown in Figure 9.35. The noise data are taken as the test motor reaches the thermal equilibrium.

**FIGURE 9.35**

Measurement of acoustic noise level of electric motor by microphones in an anechoic chamber.

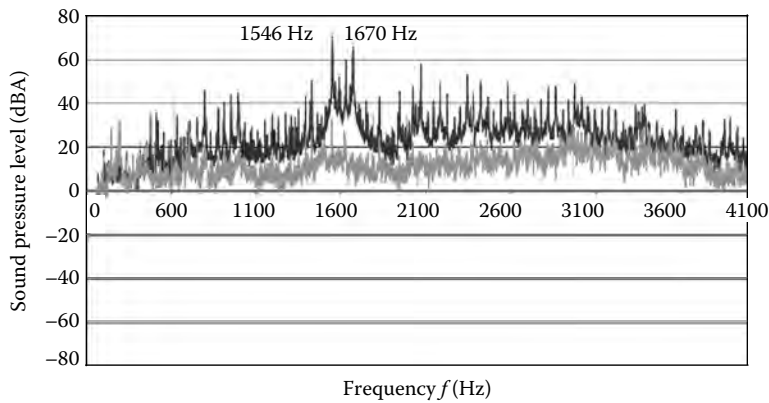
The spectrums of SPL from two PM servomotors are displayed in Figure 9.36. These two motors are identical in terms of brand and motor, and the testing has been conducted under the same conditions. However, their SPLs show distinct differences for almost all frequencies, especially in the range of 1000–3000 Hz.

9.8.4 Acoustic Noise Field Measurement

A field noise test was conducted in customer site at Texas. Acoustic noise data were acquired from a large-sized generator, which is installed in the open air at a height of 15 m above the ground. There are totally 15 measuring points, 5 on either side of a large-sized generator. These points were labeled from 1 to 5 going from the collector end (CE) to the turbine end (TE) of the generator, as well as along the CE (Figure 9.37). The data were acquired using a Rion®, handheld precision acoustic sound meter, interfaced with a Hewlett Packard® four-channel analyzer. The acoustic data were taken at three load levels of 17, 86, and 156 MW (156 MW is the full load).

9.9 Motor Noise Abatement Techniques

Controlling the noise level of electric motor is critical to the overall system performance. Therefore, noise abatement of motor is an important goal in motor design. The most cost-effective way to deal with the noise issue is to incorporate the reduction of noise level in an early design stage. Once a motor is manufactured, it is often expensive to modify the

**FIGURE 9.36**

Comparison of SPLs for two PM servomotors with same brand and same model. Courtesy of Kollmorgen Corporation.

**FIGURE 9.37**

Active noise-cancelling headphones.

design for reducing noise levels. For old motors with high noise levels, an effective method to lower the motor noise exposure is to make external acoustic treatments of the motor.

A number of techniques to minimize noise level can be adopted in motor design. These techniques can be separated into two categories, passive and active.

9.9.1 Active Noise Reduction Techniques

Effective reduction in acoustic noise can be accomplished using active noise reduction techniques. One of the techniques is called noise cancellation technique. The concept of

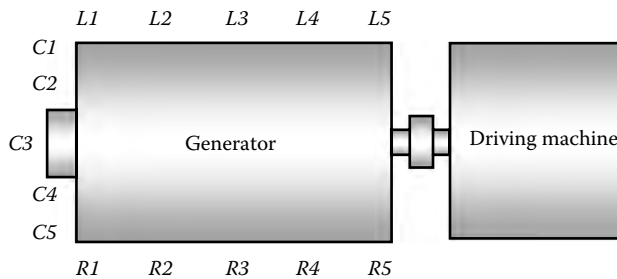


FIGURE 9.38
Measuring points around a large-sized generator.

active noise cancellation was initiated and developed in the early 1930s. Paul Lueg [9.58], a German engineer, was the first who realized the possibility of attenuating background noise by superimposing a phase flipped wave. Later, this concept was successfully confirmed by Olsen [9.59,9.60] to generate artificial sound waves that have the opposite phase to that of noise. This method is particularly useful at low frequencies because conventionally used sound absorbers do not work well at such frequencies, for which the sound wavelength is much larger than the thickness of a typical absorber.

Today, active noise reduction techniques have been successfully adopted in many commercial applications such as active mufflers, passage cars, wind tunnel testing systems, noise-cancelling headphones, and noise control in air-conditioning ducts. As a typical example, noise-cancelling headphones have been invented to eliminate any low-frequency noise from the environment. A variety of antinoise headphones is shown in Figure 9.38. These headphones involve using one or more microphones and an electronic circuitry that uses the microphone signals to generate antinoise signals, thus the generated destructive interference cancels out the ambient noise.

With today's advanced digital signal-processing techniques, it is possible to make active noise control more precisely and more efficiently. However, active damping techniques are considerably expensive because they require measurement of response and feedback control, and thus have not yet been applied to electric motors. In addition, the application of active noise cancellation on large-size electric machines is less effective due to the challenge in separating multidirectional sounds.

9.9.2 Passive Noise Reduction Techniques

Passive techniques are the simplest and least expensive means of preventing problems associated with acoustic noise. There are several conventional methods available to reduce motor noise, each has its own advantages and disadvantages for specific applications. The selection of an appropriate method depends on its effectiveness in noise attenuation and system cost.

9.9.2.1 Blocking Noise Propagation Paths and Isolating Noise Sources

Mechanical noise transmissions can be greatly reduced by blocking noise propagation paths [9.61] or isolating sources of noise. An effective and traditional way is to locate noisy machines behind purpose-built barriers. In order to achieve the maximum shielding effect, the barriers can be constructed with sound-absorbing and damping materials. For large-sized motors, noise barriers can be erected between motors and noise-sensitive

areas. Compact motors can be covered by a box with noise-absorbing and damping materials attached on its internal surfaces.

Another approach is to use mechanical dampers, such as rubber boots, reflecting enclosures, to lower the level of vibration and consequently the level of transmitted noise, as discussed in the previous sections.

9.9.2.2 Using Noise-Absorbing Materials

A common way of noise reduction is to attach sound-absorbing materials on either internal or external walls of the motor housing/endbells for absorbing sound energy. Sound-absorbing materials may come in many forms, ranging from simple foam and cellular materials to a variety of commercially made products designed specifically for the purpose. Most sound-absorbing materials contain a large number of small cavities. When sound waves pass through the air inside the cavities, certain frequencies of sound waves may result in resonance of the air inside the cavities. In such a way, sound energy is consumed and the sound pressure is reduced. The resonances depend on the size and shape of the cavities. Therefore, well-designed cavities in sound-absorbing materials can effectively reduce motor acoustic noise.

Removable and reusable blankets have been widely used for acoustical reduction applications. The benefits of using blankets include the following:

- One-time investment
- Long-term repeated use
- Without interrupting equipment normal operation
- Making equipment maintenance easier
- Providing protection for personnel

Acoustic blankets may contain single- or multifiberglass layers depending on application requirements. As an example shown in Figure 9.39, an acoustic blanket consists of two high-density layers and one low-density layer. Usually, high-density blankets are used to contact with the generator wall. Fiberglass fibers are encased within PTEE-coated (or silicone rubber-coated) fabric envelops, which are designed to withstand both chemical attack and severe mechanical abuse. Stainless steel hook-and-loop fasteners are used for jointing in blanket sections. Acoustic blankets are wrapped around the external surface of an electric machine for reducing noise levels.

In order to fit better the generator's external shape, blankets are made in a number of sections. Each section has a stainless steel identification tag that has a unique identification number corresponding to the installation drawings. Stainless steel jacketing, with a nominal thickness of 0.7 mm, is used to cover the blanket system. The jacketing normally utilizes long-reach stainless steel buckles and strikers. As with the blankets, each jacketing section has a stainless steel identification number corresponding to the installation drawings. As an alternate to separate stainless steel jacketing, integral stainless steel mesh is used. The mesh is factory-sewn to the fiberglass blanket assembly and acts to protect the blanket insulation from walk traffic and/or other abrasion damage.

A generator may be wrapped partially or completely with acoustic blankets. With an application of the generator laggings, acoustic noise at the generator bottom is blocked; only the upper half of the generator needs to be covered (Figure 9.40). The installation of the acoustic blanket on a large-sized generator is shown in Figure 9.41.

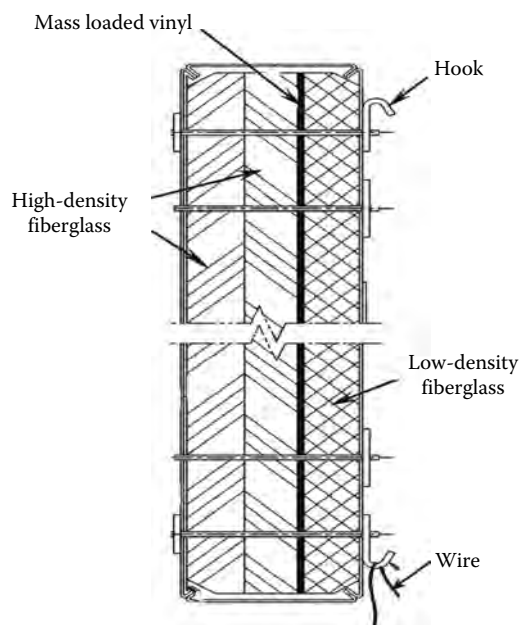


FIGURE 9.39
Acoustic blanket consisting of one low-density and two high-density fiberglass layers with a mass-loaded vinyl sandwiched between them.

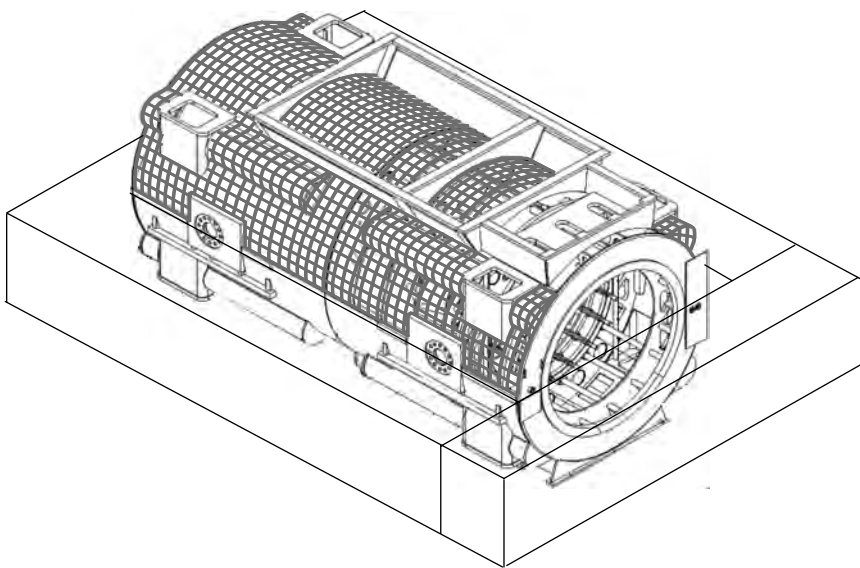


FIGURE 9.40
Covered areas by acoustic blankets on a generator.



(a)



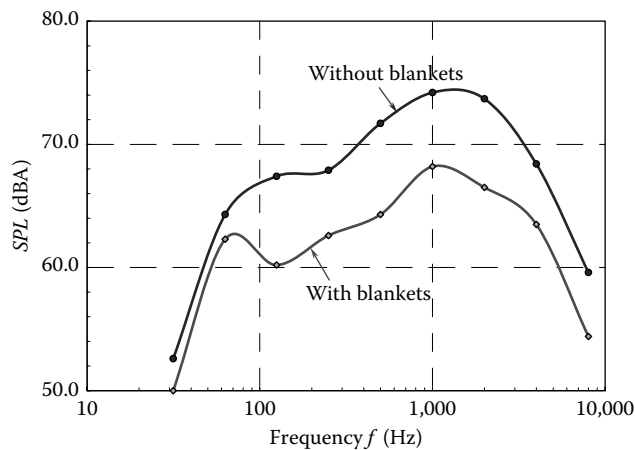
(b)

FIGURE 9.41

A large-sized generator with bared outer surfaces (a) and installed acoustic blankets for noise reduction (b).

The test data have shown that acoustic blankets have a significant impact on generator noise levels. At the generator, sides were covered with blankets; the reduction in noise levels ranges from 2 (low frequencies) to 8 dBA (Figure 9.42). It is reported that the properly used acoustic blankets on electric machines can attenuate noise pressure level by 5–10 dBA [9.62].

The spectra data at the tested generator are presented in Figure 9.43. It can be observed that blankets can reduce noise level at all frequencies. For instance, at the center of the

**FIGURE 9.42**

Effect of acoustic blanket on noise reduction for different frequencies.

generator left side, the noise level reduces from 70.8 to 66.5 dBA at $f=60$ Hz and 72.8 to 60.5 dBA at $f=120$ Hz.

One of most popular methods is to use some special materials to absorb sound energy. These materials such as foams or cellular materials comprise many small cavities. The drawback with these traditional noise reduction approaches is that they only work with some frequencies.

However, while the method is effective in noise reduction in some cases, it can impact the effectiveness of the heat dissipation from the housing to the ambient. In addition, this approach works for rather limited frequencies.

9.9.2.3 Motor Suspension Mounting

On undamped baseplate mountings, motor noise can be transmitted, amplified, and radiated by nonmotor structures. A motor suspension on rubber mounting or cushioned mounting can be added to the installation to reduce noise and vibration.

9.9.2.4 Noise-Attenuating Structure

Ramu [9.63] proposed a method of noise reduction for switched reluctance machines (SRMs). SRMs usually produce higher noise than other types of electric machines, making them less attractive for some commercial and industrial applications. For instance, medical devices often require quiet operation with less noise. The SRM noise is attributed to (1) the normal forces caused by various imbalances in the nonuniform air gap, (2) the discontinuous currents in the windings causing discontinuous torque that produces very high torque pulsations, and (3) the rotor functioning like an impeller to create turbulent flows in the machine, causing significant windage noise.

SRM motors have no windings or PMs on rotors. As a rotor rotates, its salient poles are like fan blades that generate turbulent air flow in the motor. The fan blade noise is

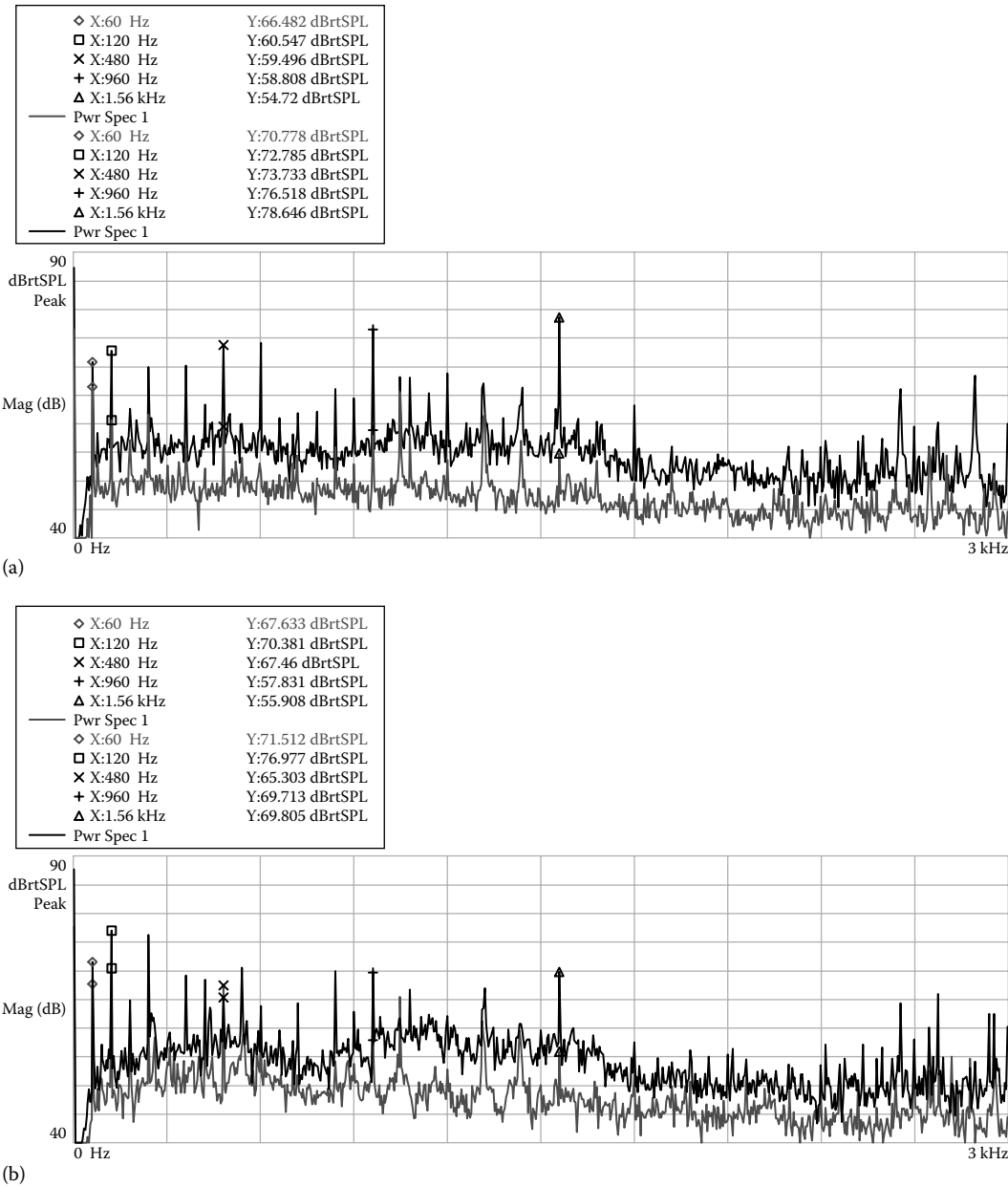


FIGURE 9.43 Spectra data at the center of generator sides: (a) left side and (b) right side; viewing from the CE. In each case, the curve with higher noise pressure levels is for uncovered acoustic blankets.

produced due to the alignment effect between the rotor and stator poles. The frequency f_{rt} of a pure tone arising from air passing through the rotor slots can be calculated by

$$f_{rt} = p_r \frac{n}{60} \quad (9.138)$$

where

p_r is the number of rotor slots

n is the rotor rotating speed (in rpm)

This tonal frequency noise has exactly the same value as the combined phase frequency of the machine.

SRM noise can be remarkably reduced by several methods, including the following:

- Encapsulating a machine's stator/rotor slots can reduce windage noise. As shown in Figure 9.44, the rotor slots are filled by epoxy to flush with the rotor outer surface and lamination end surfaces. The smooth rotor surface has much less friction than the surface with slots in which air forms recirculation flows to cause windage power loss and windage noise.
- The phase rotation of end lamination with respect to the intermediate laminations is another way to lower machine noise. It can be seen in Figure 9.45 that two end laminations are rotated oppositely about the rotational axis of the rotor to partially or fully cover the rotor slots. This creates a barrier inhibiting the air flow into and out of the slots.
- Similar to the previous discussion, the rotor/stator slots can be blocked using two end annuluses, as shown in Figure 9.46.

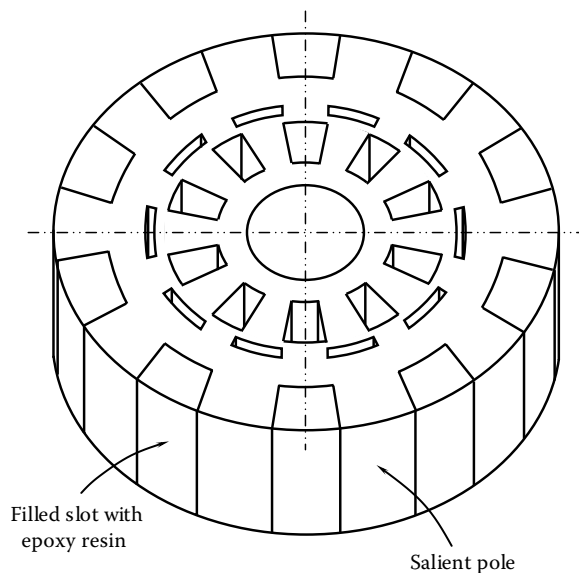
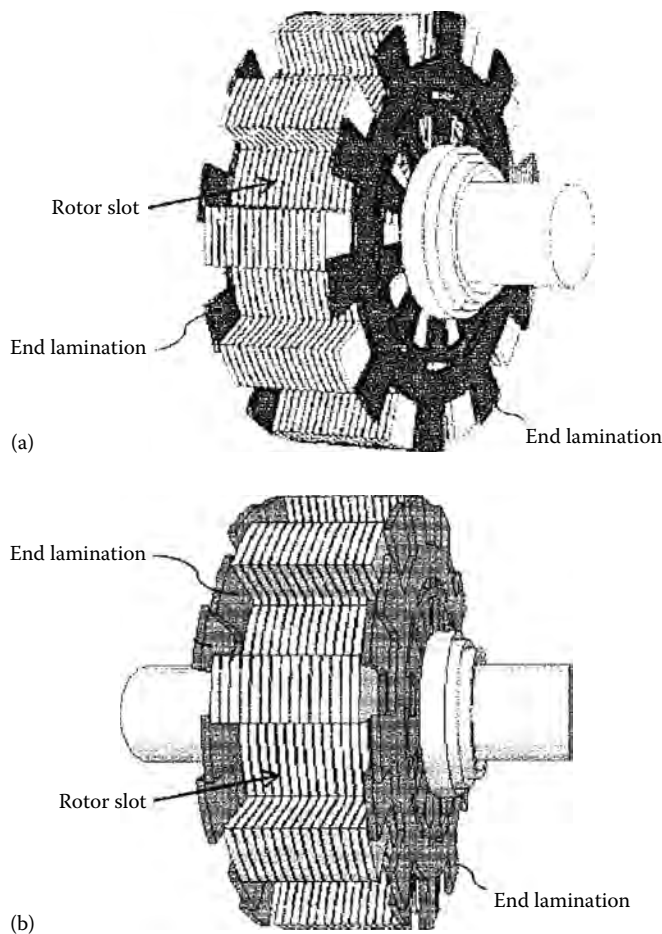


FIGURE 9.44

Rotor slots are filled with epoxy resin to reduce windage noise [9.63]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

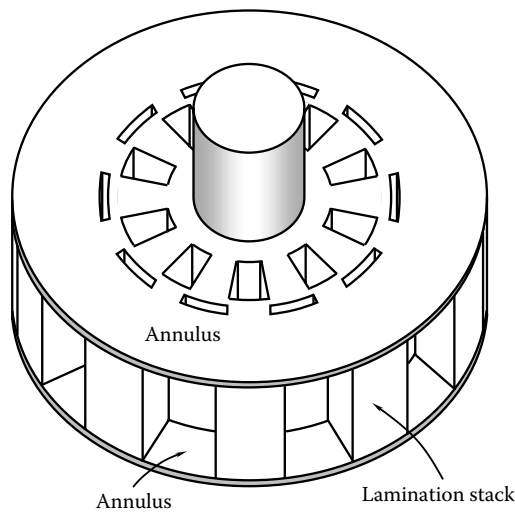
**FIGURE 9.45**

Phase-shifted end laminations for reducing noise: (a) shifting two end laminations in opposite directions with respect to intermediary laminations and (b) using multiple phase-shifted laminations for covering longitudinal sides of slots [9.63]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

9.9.2.5 Smoothing Ventilation Paths

When a motor rotor runs at a high speed, it stimulates air inside the motor to form swirl flow in the air gap and at the ends. Fans attached to the rotor also generate ventilation flows primarily in the axial direction to cool motor components. As a result, the velocity field in the motor is considerably complicated. Vortex motions are associated with regions of flow discontinuity that occur at interfaces between fluids and solids in relative motion or between parallel-moving flows of different velocities. The vorticity in turbulent flows is responsible for regions of relatively intense flow mixing and activity. Briefly, noise is produced whenever vortex lines are stretched or accelerated.

Smoothing ventilation paths can reduce the flow turbulence intensity and in turn the flow-induced noise, which usually arises in applications involving turbulent flows.

**FIGURE 9.46**

Adding an annulus at each end of rotor to block axial air flow paths and thus reduce acoustic noise [9.63]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

9.9.2.6 Selecting Low Noise Bearings

During normal operation, rolling bearings make a variety of vibration and acoustic noise. As discussed in Chapter 6, the bearing noise and vibration can be briefly grouped into four categories [9.64]—structural, manufacturing, handling, and others:

- Structural noises resulted from free vibration of the bearing components, including raceway rings, rollers, seals, and cage.
- Ideally, bearing raceways are manufactured with perfect roundness and high-quality surface finish. However, because any machine has its limited machining precision, waviness usually presents on the manufactured raceway surfaces, causing waviness noise and vibration and impacting bearing frictional performance and endurance.
- Handling noise consists of flaw noise and contamination noise. Manufacturing flaws can lead to abnormal operation of bearing to produce noise. Contamination can dramatically shorten bearing life. According to failure analysis statistics, about 15% of all motor bearing failures is attributed to contamination [9.65]. Since the lubricant film thickness, which separates the mating surfaces in the rolling contact, is very thin (usually less than 1 μm), any particles in size larger than the film thickness can disturb the bearing smooth running.
- The last category of noise/vibration includes lubricant noise and seal noise. It is long recognized that oil-lubricated bearings have longer lifetime than grease-lubricated bearings because greases always contain solid particles. There are a few greases in the market that fall into the *noisy* class due to their thickener or solid additives. Typical examples are some calcium-complex greases that contain large particles of calcium salts, producing small permanent dents of raceway surfaces [9.66]. Inadequate lubrication of bearings may not only create higher motor noise but also cause higher bearing temperatures.

Most motor manufacturers randomly sample bearings for inspections. However, considering the fact that most motor failures can be attributed to premature bearing failures, it is highly recommended to conduct 100% bearing inspections, especially for some special applications under severe environmental conditions (e.g., deep sea, outer space, or desert).

Among all types of bearings, sleeve bearings make lowest noise and ball bearings generally make more noise than roller bearings [9.67].

9.9.3 Innovative Noise Abatement Methods

To address noise abatement problems, acoustic engineers presently turn to innovative methods that make new approaches to the physics of noise reduction. With the fast development of nanotechnologies in recent years, motor noise can be significantly reduced by applying nanofibers and by coating the housing and endbell internal surfaces with micro-silicone paint (e.g., *Lotusan*).

The advantages of using nanotechnology in motor noise reduction include the following:

- High efficiency in noise reduction
- Taking a little space
- Low-density structure
- Reliable
- Easy to fabricate

Engineers in Georgia Tech Research Institute developed a porous metal structure that contained many cylindrical channels for attenuating aviation turbine noise [9.68]. The diameter of each channel ranges approximately from 0.1 to 0.3 mm. Later, this structure further evolved into a microscale honeycomb structure, which is composed of a large number of metallic nanotubes. This technique, unlike traditional techniques that absorb sound using a more frequency-dependent resonance, dissipates acoustic waves by wearing them down through a process called viscous shear. The reduction of sound pressure level in this approach is independent of frequencies or resonance.

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10

Motor Testing

Safe, reliable, steady, and efficient operation of electric motors is essential for all motor applications. For example, elevator motors must have very high operating reliability and factors of safety because they are directly related to people's lives. Motor failure in a nuclear plant or in a launched rocket may cause catastrophic results, as well as tremendous economic losses and long-lasting impacts.

New motor design and development programs often incorporate prototype testing activities. Motor testing is a crucial step to ensure motor performance, efficiency, and manufacturing integrity. It can detect any potential faults, improve operation reliability, and optimize the motor design. For these purposes, a variety of tests must be conducted under different operating conditions. Testing variables may include the load (torque) level, rotating speed, voltage, current, induction, resistance, temperature distribution, and cooling flowrate.

10.1 Motor Testing Standards

There are a large number of IEEE, IEC, ANSI/NEMA, and EASA standards dealing with electric motor testing. Some of them are listed as follows:

- IEEE Standard 43-2000. Recommended practice for testing insulation resistance of rotating machines [10.1]. This document describes the recommended procedure for measuring insulation resistance of armature and field windings in rotating machines. It has addressed the general theories of insulation resistance and polarization index (PI).
- IEEE Standard 56-1977. Guide for insulation maintenance of large-current rotating machinery [10.2]. This standard provides guidelines for testing and inspection of insulation systems on motors.
- IEEE Standard 118-1978. Test code for resistance measurement [10.3]. The purpose of this code is to present methods of measuring electrical resistance that are commonly used to determine the characteristics of electric machinery and equipment.
- IEEE Standard 95-2002. Recommended practice for insulation testing of AC electric machinery (2,300 V and above) with high direct voltage [10.4]. This document provides information on the use of high direct voltage for proof tests and for periodic diagnostic tests on the ground wall insulation of stator windings in AC electric machines.
- IEEE Standard 522-2004. Guide for testing turn insulation of form-wound stator coils for alternating-current electric machines [10.5].

- IEEE 1068-1990. Recommended practice for the repair and rewinding of motors for the petroleum and chemical industry [10.6].
- ANSI/IEEE Standard 112-2004. Standard test procedure for polyphase induction motors and generators [10.7].
- ANSI/NETA MTS-2007. Standard for maintenance testing specifications for electrical distribution equipment and systems [10.8].
- ANSI/NETA ATS-2009. Acceptance testing specifications for electrical power distribution equipment and systems [10.9].
- ANSI/NETA ETT-2010. Standard for certification of electrical testing technicians [10.10].
- NEMA MG1-1993. Motors and generators [10.11]. It provides practical information concerning performance, safety, test, construction, and manufacture of AC and DC motors and generators.
- IEC 60034-1: 2010. Rotating electrical machines—Part 1: Rating and performance [10.12].
- IEC 60034-2-1: 2007. Rotating electrical machines—Part 2-1: Standard methods for determining losses and efficiency from test (excluding machines for traction vehicles) [10.13].
- IEC 60034-4: 2008. Rotating electrical machines—Part 4: Method for determining synchronous machine quantities from tests [10.14].
- IEC 60034-19: 1995. Rotating electrical machines—Part 19: Specific test methods for DC machines on conventional and rectifier-fed supplies [10.15].
- IEC 60034-29: 2008. Rotating electrical machines—Part 29: Equivalent loading and superposition techniques—indirect testing to determine temperature rise [10.16].

Because there are so many standards presently, motor manufacturers in different countries often follow different standards in their production. For instance, motor efficiency testing protocols differ from the world, and their applications on any given motor can lead to significantly different efficiency values. Therefore, it is important to unify the standards all over the world.

10.2 Testing Equipment and Measuring Instruments

There are a large variety of instruments and equipment used in motor test laboratories. The understanding of the functions and fundamental principles of these instruments and equipment, as well as correctly using them, is essential for the success of motor testing. In addition, all of the instruments must be carefully calibrated for obtaining accurate and reliable testing results. A brief introduction of the main instruments and equipment is given in the following.

10.2.1 Dynamometer

As one of the key devices in electric motor testing, dynamometers (or dynos in short) have been used extensively for measuring rotating speed, torque, power output, and force from

power sources such as motors and engines. Based on the measured operation characteristic data, motor efficiency and other useful information can be determined. Dynamometers can operate in two basic modes: (1) the absorbing or passive mode in which a dynamometer is driven by motor under test and provides a specific brake torque load to the testing unit and (2) the driving mode in which a dynamometer drives a machine to determine the torque and power required for operating such a driven machine. A dynamometer that can either drive or absorb is called a universal or active dynamometer.

Motor testing dynamometers apply braking or drag resistance to motor rotation under various operation conditions. Consequently, the absorbing dynamometer is used to absorb the power developed by the motor in motor testing. The brake power P_b , which is generated by a dynamometer and applied to the tested motor, is calculated by multiplying the torque T_{dyno} (N-m) and rotating speed ω_{dyno} (rad/s), that is,

$$P_b = T_{dyno} \omega_{dyno} \quad (10.1)$$

A variety of different dynamometers, either for absorption or drive mode or both, have been used for electric motor testing, including

- Powder brake dynamometer (absorption only)

As its name implies, a powder brake dynamometer contains fine magnetic powder in the air gap between the rotor and stator of the brake. When current is applied to the excitation coil, a magnetic field is generated in the brake. Under the magnetic field, the magnetic powder is arranged in *magnetic chains* in accordance with magnetic flux lines, altering the powder from the free moving condition to the solid rock condition. As the brake rotates, the magnetic chains are constantly built and broken to create a great loading torque to the rotor. Thus, the loading torque can be easily adjusted by altering the excitation current. This type of dynamometer is ideal for high torque, low speed applications. It can provide full torque at zero rotating speed. Powder brake dynamometers are normally cooled with water and is not suitable for vertical operation.

- Hysteresis brake dynamometer (absorption only)

A hysteresis brake dynamometer produces braking torque through a magnetic air gap without using magnetic particles or friction components. The hysteresis brake dynamometer usually consists of a cup-shaped rotor, a gear-shaped stator, and an excitation coil, providing precise torque loading that is independent of the shaft speed. The rotor is made of magnetic material with hysteresis characteristics. When the stator is energized, the cup-shaped rotor can spin freely. As current passes through the excitation coil, a magnetizing force is applied to the stator and the rotor is magnetically restrained, providing a braking action between the rotor and stator. This method of braking provides far superior operating characteristics such as smooth torque loading, long lifetime, high repeatability, good controllability, and less maintenance and downtime, making them the preferred choice for precise torque control in electric motor testing. Because the energy absorption through hysteresis loss is proportional to the rotating speed of the rotor, the break torque generated in the hysteresis brake dynamometer does not depend on speed. In fact, the dynamometer functions as a stable brake in the full motor ramp from the stopped state through the full rotating speed. Cooling of this type of dynamometer usually relies on either natural convection or forced convection (using

**FIGURE 10.1**

A motor is being tested by using hysteresis brake dynamometer.

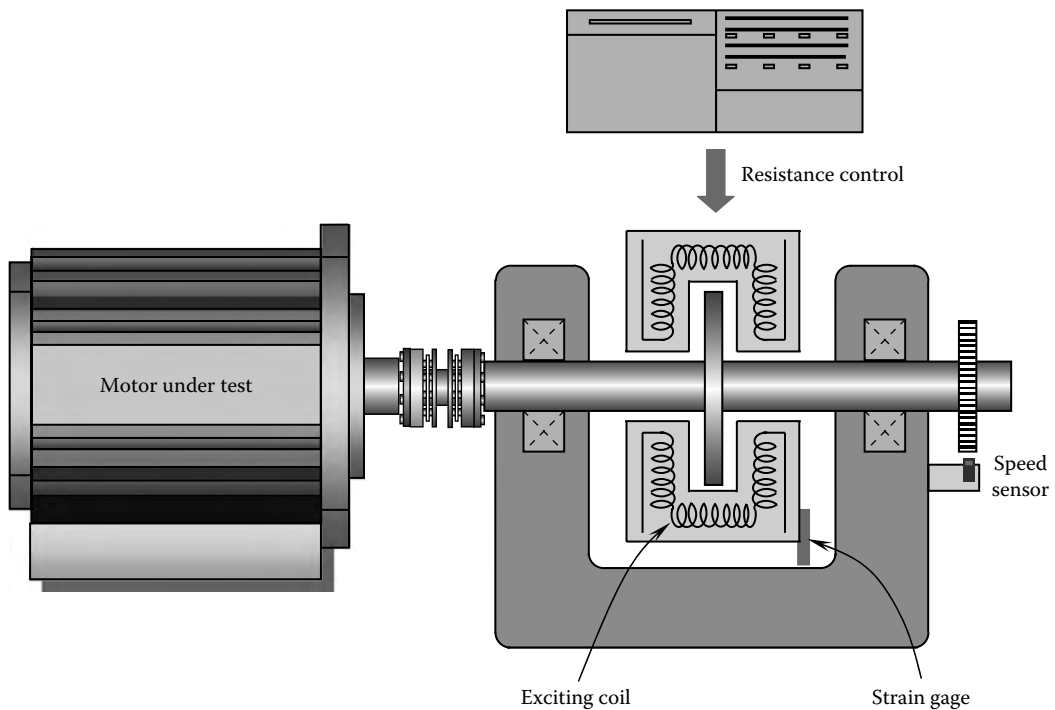
fans or compressed air). Figure 10.1 shows a motor being tested using a hysteresis brake dynamometer.

- Eddy-current brake (absorption only)

Eddy-current brake dynamometers produce braking torques using the principle of eddy currents induced in rotating metallic disks, which are immersed in magnetic fields. As demonstrated in Figure 10.2, when electrical current passes through coils around the disk, it generates a magnetic resistance to the rotation of the disk. The resistance load is controlled by changing the current. They are ideal for applications requiring high speeds and operating in the middle-to-high-power range. This type of dynamometer is capable of changing the load effectively at high speeds. The advantages of this type of dynamometer include low maintenance cost, precise control, simple construction, and desirable speed–torque characteristics.

- Electric motor/generator dynamometer (absorption or drive)

Equipped with appropriate control units, electric motor/generator dynamometers can operate as either absorption or universal dynamometers, depending on applications. A motor/generator dynamometer typically uses an electric motor (either AC or DC) with its drive to simulate mechanical loads on the motor being tested. In a test, the motor under test is attached to the loading motor which acts as a generator. The main advantage of electric motor/generator dynamometers is that

**FIGURE 10.2**

Eddy-current dynamometer used for motor testing, relying on eddy current generated in the rotating disk to provide resistant load to the tested motor.

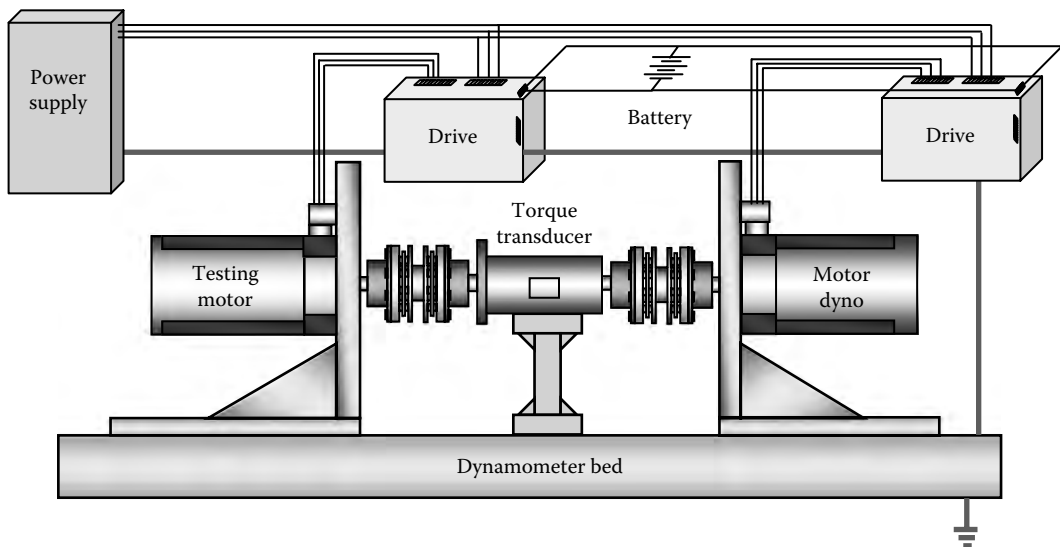
they can adjust load in a large range within a very short time. However, electric motor/generator dynamometers are generally more expensive and complex than other types of dynamometers. A testing system is shown in Figure 10.3.

- Mechanical friction brake dynamometer (absorption only)

Mechanical friction brake (Prony brake) dynamometers were invented by Gaspard de Prony in 1821. Using friction to load testing motors, this type of dynamometer is considered as one of the most simple, standard, and earliest absorbing dynamometers. Their advantages include ease of design, cost-effectiveness, simple maintenance, compactness, and high power absorption. However, due to the problems of brake torque control, as well as the rapid development in modern electronic control technologies, mechanical friction brake dynamometers are rarely used in motor labs today.

- Inertia brake dynamometer (absorption only)

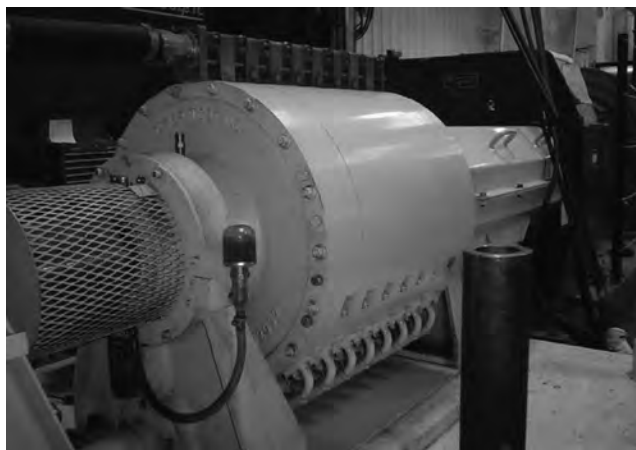
All brake types of dynamometers measure power from the torque reaction of the brake at a specific speed. An inertia brake dynamometer utilizes the test motor to accelerate a flywheel from a standing start-up through the motor speed range. This method involves measuring the change in the flywheel speed as the motor accelerates. The kinetic energy stored in the rotating flywheel is proportional to the square of the rotating speed. Thus, the flywheel has a significant amount of inertia to resist any change in speed. Today, inertia dynamometers become the standard test method for electric motors.

**FIGURE 10.3**

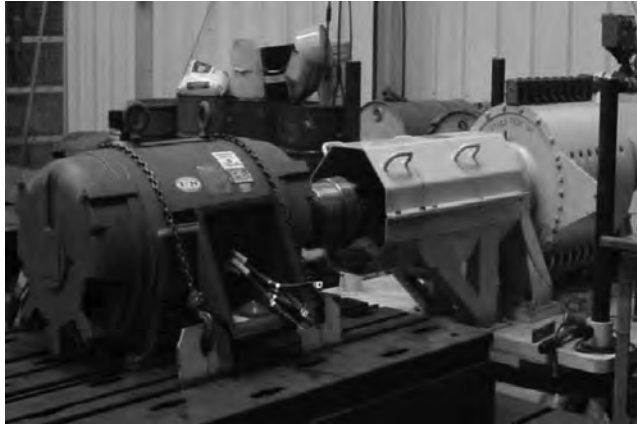
Motor testing system with the motor/generator dynamometer.

- Water brake dynamometer (absorption only)

Water brake dynamometers utilize water flowing through absorbers to create loads on testing motors (Figure 10.4). This type of dynamometer is the most prevalent choice for applications that require high rotating speed, low inertia, high-power capacity and low cost. The loading on the motor being tested is dependent on the water flowrate through the brake. For their high-power capability, water brake dynamometers are commonly used in high-power, large-size motor testing. Unlike other types of dynamometer, water brake dynamometers eliminate the need for cooling and thermal overload protection systems and thus provide the most cost-effective solution for motor testing. However, using of this type of

**FIGURE 10.4**

Water brake dynamometer for motor testing.

**FIGURE 10.5**

The motor being tested is driven by a water brake dynamometer.

dynamometer may take relatively long periods of time to reach the stabilized loading condition.

The testing system layout using a water brake dynamometer is shown in Figure 10.5. The motor is fixed on the test platform via steel chains and is connected with a torque transducer, which in turn is connected with the water brake dynamometer through a coupling.

- Hydraulic brake dynamometer (absorption only)

Equipped with a hydraulic pump, a fluid reservoir, and a piping system, a hydraulic brake dynamometer provides braking drag against the motor being tested. This type of dynamometer can be used in a wide range of power. It is suitable to test extra-large motors with megawatts. Like water brake dynamometers, this type of dynamometer has no need for cooling systems.

- Chassis dynamometer (both absorption and drive)

Chassis dynamometers are capable of very accurately measuring the speed, torque, and power. This type of dynamometer is widely used in the auto industry for performing engine tests. The typical loading modes used with a chassis dynamometer are constant force, constant speed, or a vehicle-simulation value. With the measured dynamometer shaft torque T_{dyno} , rotating speed ω_{dyno} , and the engine rotating speed ω_{eng} , the engine torque T_{eng} can be determined as [10.17]

$$T_{eng} = \frac{T_{dyno} \omega_{dyno}}{\omega_{eng}} \quad (10.2)$$

10.2.2 Thermocouples and Other Temperature Measuring Devices

Thermocouples are used to measure temperatures at different locations of the testing motor, as well as the ambient temperature. A thermal couple consists of a pair of wires, made of dissimilar thermoelectric materials. Because the two materials have different Seebeck coefficients, when the two wires are joined together at both ends, an electric potential is generated across the wire junctions. This electric potential is proportional

TABLE 10.1

Seebeck Coefficients for Various Thermocouple Types at 25°C

Thermocouple Type	Seebeck Coefficient ($\mu\text{V}/^\circ\text{C}$)
E	61
J	52
K	41
N	27
R	9
S	6
T	41

to the temperature gradient across the wire junctions. Thus, an unknown temperature can be measured by placing one end of the wire junction (measurement junction) at the desired location and connecting the other end of the wire junction (reference junction) to a multimeter. The Seebeck coefficients for various thermocouple types are given in Table 10.1 [10.18].

For high-performance devices, the materials are typically *p*- and *n*-type semiconductors having positive and negative Seebeck coefficients, respectively. One end of the thermocouple analyzed here is connected by a metal tab. Metal tabs are also found at the other end of the thermocouple for external connections. Ceramic plates having high thermal conductivity are affixed at the ends of the thermocouple for heat spreading to ensure a uniform temperature at either end. The ceramic plates and metal interconnects are assumed to have different uniform temperatures at the top and bottom. Current density is also assumed to be constant over the areas of the junctions between different materials.

For most practical applications below 540°C, the use of resistance–temperature detectors (RTDs) is a good choice. RTDs use the principle that some materials (e.g., platinum) increase their electrical resistances with the increase of temperature. With very stable electrical properties, platinum exhibits a linear resistance–temperature relation over a wide range of temperature. This has made platinum to be the best material for RTDs. In fact, RTDs are more accurate and have better sensitivity and linearity than thermocouples. By correlating the resistance of an RTD to temperature, the RTD can be used to measure temperature precisely. However, RTDs require external power sources, have slower response time and narrower measuring range, and are more expensive compared with thermocouples.

Another alternative temperature measuring device is thermistor, which is a thermally sensitive resistor. Thermistor uses a semiconductor device to change its resistance. By comparing with thermocouples and RTDs, thermistors have the narrowest measuring range, lowest stability and linearity, highest sensitivity, and comparable accuracy and response time to thermocouples.

10.2.3 Control System

The control system in a dynamometer is one of the key elements to making stable operation of the dynamometer and getting accurate data. In motor testing, the precise control of rotating speed and torque of the dynamometer is required. This is achieved automatically through the digital dynamometer controller, which is integrated into the testing system.

An absorption dynamometer acts as a load that is driven by the prime motor that is under test. The dynamometer can be equipped with two types of control systems, that is, constant torque and constant speed, to provide different test conditions [10.19]. With a torque regulator, the dynamometer operates at a set torque, while the prime motor operates at any speed and develops the torque that has been set. Similarly, with a speed regulator, it develops whatever torque is necessary to force the prime motor to operate at the set speed.

10.2.4 Data Acquisition System

During motor testing, many parameters need to be measured and recorded, including voltage, current, power factor, rotating speed, rotating direction, torque, electrical power, mechanical power, efficiency, and switch cutout speed, to name a few. The testing facility should have the capability to deal with the majority of the requirements of the motors, and the architecture allows customized tests to be easily added.

Data acquisition systems are used to collect motor operation characteristic data in motor testing. A typical data acquisition system consists of (1) data acquisition hardware, (2) sensors (e.g., torque transducers) for measuring motor operation characteristic data, (3) signal conditioning hardware (e.g., filter), (4) control and data acquisition software, (5) analog-to-digital convertor, and (6) a computer controlling data acquisition software. Most dynamometers are equipped with a data acquisition system. In selecting a proper data acquisition system, high accuracy, fast processing speed, and high-channel-count capacity are the main factors to be considered.

10.2.5 Torque Transducer

During motor testing, motor torque can be measured by connecting the testing motor to the dynamometer through a torque transducer or torque sensor, as shown in Figure 10.6. Torque transducers fall into two categories of measurement [10.20]:

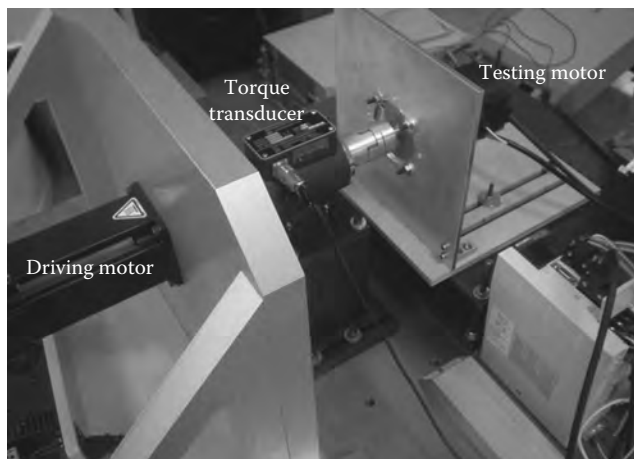


FIGURE 10.6

A torque transducer is placed between the testing motor and the driving motor.

- Reaction torque transducers. This type of transducers are nonrotating torque measuring devices and typically mounted in fixed positions. With no moving parts, reaction torque transducers offer long term reliability. They are suitable for a wide array of torque measurement applications including torsional testing machines, brake testing, bearing friction studies, and dynamometer testing.
- Rotary torque transducers. This type of transducers employs a freely rotating shaft within a fixed housing. When the shaft is torsionally stressed, a proportional change in the signal is observed. With the broad measurement range and high-precision measurement capabilities, rotary torque transducers are extensively used in motor testing, automotive engine testing, fan testing, and hydraulic pump testing. Torque-angle data obtained from motor testing are often used to draw torque-time or torque-angle plots that reveal the actual motor performance characteristics.

10.2.6 Power Supply

Power supply is a basic equipment in a motor test laboratory to provide the required power not only for the motor testing system and instruments but also for the control system for controlling the testing environment. It is widely accepted that the quality of power supply can significantly impact the motor efficiency. For instance, a phase unbalance of just 2% can increase losses by 25% [10.21]. Harmonics resulted from the voltage distortion can increase motor losses, produce vibration and noise, lower motor torque, induce torque pulsation, and result in motor overheating.

The power supply system must have a capability to provide a wide range of AC or DC voltage and current. In order to obtain accurate test results, the supply power, voltage, current, and frequency must be controlled within certain accuracies, defined by different standards (Table 10.2).

It can be seen that the requirements of IEEE Standard 112-B are significantly stricter than those of IEC 34-2. Furthermore, because harmonic distortion can increase steeply the losses of electric motors, the total harmonic distortion of the three-phase power supply should be limited within 1%. In cases that significant nonlinear loads are involved, harmonic filters may be required to reduce harmonic distortion.

10.2.7 Motor Testbed

Because testing motors come in a wide range of frame sizes, weights, loads, speeds, and power outputs, the motor testbed must be able to accommodate various testing motors,

TABLE 10.2
Accuracies of Power Supply

Item	Standard IEC 34-2 [10.13]	Standard IEEE 112 (Method B) [10.7]
Power (W)	±1.0%	±0.2%
Current (I)	±0.5%	±0.2%
Voltage (V)	±0.5%	±0.2%
Frequency (Hz)	±0.5%	±0.1%

especially large-scale motors. In order to reduce the impact of vibration on motor test, good vibrating damping materials such as cast iron are commonly used to construct motor testbeds. To firmly hold large testing motors on the testbed, T-shaped slots are usually made on the testbed. Small- and medium-sized motors can be directly installed on an L-shaped stand. During a motor testing process, a good alignment between the testing motor on the testbed and the dynamometer must be maintained.

10.3 Testing Load Level

There are a variety of methods to generate rotating torque loads to testing motors, such as the use of hysteresis brakes, eddy-current brakes, water brakes, and AC vector drives.

Motor performance test techniques generally fall into three categories: no-load, signature, and load tests. Each of these techniques contributes useful information to motor performance to a certain extent. However, each also has its limitation.

- No-load test

As the simplest test technique, no-load test is also called *open-circuit test*, which is widely employed in the motor industry as one of the standard tests. During the testing, the rated voltage (balanced) and frequency are applied to the testing motor, and the motor shaft is uncoupled to any load. The resulting current, power, and speed are measured and compared to similar measured data made on a master motor. Therefore, this type of test requires less use of the test equipment and instrumentation. Three types of no-load test can be provided: disconnected, inferred no load, and measured no load. For no-load test, the motor torque is close to zero, and current is very small.

According to IEEE Standard 112 [10.7], the measured input power in a no-load test is the total of the losses in the motor, including copper, friction, windage, and core losses. To determine the friction and windage losses, subtract the copper losses from the total loss (i.e., input power) at each of the test voltage points and plot the resulting power curve versus voltage, extending the curve to zero voltage. The intercept with the zero-voltage axis is the friction and windage losses. The core losses at each test voltage are obtained by subtracting the value of friction and windage losses from the input power minus copper losses.

- Load point test

Many different load steps are programmed for motor testing, for instance, from 25% to 150% load levels. There are two test control modes: (1) speed mode in which a speed is established by the dynamometer and torque is produced based on the capability of the motor and (2) the torque mode in which a load is established by the dynamometer and speed is produced based on the capability of the motor. At each load level, the parameters that are measured include speed, torque, voltage, and current.

In practice, the load tests are conducted at the load level of 25%, 50%, 75%, 100%, 125%, and 150%. Among these, the 100% load point is most commonly adopted. If the speed mode testing is used, the test speed is set at the rated speed. Similarly, for the load mode testing, the torque is set at the rated torque.

10.4 Testing Methods

There are a number of testing methods for electric machines. According to the testing setup, motor testing methods can be classified as [10.22]

- Mechanic differential testing method
- Back-to-back testing method
- Indirect loading testing (e.g., phantom, two-frequency, and inverse-driven) method
- Forward short-circuit testing method
- Variable inertia testing method
- Mixed-frequency testing method

Each of these testing methods has its own pros and cons. Among them, the most accurate and basic testing method is to directly load the testing motor either mechanically or electrically. Some of these testing methods are addressed in the following sections.

10.4.1 Mechanical Differential Testing Method

The mechanical differential method uses two identical motors, coupled with their main shafts through a differential cage. The two motors are connected to a common power supply in such a way that they rotate in opposite directions. As two motors rotate at the same speed, the differential cage remains stationary. If this cage is rotated, one of the motors accelerates and begins to act as a generator, while the other reduces its speed and commences to supply mechanical power to the generator through the differential gearbox. By adjusting the speed of rotation of the differential cage, it is possible to vary the loading of both motor and generator from no load to full load.

10.4.2 Back-to-Back Testing Method

This testing method also uses two motors that are not necessarily identical. This testing method has two distinct merits as the economy and the accuracy. In this method, the major part of the test power is circulated rather than dissipated, and the loss is measured as a net input rather than as the small difference between two separately measured large powers. As shown in Figure 10.7, the two motors are placed back to back each other and coupled to a floating gearbox. The two electric machines are powered from the same three-phase power supply. When they run at different speeds ω_1 and ω_2 , respectively, the speed differences ($\omega_1 - \omega_2$) can be adjusted by the floating gearbox [10.23]. A torque booster is mounted on the gearbox housing and arranged to boost the shaft rotation and to exert a reaction torque on the housing. The torque booster motor can drive either the input shaft, the output shaft, or the intermediate shafts. Thus, the power loss of the gearbox can be compensated by the torque booster to ensure that the torques on both sides of the gearbox are equal.

10.4.3 Indirect Loading Testing Method

Direct loading methods often suffer from high costs due to the complex coupling and loading mechanisms. To be able to conveniently perform heat run tests on electric motors

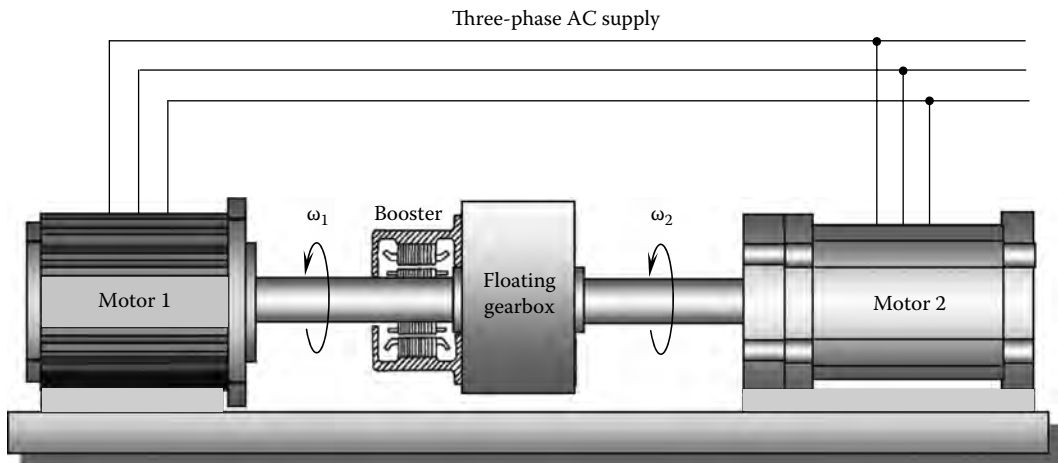


FIGURE 10.7
Back-to-back test using a floating gearbox.

with indirect loading is very important for motor manufacturers. Indirect loading schemes include phantom loading methods, two-frequency methods, and inverse-driven methods. In the phantom testing method, two identical motors are connected electrically together, with their losses supplied externally [10.24]. With indirect loading on the testing motors, no loading machine is required, nor have the test motors to be mechanically coupled. Experiments on electric machines have shown that the test simulates a true temperature run with a high degree of accuracy.

As shown in Figure 10.8, with the phantom loading method, the phase voltage of stator winding has both AC and DC components. Thus, the phase voltage of stator winding is the superimposition of DC and AC components:

$$v = \sqrt{2}v_1 \cos(2\pi f_1) + V_{DC} \quad (10.3)$$

where the magnitude of V_{DC} depends upon the connection pattern of the state winding on phantom loading.

Figure 10.8 depicts the phantom loading testing for two identical star-connected motors, without mechanical coupling between them [10.25]. Balanced three-phase voltages are applied parallelly on the star-connected stator windings of both the main and auxiliary motors. The DC current is injected into the windings in series, and a stationary DC field appears in the stator-rotor air gap. Thus, both AC and DC circuits are symmetric with respect to the AC and DC power supply, respectively. In such a configuration, the AC and DC sources can feed the two star-connected circuits at the same time. However, this configuration requires that the voltage supply is $\sqrt{3}$ times higher than the normal value if the motors are connected in delta, as often encountered in industrial applications.

Most industrial-sized IMs are connected in a delta connection, as shown in Figure 10.9 [10.26]. It can be seen from this figure how the DC loading current is injected into the windings of a delta-connected motor. The three-phase AC power supply is applied on the main motor and the auxiliary motors in a parallel manner. Three resistors R are inserted into the supply source to prevent the DC current from straying into the supply sources.

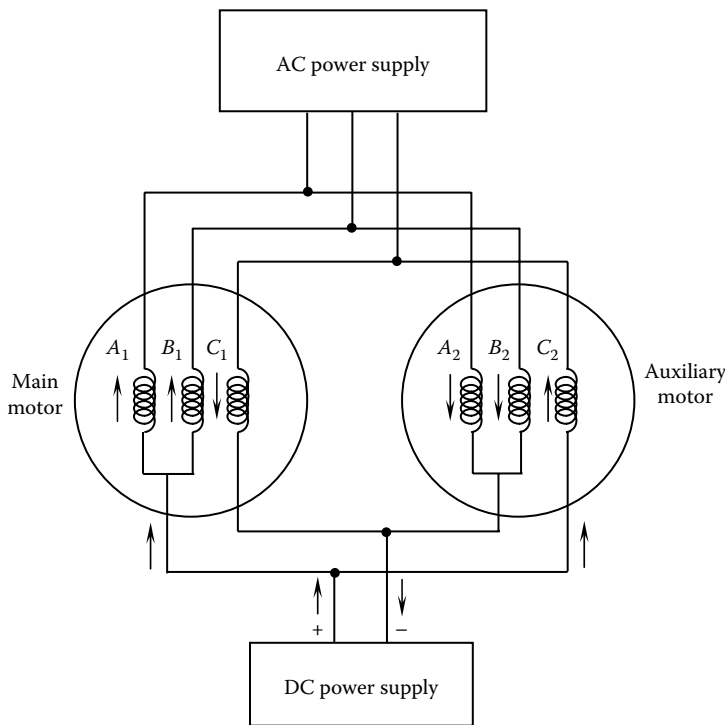


FIGURE 10.8
Phantom loading with star-connected stator windings.

The capacitor C is connected in parallel with the DC source to provide a path for the AC to feed into the auxiliary motor. Switch S should be closed during the starting process. The effective RMS values of the combined AC and DC currents flowing in each of the three-phase windings with such phantom loading method should be equal to the rated RMS currents in the motor windings [10.27].

The use of the two-frequency testing method may be traced back to the 1920s, when Ytterberg proposed a concept of connecting two voltage suppliers at different frequencies in series to produce synthetic loading of induction machine [10.28,10.29]. This method is generally known today as the two-frequency or dual-frequency method.

There are two possible loading schematics available in the two-frequency method, as presented in Figure 10.10. In each method, the main voltage v_1 has a rated frequency f_1 , and the auxiliary voltage v_2 has a lower frequency f_2 , where $f_2 = (0.60 - 0.95) f_1$. The magnitude of the auxiliary voltage v_2 is about 5%–25% of the main voltage f_1 . Hence, the voltage at the stator winding becomes

$$v = \sqrt{2}v_1 \cos(2\pi f_1) + \sqrt{2}v_2 \cos(2\pi f_2) \quad (10.4)$$

Therefore, the IM draws motoring current with respect to the higher-speed field and supplies generating current with respect to the lower-speed field. The two currents beat together to form a modulated current, which is made equivalent to the normal load current for the motor [10.30].

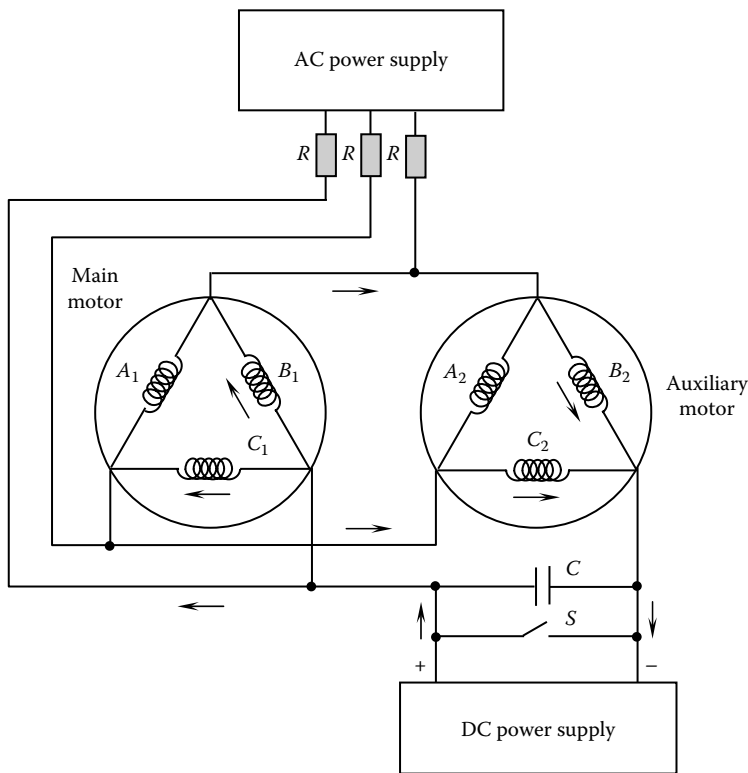


FIGURE 10.9
Phantom loading with delta-connected stator windings.

It is noted that during the testing process, the two induction machines are only coupled electrically together without mechanical shaft coupling. In this arrangement, the electromagnetic torque oscillates at the frequency of $(f_1 - f_2)$. Such oscillations may cause serious problems in vertically mounted motors [10.31]. Essentially, two induction machines operate concurrently as motors and generators.

An inverter-driven method uses a microprocessor or a digital signal processor (DSP) to generate the necessary logical control signals for PWM inverters to continually accelerate and decelerate the induction machine (Figure 10.11). The use of the power electronic inverter allows comprehensive control over all the quantities that determine the rate at which the power is dissipated inside the induction machine.

10.4.4 Forward Short-Circuit Testing Method

This testing method is an alternate loading method that manufacturers could employ with a minimum capital expenditure, especially suitable for testing ultra-large electric motors with outputs greater than 1 MW [10.32]. This conventional indirect loading method has been applied by a leading motor manufacturer for more than a half century. In this method, the motor under testing is fed with a variable frequency power supply or an alternator and is driven at the rated speed by a DC motor. The speed

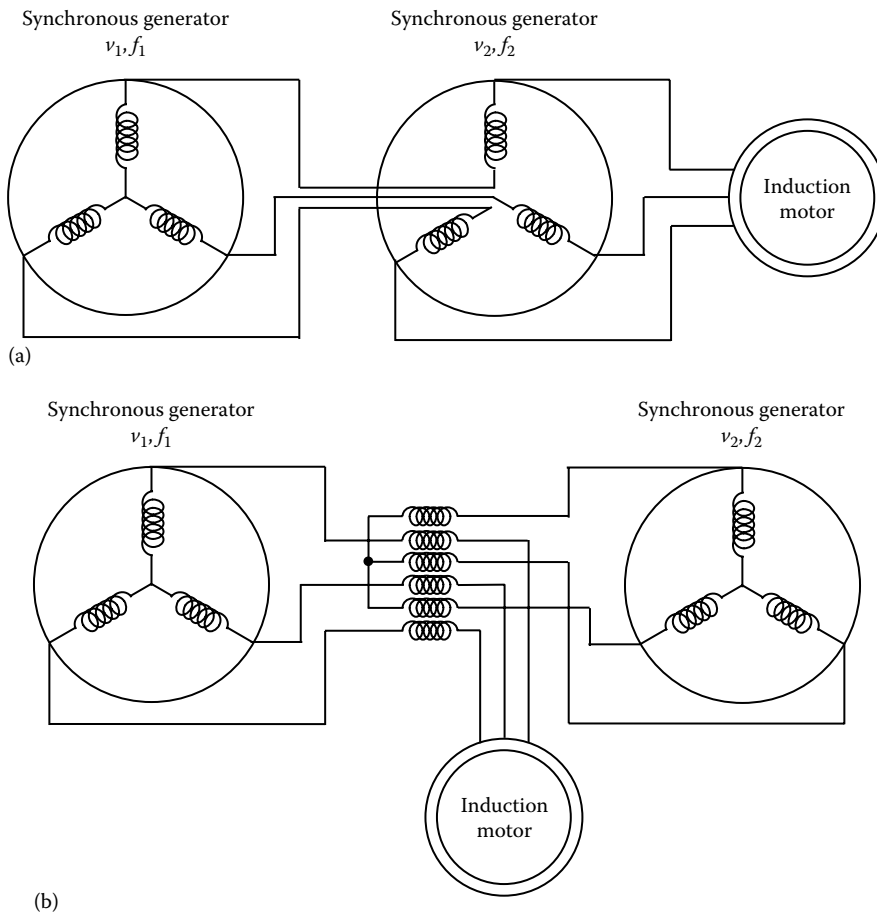


FIGURE 10.10
Two possible two-frequency loading methods.

difference between the test machine and the alternator can be correlated by adjusting the excitation of the alternator. The forward short-circuit testing method is given in Figure 10.12.

10.4.5 Variable Inertia Testing Method

Full-load temperature-rise testing of induction motors can be done by using the variable inertia testing method. This method, as a pure mechanical method in nature, is to load a test machine shaft directly with a flywheel (called “free-running inertia”) and does not need to connect electrically the test machine to any other machines.

In a variable inertia testing, the testing motor is connected to the flywheel through a gearbox. This gearbox has a nonconstant speed ratio that causes the inertia referred to the motor shaft to vary smoothly with the shaft rotation angle. The inertia variation results in the change in the shaft torque and in turn the change in the rotor speed [10.33].

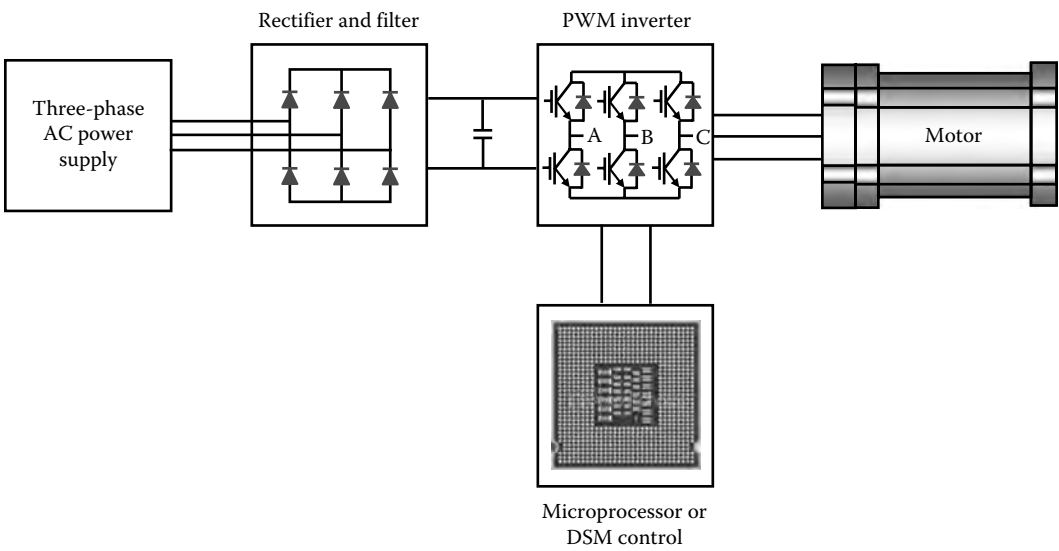


FIGURE 10.11
Block diagram of inverter-driven method.

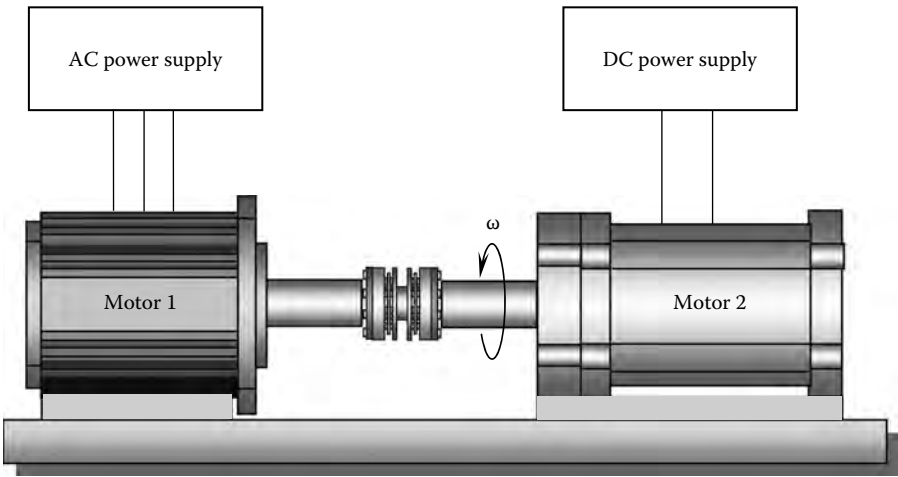


FIGURE 10.12
Forward short-circuit testing method.

10.5 Off-Line Motor Testing

Off-line testing, or static testing, is a well-known means to comprehensively test electric motors. This testing method measures the integrity of the insulation and finds stator winding faults before they become catastrophic failures. A complete set of off-line tests

include winding resistance test, megohm test, PI or dielectric absorption, step-voltage or high-potential (hipot) test, and surge test [10.34]. Off-line testing is also used as a quality assurance tool.

10.5.1 Winding Electrical Resistance Testing

Electrical resistance testing involves the measurements of individual coil resistance and phase resistance using ohmmeters. For a three-phase motor, measured phase electrical resistance data (i.e., U-V, V-W, and W-U) should be well within 5% of each other. For large-size motors, the phase resistance imbalances can be much lower. Variations between phases larger than the tolerance range for resistance are not allowed for any machine. As an example, the measured electrical resistance data of two large-size motors are shown in Table 10.3.

By taking a resistance reading of each phase, high-resistance connection can be identified. If there is a resistive imbalance exceeding a predetermined level, it can be caused by the circuit and/or the windings. In many cases, a high-resistance connection results from a solder joint between coils. Otherwise, an abnormal winding must be replaced.

10.5.2 Megohm Testing

Megohm testing, or insulation resistance testing, is to measure stator winding insulation resistance. This can help detect deteriorating insulation where the windings may potentially short to the ground. Megohm tests are usually conducted at constant DC voltages of 500–10,000 V, according to the motor's rated voltage. Readings of insulation resistance are taken after the DC voltage has been applied on the motor windings for at least 1 min. In a megohm test, the motor housing must be grounded for the safety consideration.

The insulation resistance of electric machine windings strongly depends on the insulation materials and the surface condition of the material. Generally, the insulation resistance varies proportionally with the insulation thickness and inversely with conductor surface area and temperature. Low megohm values are an indication of impending failure. However, high megohm values do not insure a good motor. Performing a PI testing can further help determine poor insulation systems. It is recommended by IEEE 43-2000 [10.1] that (1) for most machines with random-wound stator windings and form-wound windings rated below 1 kV, the minimum insulation resistance is 5 M- Ω ; (2) for most DC armature and AC windings built after about 1970 (form-wound coils) it is 100 M- Ω ; and (3) for most windings made before about 1970 (all field windings) it is (kV + 1) M- Ω ,

TABLE 10.3

Measured Electrical Resistances from Two Large-Size Motors

Phase-to-Phase	Electrical Resistance (Ω)	Phase Resistance Unbalance (%)
U1-V1	0.02488	<0.5
V1-W1	0.02487	
W1-U1	0.02476	
U2-V2	0.02477	<0.2
V2-W2	0.02480	
W2-U2	0.02481	

where kV is the rated machine terminal-to-terminal voltage. It should be noted that all measured insulation resistances must be corrected to the temperature of 40°C. To get accurate measurements, the testing winding must be clean and free of moisture, and the winding temperature stable.

The electrical resistance R of a material is proportional to the resistivity of the material ρ (in $\Omega\text{-m}$) and the material length l (in m) and inversely proportional to the cross-sectional area of the material A (in m^2), that is,

$$R = \frac{\rho l}{A} \quad (10.5)$$

where ρ is a function of temperature T ,

$$\rho(T) = \rho_o(T_o) [1 + \alpha(T - T_o)] \quad (10.6)$$

where

α is the temperature coefficient of resistance of the material

T_o is the reference temperature

Substituting Equation 10.6 into Equation 10.5, it yields

$$R(T) = \frac{l}{A} \rho_o(T_o) [1 + \alpha(T - T_o)] = R_o(T_o) [1 + \alpha(T - T_o)] \quad (10.7)$$

Thus, a measured resistance R_m at an arbitrary measuring temperature T_m can be converted into the resistance at the reference temperature, for example, $T_o = 40^\circ\text{C}$:

$$R(T = 40^\circ\text{C}) = \frac{R_m(T_m)}{1 - \alpha(T_m - 40^\circ\text{C})} \quad (10.8)$$

This method can be useful in comparing the measured resistances of various materials at different measuring temperatures.

10.5.3 Polarization Index Testing

The purpose of polarization index (PI) testing is to determine the condition of motor insulation. In fact, the PI testing measures the time required for molecules of insulation to align (i.e., polarize) from their random orientation to resist the flow of current. According to IEEE Standard 43-2000 [10.1], the PI is normally defined as the ratio of 10 min insulation resistance to the 1 min insulation resistance. This PI ratio provides an indication of the condition of the winding. As recommended by IEEE 43-2000, the minimum value of PI is 1.5 for the thermal class A and 2.0 for the classes B, F, and H.

The winding surface condition can significantly affect the insulation resistance or PI. For instance, carbon dust or other contaminations on winding insulation surfaces may become partially conductive when exposed to moisture or oil and thus can markedly lower the insulation resistance or PI. Lower PI values indicate that the winding insulation may be damaged or influenced by contamination (e.g., dust and moisture) or/and embrittlement.

10.5.4 High-Potential Testing

The objective of the hipot testing is to verify the dielectric voltage withstand of motor insulation system. It applies a very high DC voltage to the test components (e.g., stator and motor) to measure overall insulation resistance to ground. The results provide very useful information on insulation dielectric strength. Today, this test is mandated to protect users from electric shock, fire, or smoke damage.

It is recommended by IEEE Standard 95-2002 [10.4] that the maximum test voltage is double the motor's rated voltage plus 1000 V. At the beginning of the test, the motor frame is grounded first, and a DC voltage is applied on the insulation system with slow increase in step increments until reaching the maximum test voltage. This operation avoids an electric shock on the insulation system. During the process, the leakage current data are recorded and plotted against the corresponding DC voltages on the computer monitor.

10.5.5 Surge Testing

It is reported that copper-to-copper faults are the main cause of over 80% of all winding-related failures [10.35]. During manufacturing processes, winding insulations may be damaged from bending, twining, or wrapping. The aim of winding insulation surge testing is to detect faults in turn-to-turn, coil-to-coil, and phase-to-phase insulations that cannot be discovered by other testing methods (e.g., megohm or hipot testing). Using this technique as a preventative maintenance tool, the damaged windings can be repaired or replaced to avoid motor failure in applications, as well as caused emergency outage.

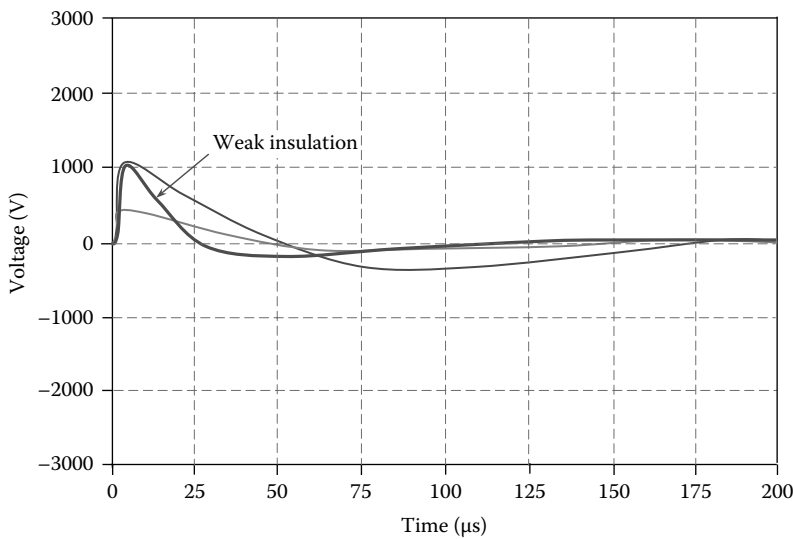
Surge testing applies very short high-voltage pulses to test electric winding and their insulation. The test is performed when all windings are installed into a motor but before they are connected to each other. This test is the most effective method of insuring integrity of the copper-to-copper insulations and diagnosing actual shorted conditions between turns, coils, and phases. With an oscilloscope, some winding insulation faults, such as insulation weakness, incipient faults, and other winding problems become visible on the oscilloscope screen.

Prior to surge testing, a number of measurements must be performed on the motor with acceptable results, including winding resistance, stator winding insulation, and PI measurements.

Surge testing has been known as surge comparison testing for a long time because insulation defects have been detected by comparing the waveforms from different phases of winding. While performing a surge comparison test, each phase of stator winding is tested against the others. Because all windings are assumed to be identical, the waveforms of the windings must coincide. Otherwise, there are some insulation issues in the winding. However, with today's advanced high-speed and high-accuracy measuring devices and computer systems, surge comparison is no longer necessary. Weak insulation can be diagnosed from frequency shifts and compared to successive waveforms within one phase [10.36]. Today, surge comparison testing is still used by some motor manufacturers.

In cases that there is a turn-to-turn short, the waveforms observed during testing exhibit distinct patterns on the display of the instrument. As a coil with weak insulation undergoes a surge test, the voltage applied on the coil can jump across the weak insulation, causing the shift of the waveform, as shown in Figure 10.13.

In a motor lab test, the surge testing is conducted with the rotor removed. But in a field testing, surge testing is performed with rotor installed.

**FIGURE 10.13**

Surge test results. Waveform shift as an indication of weak insulation.

10.5.6 Step-Voltage Testing

Step-voltage testing is designed to provide an additional evaluation of the motor winding insulation. Usually, step-voltage testing follows a successful PI testing. Actually, it is similar to the PI test but more extensive because a number of measurements are taken at various voltage step levels. The test is necessary to ensure that the winding insulation can withstand the normal voltage spikes during operation. However, step-voltage testing is not suitable for low-capacitance machines.

As shown in Figure 10.14, as the test progresses, a DC voltage is applied to all three phases of the winding and raised slowly to a preprogrammed voltage step level and held for a period of time. Then, it is raised to the next voltage step and held for a period of time. This process continues until the desired test voltage is reached. Typically, the steps in the test are made at 1 min intervals. The voltage interval is determined by dividing the maximum voltage with the number of steps. For small or middle motors, the voltage interval can be 500 V or 1000 V.

At each voltage step level, the leakage current is monitored and plotted graphically against time. From the figure, it can be seen that for stable insulation, the leakage current exhibits nearly a linear pattern with time. A highly nonlinear behavior or abnormal increase in leakage current may be indicative of insulation issues.

10.5.7 Determination of Rotor's Moment of Inertia

There are several experimental methods to determine the rotor moment of inertia. The traditional process for estimating the rotor moment of inertia involves operating a motor through an acceleration/deceleration profile [10.37].

In a motor test, for determining the moment of inertia of the rotor and its load, three torques, that is, the motor torque T_m , the load torque T_L , and the acceleration torque T_a (or deceleration torque T_d), vary with time during acceleration (or deceleration), as demonstrated in Figure 10.15. The relationships of these torque components are given as

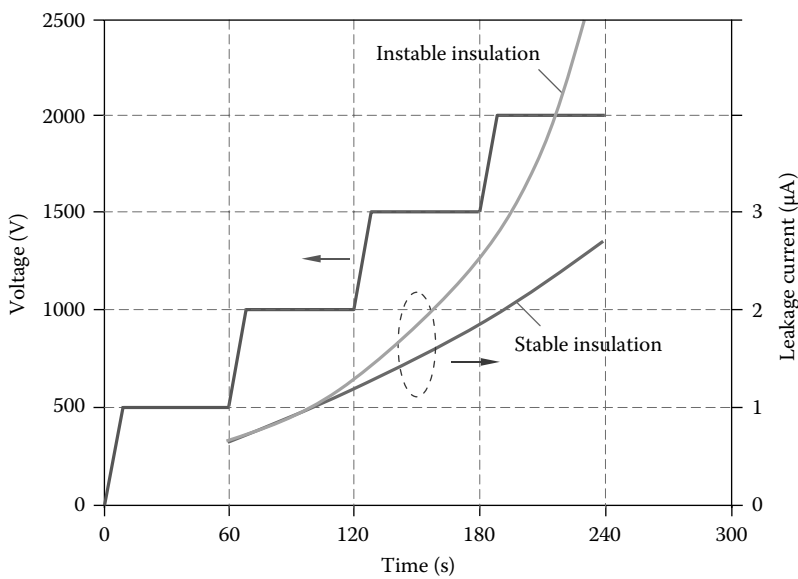


FIGURE 10.14
Step-voltage testing procedure.

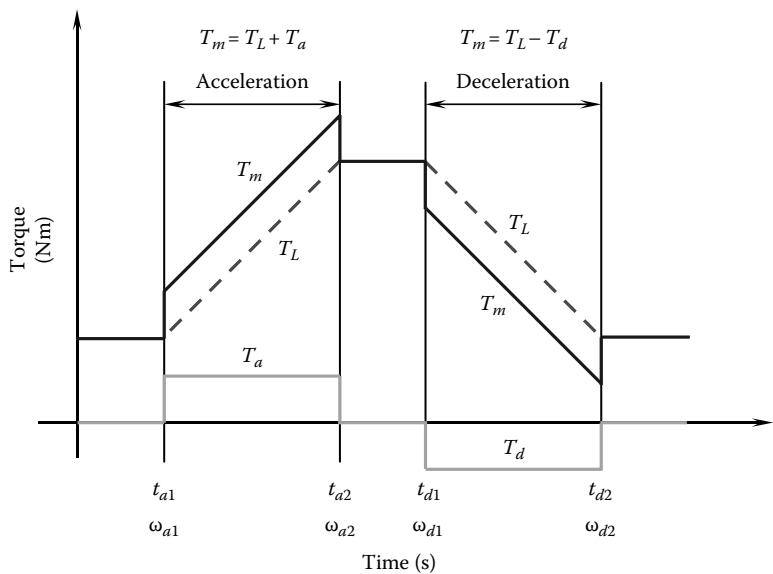


FIGURE 10.15
Relationships of motor torque, load torque, acceleration torque, and deceleration torque during the acceleration and deceleration processes in a test for determining the inertia of the test motor and load.

$$T_m = T_L + T_a \quad \text{during acceleration} \quad (10.9a)$$

$$T_m = T_L - T_d \quad \text{during deceleration} \quad (10.9b)$$

It is noted that the electric motor accelerates or decelerates in a linear manner. The acceleration torque T_a and the deceleration torque T_d are constant during the acceleration and deceleration processes. Then the rotor moment of inertia can be calculated by either of the following equations:

$$I \frac{d\omega}{dt} = T_m - T_L = T_a \quad \text{in the acceleration process} \quad (10.10a)$$

$$I \frac{d\omega}{dt} = T_m - T_L = -T_d \quad \text{in the deceleration process} \quad (10.10b)$$

Integrating both sides of the above equations corresponding to their time and angular intervals, the moment of inertia can be easily obtained. Assuming that the velocity difference in the acceleration process $\Delta\omega_a$ is equal to the velocity difference in the deceleration process $\Delta\omega_d$, that is, $|\Delta\omega_a| = |\Delta\omega_d| = \Delta\omega$, the averaged I value is given by the following equation [10.38]:

$$I = \frac{1}{2} \left[\frac{\int_{t_{a1}}^{t_{a2}} (T_m - T_L) dt}{\Delta\omega} + \frac{\int_{t_{d1}}^{t_{d2}} (T_m - T_L) dt}{-\Delta\omega} \right] \quad (10.11)$$

Rehm and Golownia [10.39] proposed a method to accelerate and decelerate a testing motor in a nonlinear manner. During acceleration and deceleration processes, the torque produced by the electric motor is sampled periodically, and the averaged torque during the acceleration/deceleration process is calculated by dividing the summation of all measure torque values with the number of torque samples. The rate of acceleration/deceleration is also determined. Therefore, the inertia (in unit of kg-m²) of the rotor and the load connected to the motor is calculated as

$$I = \frac{\left| \frac{1}{N_a} \sum_{i=1}^{N_a} T_{a,i} \right| + \left| \frac{1}{N_d} \sum_{j=1}^{N_d} T_{d,j} \right|}{\left| \frac{\Delta\omega_a}{t_a} \right| + \left| \frac{\Delta\omega_d}{t_d} \right|} \quad (10.12)$$

where

$T_{a,i}$ is the i th torque sample acquired during the motor acceleration process

$T_{d,j}$ is j th torque sample acquired during the motor deceleration process

N_a and N_d are the numbers of torque samples corresponding to acceleration and deceleration, respectively

ω_a and ω_d are the rotating speed changes in the unit of rad/s during acceleration and deceleration, respectively

t_a and t_d are the acceleration time and deceleration time, respectively

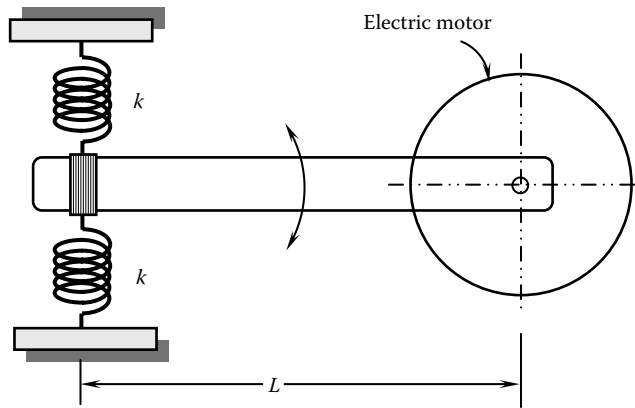


FIGURE 10.16
Oscillation test setup for measuring rotor moment of inertia.

Another effective method is called the oscillation method. To get an oscillating motion about the rotation point of the electric motor, a lever is attached to the rotor. On the end of the lever, an elastic spring with stiffness k is fixed on it. By measuring the period of time T of an oscillation, the moment of inertia can be calculated as [10.40]

$$I = \frac{kT^2}{4\pi^2} (1 - \zeta^2) \quad (10.13)$$

where ζ is the damping ratio. It is noted that Equation 10.12 is valid for one spring. When two springs are used to fix on the end of the lever in opposite direction, with the distance L from the springs to the rotation point (see Figure 10.16), the equation for calculating I becomes

$$I = \frac{2kT^2L^2}{4\pi^2} (1 - \zeta^2) \quad (10.14)$$

10.6 Online Motor Testing

Online testing or dynamic testing, as its name implies, is referred to the condition that power is supplied to motor for operation. Online testing is usually designed to evaluate the dynamic performance and operating characteristics of the entire motor. Therefore, online testing differs from offline testing in many aspects such as testing technologies and diagnostic methods.

10.6.1 Locked Rotor Testing

Locked rotor tests are used to determine the starting torque and current of electric motors, which are the two main indicators for assessing motor performance. In this

testing method, the testing motor is loaded to the maximum limits by applying maximum current to a hysteresis or powder brake type of dynamometer or by mechanically holding the rotor shaft from turning and energizing the stator. The voltage supplied to the stator is reduced to prevent stator winding overheating. At the rotor-locking condition, the motor operation parameters such as torque, current, and voltage are measured and displayed. Sometimes, locked rotor test is also called stalled torque test or blocked rotor test.

Locked rotor testing is very important for ensuring that the tested motor has sufficient torque to accelerate the motor when driving a specific load. If the motor cannot produce enough torque to overcome the friction in the load, the motor can be energized, but it will not drive the load. The test data can reflect the rationality of the magnetic circuit in the stator and rotor. Based on the test results, engineers can optimize the design and improve the production process.

However, this test is very hard on the motor. During testing, there is a large amount of current that flows into the motor, causing the motor to heat up rapidly. As a result, this test must be performed very quickly. With the locked rotor, a motor may draw up to six to eight times its rated current. The power supply used must be capable of regulating the motor voltage adequately during rapid changes in current to ensure the proper voltage is maintained when the data are being taken.

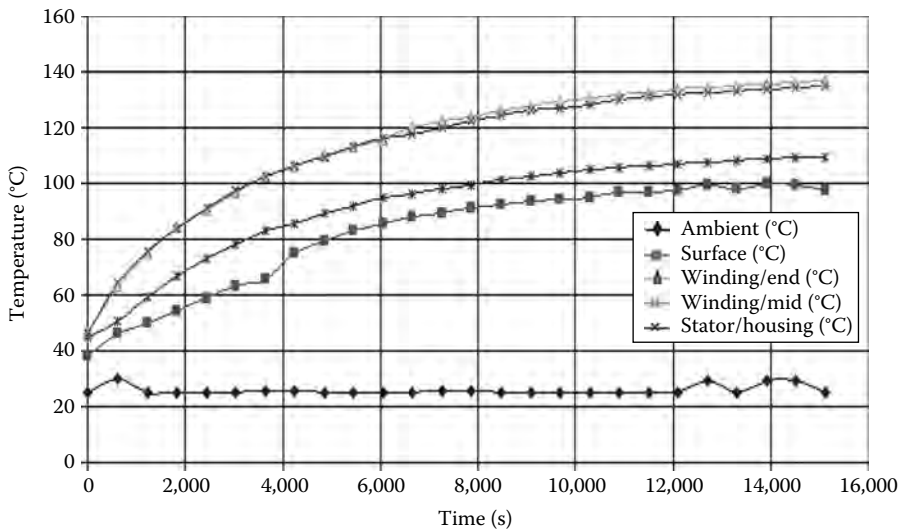
10.6.2 Motor Heat Run Testing

Motor heat run testing is sometimes known as motor continuous performance testing. This testing is extremely important to both motor manufacturers and end users. It was estimated that for every 10°C increase in temperature, the winding insulation life is reduced by half [10.41]. Exceeding temperatures can cause PM demagnetization and thermal aging of insulation materials.

The results of heat run tests are critical for verifying the engineering design and analysis and optimizing the motor design. More specifically, heat run tests can determine the maximum operating temperature of the tested motor and confirm that the maximum operating temperature is lower than the highest allowable temperature, which is defined by the motor insulation class based on an average 20,000 h lifetime.

A sustained heat run test is carried out at motor's full load. During the testing process, the temperature of the motor increases exponentially as various power losses are generated in the motor and converted into heat to dissipate to the environment. After a period of time, the motor reaches the thermal equilibrium condition, and the motor temperature tends to be constant. The time taken to reach equilibrium can vary widely depending on the machine size, cooling method, motor power losses, operation conditions, and ambient temperature. To obtain the temperature distribution of the motor, the temperatures of various motor components such as stator windings, bearings, PM, frame, endbells, encoder, and inlet and exit coolant, just to name a few, are measured and recorded. Temperatures on the rotating components (e.g., the shaft temperature at the shaft seal) can be conveniently measured using a thermal camera.

Ambient temperature T_a is the temperature of the air surrounding the motor. For most motor manufacturers, this temperature is defined at 40°C in their product catalogs. This does not mean the ambient temperature is set at 40°C during a heat run test. As a matter of fact, the ambient temperature in a test is controlled to maintain relatively constant at any given temperature.

**FIGURE 10.17**

Results of heat run test for a servomotor at 2500 rpm and 45 N-m torque.

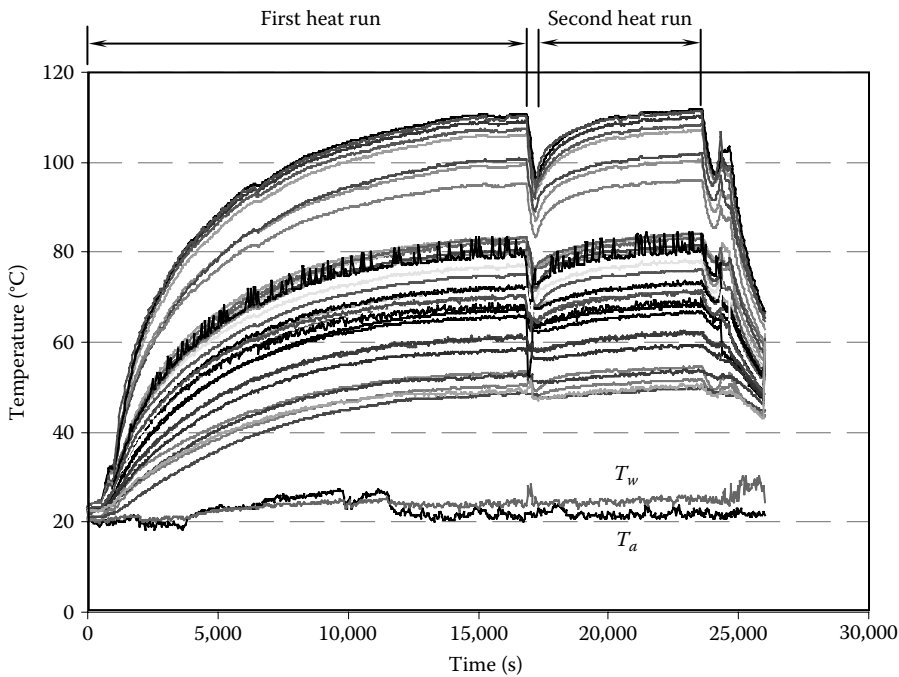
The temperature rise of a motor is the change in motor temperature with respect to the ambient temperature when power is applied to the motor. Obviously, the higher the ambient temperature, the lower the temperature rise of a motor. For instance, class B insulation is rated for a maximum operating temperature of 130°C. When the ambient temperature is 40°C, the allowable motor temperature rise is 90°C. As the ambient temperature becomes 50°C, the temperature rise reduces to 80°C. By considering the effect of hot spots occurring in motor windings, a 10°C hot-spot allowance is subtracted from the allowable temperature rise for the safety consideration. Thus, for $T_a = 40^\circ\text{C}$, the allowable temperature rise for class B insulation in the earlier example becomes 80°C rather than 90°C.

As an example, the results of a heat run test for a servomotor at 2500 rpm and 45 N-m torque are provided in Figure 10.17. It shows that it takes about 4.2 h to reach thermal equilibrium. While the temperatures of stator windings at the middle and end turns approach 140°C, the temperatures of motor housing reach 110°C. The ambient temperature is controlled to maintain in the range of 25°C–30°C.

For large-size motors, the time for reaching thermal equilibrium can be very long, up to more than 10 h. For these cases, performing heat run testing continuously can save a tremendous heating time and testing resources. As shown in Figure 10.18, when the first heat run completes, the motor temperature drops sharply. As the motor temperature becomes lower than the expected thermal equilibrium temperature, the second heat run test starts immediately. It can be seen from the figure that the heating time of the second test is only one-third of the first one.

10.6.3 Motor Efficiency Testing

The aim of motor efficiency testing is to determine the motor characteristic performance. Motor efficiency is one of the most important characteristic parameters for users and manufacturers. This parameter measures how effectively electrical energy is converted

**FIGURE 10.18**

Thermal testing is performed at the rated speed and rated torque until thermal equilibrium is achieved. The testing data show that the successive test significantly reduces the motor heating time.

into mechanical energy by a motor. A number of factors that may affect the overall motor efficiency include power supply quality, motor size, various power losses, the transmission and mechanical components, maintenance, and mismatch of the load to motor inertia.

Motor efficiency can be measured either directly or indirectly. Usually, direct measuring methods can provide accurate results but often require high-priced instruments/equipment and long test setup time. By definition, motor efficiency is the ratio of mechanical power output P_{out} over electrical power input P_{in} :

$$\eta = \frac{P_{out}}{P_{in}} \quad (10.15)$$

As an example, the mechanical output for an IM is the production of the motor torque and rotating speed, that is,

$$P_{out} = T\omega = T\omega_s(1-s) \quad (10.16)$$

where

s is the slip rate

ω and ω_s are the rotating shaft speed and rotating synchronous speed in rad/s, respectively

For a PM motor, $s=0$.

The electrical power input can be easily measured with good accuracy. The measurement of rotating speed is relatively simple and straightforward, with an achievable accuracy of ± 1 rpm. The torque measurement is obtained using a highly accurate torque transducer.

The indirect way of determining the motor efficiency involves the measurement of various power losses of the motor. The motor efficiency can be alternatively expressed as

$$\eta = \frac{P_{in} - \sum P_{loss}}{P_{in}} = \frac{P_{out}}{P_{out} + \sum P_{loss}} \quad (10.17)$$

Most of these losses such as copper, core, windage, and friction losses can be determined using no-load and load tests. However, stray load losses, which consist of miscellaneous losses except for major power losses, are hard to be directly measured. These losses may be estimated using some correlations from the literature or standards. For instance, the stray losses can be estimated using IEEE Standard 112-2004, which gives the percentage of stray losses to the rated load as 1.8% for the motor rating 1–90 kW, 1.5% for 91–375 kW, 1.2% for 376–1850 kW, and 0.9% for >1850 kW.

The test procedure is defined by the IEEE Standard 112-2004, which estimates the motor efficiency by the direct method [10.7]. As a contrast, IEC 34-2 Standard [10.13] estimates the motor efficiency using the indirect method. In fact, the indirect method has a significant degree of uncertainty, because of the instrumentation lower-accuracy specifications and the incorrect winding losses in relation to the temperature [10.42].

To avoid loading shock, the motor load maintains low levels and increases in small increments. Motor performance testing is conducted when a motor operates under normal environmental conditions.

10.6.4 Impulse Testing

Statistical data show that insulation faults are accounted for about a quarter to one-third of motor failures. The breakdown of insulating materials of motor winding can directly lead to short-circuit faults. Most stator failures often begin as turn-to-turn shorts caused by steep-fronted surges due to switching. These turn-to-turn shorts eventually develop into catastrophic copper-ground or phase-phase fault.

Impulse testing is performed by applying defined impulse voltage waveforms on the motor windings to assume lightning strike. The waveform shape, peak voltage, impedance, and application of the pulse vary and depend on the characteristics of the test motors. The purposes of the impulse test as special test are (1) to confirm the withstand of the motor's insulation against excessive voltages occurring during impulse testing and (2) to detect the weak insulation in the motor windings. In motor manufacturing, the on-off impulse voltage is normally generated by charging and discharging a capacitor in the motor windings. This steep-fronted voltage sets up a nonlinear voltage distribution that creates a turn-to-turn differential voltage and results in an oscillation between motor's inductance, capacitance, and resistance.

10.6.5 Cogging Torque Testing

Most applications of electric motors require smooth operation and constant torque. The examples include engraving machines, telescopes, nuclear magnetic resonance (NMR), and

optical instruments, just to name a few. Any pulsating torque can have a significant negative impact on the motor performance characteristics in these applications. Furthermore, cogging torque causes acoustic noise and vibration, as well as torque and speed ripples. As a result, it is highly desired to control the cogging torque to a lower level, say, less than 1% of the rated torque.

Cogging torque in a PM motor is generated from the interaction of the rotor-mounted PMs and the stator slots. Cogging measurement can be performed by either static (e.g., leverage measuring method), quasistatic, or dynamic methods. In most practical cases, the dynamic measurement method is adopted to determine cogging torque. With this method, the motor being tested has no input current and is driven by a dynamometer or motor. A torque transducer is installed between the tested motor and the dynamometer to measure the cogging torque produced with an oscilloscope. In practice, each type of motor uses different ways to measure the position of the machine being tested such as encoder, resolver, proximity sensor, and sensor bearing.

The diagram of a dynamic cogging torque measurement system is given in Figure 10.19. An electric motor is employed as the driving element. A torque transducer is placed between the motor being tested and the driving motor to detect the torque from the strain in the shaft. During the test, there is no input current to the motor being tested. The driving motor is controlled by a motor drive. In this case, the encoder is installed on the motor being tested to measure the rotor position.

It is noted that the measured torque contains the cogging torque of the driving motor. This requires that the driving motor must have very small or no-cogging torque to minimize the measuring errors. It has been proven that using a no-cogging effect, DC or hydraulic motor as the driving motor promises a very high accuracy of the measuring results. Since this setup works without any offset, the measurements are performed in a close range around the zero value.

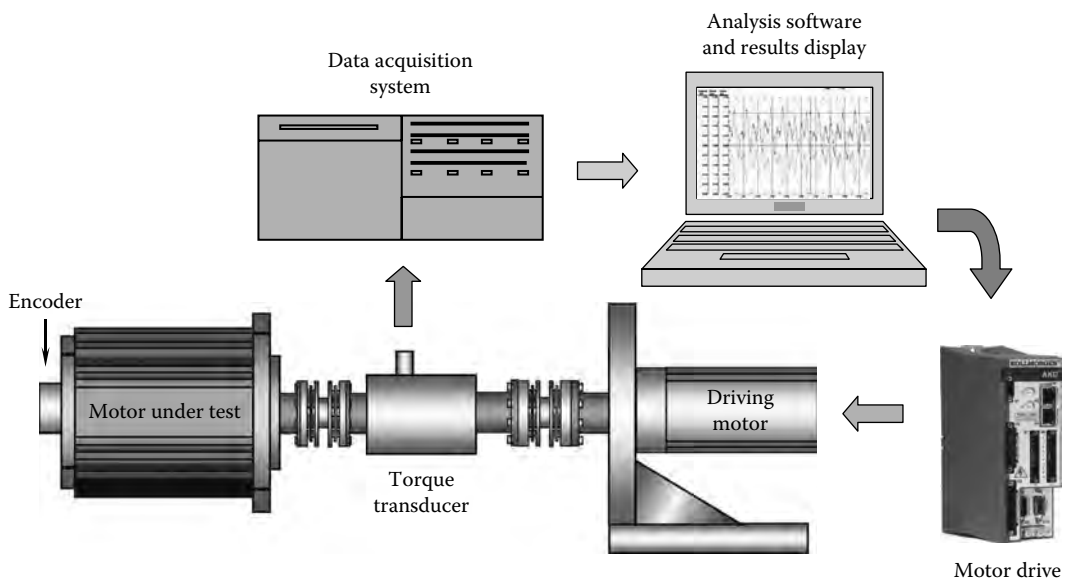
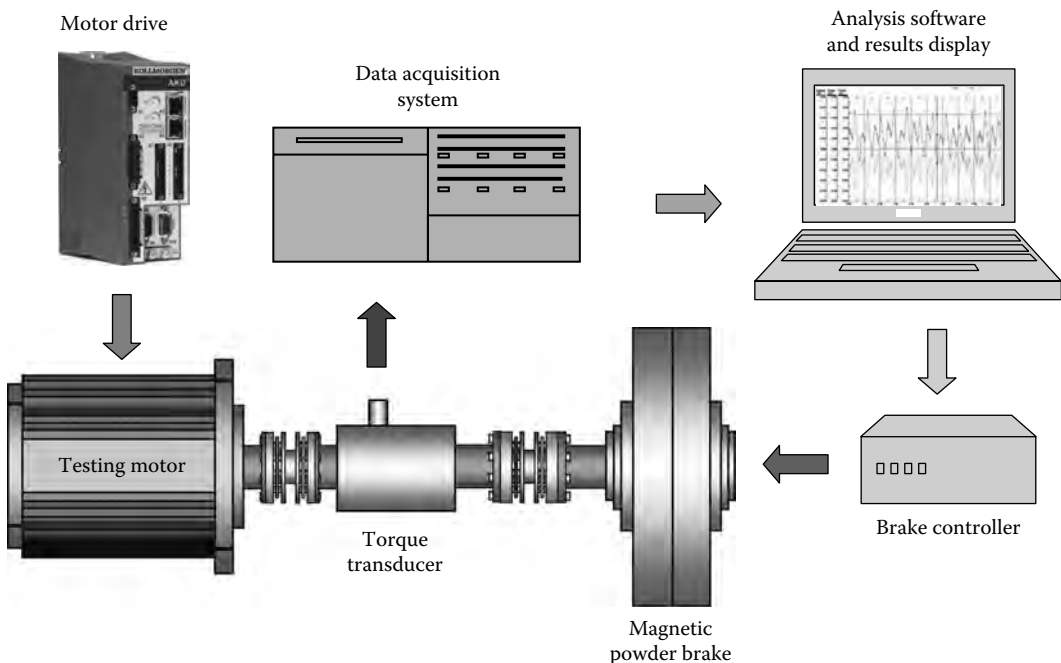


FIGURE 10.19

Dynamic cogging torque measurement employing a noncogging motor as a driving element.

**FIGURE 10.20**

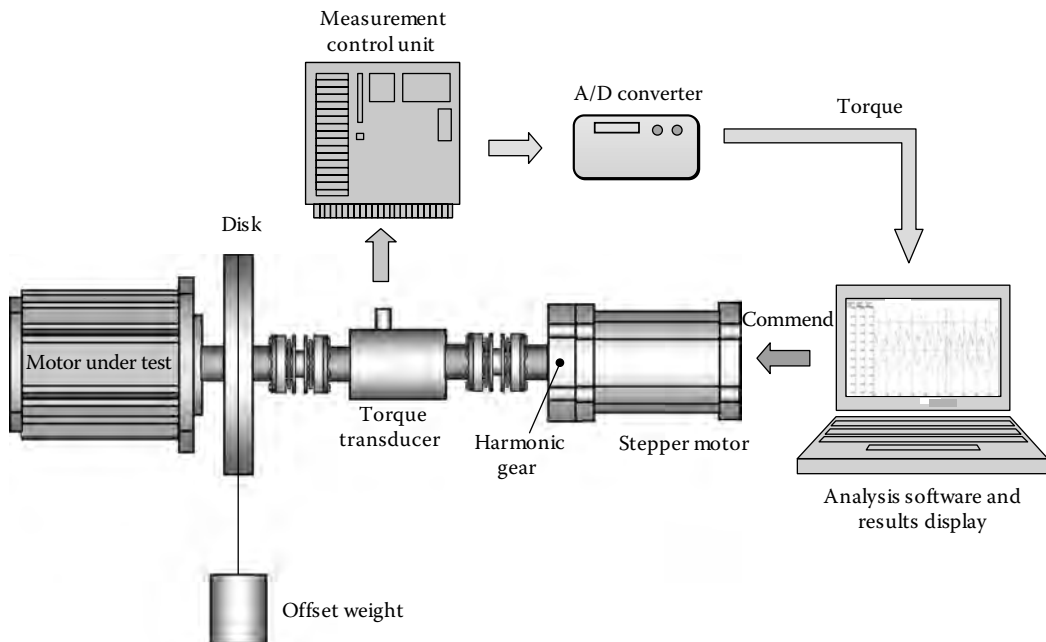
Alternative dynamic cogging torque measurement employing a magnetic powder brake as the driving element.

An alternative dynamic cogging torque measurement method is demonstrated in Figure 10.20. Unlike the previous testing configuration, the testing motor in this measurement method is the driving element. A magnetic powder dynamometer is used as the load device to provide the brake load to the testing motor. The resistance of the magnetic power dynamometer is controlled by the brake controller. The cogging torque of the testing motor is thus obtained from the measured torque data by the torque transducer.

A cogging torque testing system that uses a stepper motor as a driving element is shown in Figure 10.21. A harmonic gear is used to reduce the speed of the stepper motor to drive the testing motor. The high gear ratio avoids a measurable impact of the stepper motor's torque ripple to the cogging torque of the testing PM motor. The high harmonics of the stepper motor are filtered from the digital signal.

Similar to the previously addressed testing systems, a torque transducer is placed between the testing motor and stepper motor to measure the torque. The signal from the transducer is then fed into a measurement control unit. The measured torque is converted to digital values with an A/D converter and displayed on the computer monitor. As discussed by van Riesen et al. [10.43], the weight is used as an offset torque, which allows for the torque transducer to measure in its optimum scale range. The stepper motor driving the whole setup is controlled by two voltage sources. The resolution of the stepper motor can be switched from 1/1, 1/2, 1/4, to 1/8. By additionally varying the voltage from 0 to 8 V, the speed can be adjusted from 0 to 64 rpm.

The particularly prominent advantage of using the stepper motor is that it does not affect the measurement of the cogging torque of the testing motor, even though it has some cogging torque itself. In addition, because the stepper motor can determine the angular position of the testing motor, no position sensor is required.

**FIGURE 10.21**

Dynamic cogging torque measurement using a stepper motor as the driving element.

There are primarily two different cogging torque analysis criteria: the harmonic analysis and the peak-to-peak value of the cogging torque. Many motor manufacturers often use cogging torque as a percentage of the peak-to-peak cogging torque over the motor continuous torque. The measured cogging torque from a servomotor is given in Figure 10.22, showing the peak-to-peak value of 0.04 N·m. With the measured continuous torque of 1.52 N·m from the performance test, the percentage of cogging torque is 2.63%.

10.6.6 Torque Ripple Measurement

Torque ripple of electric motors has adverse effects on the surface finishing and tolerance control in some demanding motion control applications, such as machine tool spindle drives and optical devices. In some circumstances, torque ripple may excite resonances in the mechanical portion of the drive system and produce acoustic noise. Therefore, precise torque ripple measurements are essential for identifying root causes and optimizing motor design.

The experiment setup for measuring torque ripple is depicted in Figure 10.23. The eddy-current brake is used to provide loading to the testing motor. The motor is rigidly connected to a torque transducer for measuring the maximum torque load. Isolated current and voltage transducers are used since the signals can be connected to a single common point. This is a great convenience to the data acquisition system provided that additional instrumentation errors are negligible [10.44].

Tests are conducted at different load points and different speed levels (e.g., 80% and 100% of rated speed). Accuracy of the torque load during the torque ripple test is well within the manufacturing specification of the torque transducer. Electric grounding is made on all measuring instrument such as torque box, torque transducer, and output negative signal.

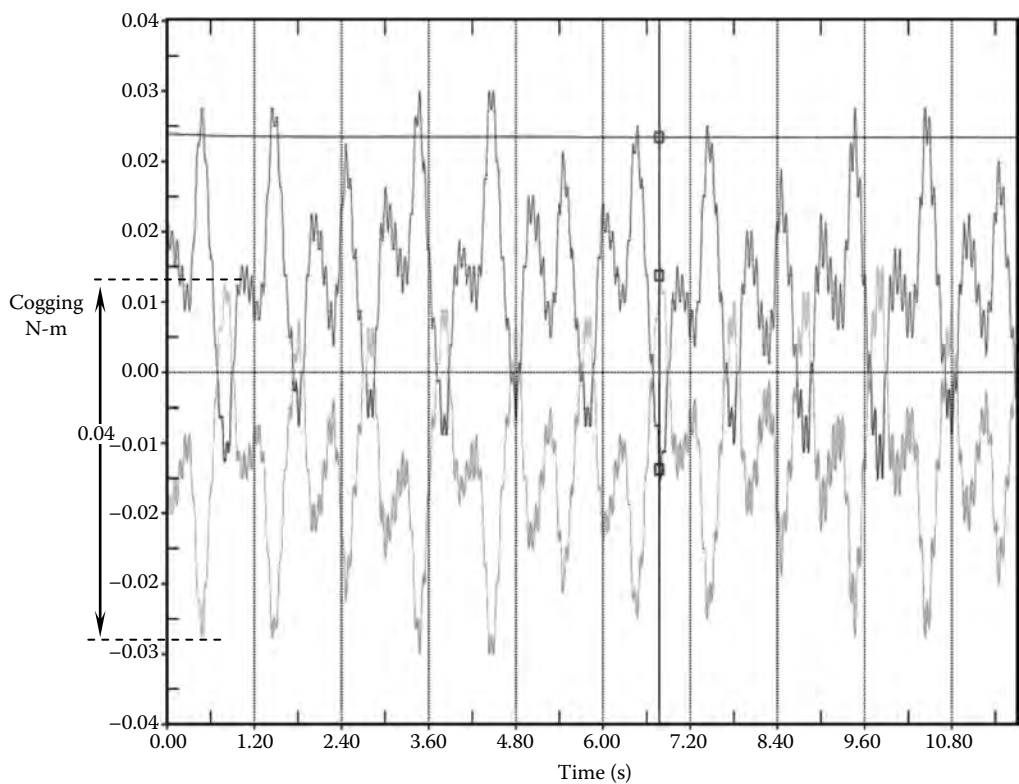


FIGURE 10.22
Measured cogging torque from a PM servomotor.

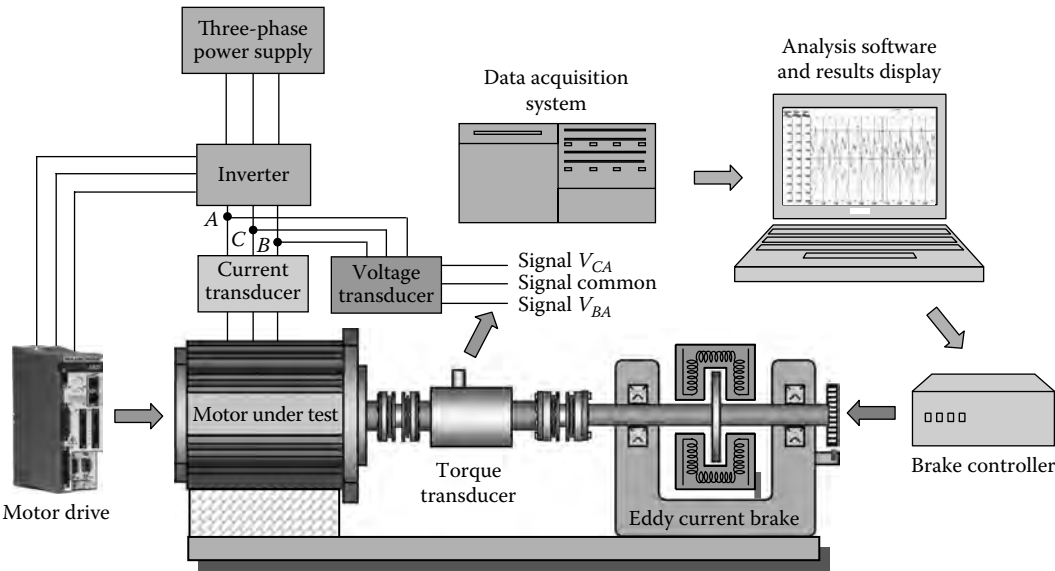


FIGURE 10.23
Experimental setup for torque ripple measurement.

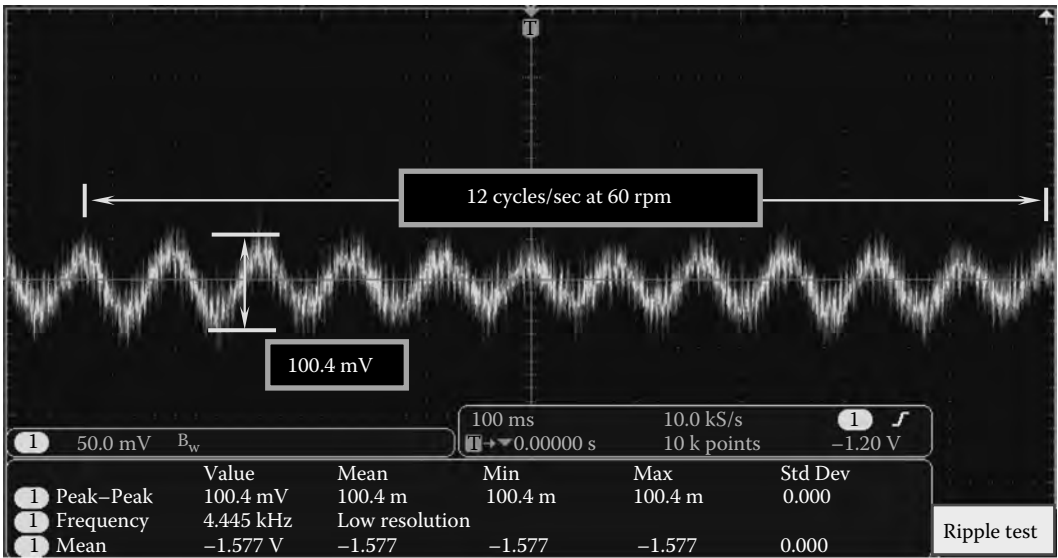


FIGURE 10.24
Results measured from a torque ripple test. The ratio of peak-to-peak torque ripple to rated torque is 5.4%.

The result obtained from a torque ripple test has shown that the peak-to-peak voltage is 100.4 mV (Figure 10.24). This value is converted into the peak-to-peak torque difference ΔT_p (where $\Delta T_p = T_{\max} - T_{\min}$). Thus, the torque ripple, which is defined as the ratio of ΔT_p to the main torque T_m , is determined to be about 5.4%.

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Modeling, Simulation, and Analysis of Electric Motors

In design and development of modern electric motors, a wide variety of engineering analyses and simulations must be carried out to ensure robust design and product optimization. These analyses may include, but not limited to, thermal analysis, CFD analysis, stress/strain analysis, resonance analysis, modal analysis, fatigue analysis, bucking analysis, burst analysis, motor mounting analysis, thermal expansion/contraction analysis, vibration and acoustic noise analysis, electromagnetic analysis, and design optimization analysis. The successful integration of these analyses into the product design cycle can significantly enhance the design quality, raise design standards, and accelerate design processes.

11.1 Computational Fluid Dynamics and Numerical Heat Transfer

As motor designs and processes grow in sophistication, motor-cooling problems become too complex to solve analytically. This forces engineers to perform numerically large simulations to gain insight into the details of fluid flow and heat transfer processes in motors. In fact, solutions generated by numerical methods can provide more insights than those by other methods in some heat transfer and fluid dynamics problems, for instance, transient fluid flow and heat transfer, 3D turbulent flows with temperature-dependent properties, multiphase flows, and compressible flows.

With the growing heat generation in modern electric motors, limited cooling capabilities may cause degradations of the motor performance and operation reliability. Today, CFD, being an integral part of the motor design for shortening the design cycle time and lowering the design cost, has played an important role to help engineers and designers gain insights of physical aspects of motor cooling. The use of CFD at the stage of the conceptual design enables motor engineers to explore alternative design options. The detailed results from a CFD simulation allow visualization of flow fields and temperature distributions in even the most inaccessible locations of a complex fluid flow and heat transfer system. A key advantage of CFD is that it provides the flexibility to readily change design parameters and determine the corresponding impacts of those changes on performance [11.1].

However, it must be noted that one of CFD issues is *garbage in, garbage out*. Despite the maturity of advanced CFD techniques, there is no guarantee that a CFD tool can automatically provide correct results. To obtain accurate simulation results, it requires thermal engineers to use their technical expertise and experiences in design and CFD work. Proper verification and validation with either experimental data or theoretical solutions are always essential parts of a CFD process.

11.1.1 Strategies in Modeling and Performing CFD Analysis

To make numerical simulations successful with available computational resources, some effective strategies may be adopted in modeling and performing CFD analysis:

- To take full advantage of geometric symmetry and periodic boundary conditions, a CFD analysis can be greatly simplified by modeling only a portion of the actual system.
- Some 3D fluid dynamics and heat transfer problems may be treated as 2D, axisymmetric problems.
- By getting rid of any unnecessary small features, which have little effect on simulation results, such as small faces, holes, edges, and fillets, it does not only simplify the problem but also make the simulation convergent. This is especially critical in cases that CFD models are converted directly from 3D CAD models, which usually contain a large amount of such small features.
- The mesh quality is the key for the success of a CFD analysis. It is highly designed to carefully select an appropriate meshing method and apply advanced meshing control techniques such as the refinement technique.
- In complex simulation problems, the iterations converge slowly and in some cases may even diverge. To ensure achieving convergence, it is suggested to run an analytical model with properly altered boundary and initial conditions. For instance, a common and effective strategy used in CFD codes for steady-state problems is to solve the unsteady form of the governing equations and then *march* the solution in time until the solution converges to a steady value. In this case, each time step is effectively an iteration, with the guess value at any time level being given by the solution at the previous time level [11.2].
- In order to facilitate convergence, the rotational speed should be initially set at a very low value and increased by small increments until reaching the normal rotating speed.
- For some large-size, complex geometry problems, a fully refined model may be too large and usually run into convergence problems. It is suggested to run the model with relatively coarse meshes first to quickly find out critical areas of interest. The critical areas are then successively refined to get the desired information in these areas.
- The addition of radiation heat transfer can significantly slow down the numerical calculation and increase the requirement of computer resources. In practice, the contribution from radiation to the overall heat transfer is generally insignificant except in a case where the temperature differences between motor components and/or between the motor and the environmental surrounding are relatively large.
- At an early design stage, some levels of approximation can be made for quickly exploring different designs.
- In dealing with heat conduction in silicon steel laminations, it is to be noted that the laminations have anisotropic thermal conductivity.

11.1.2 Rotating Flow Modeling

Rotating and swirl flows are commonly encountered in turbomachinery electric rotating machines and a variety of other applications. In an electric motor, while the stationary

parts such as the stator and housing are held statically, all parts attached to the rotor rotate at a certain velocity with respect to the rotor axis. Therefore, rotating flows are generated in the air gap between the rotor and stator and at the surroundings of fan blades, whereas interactive flows are mainly developed at the end-winding regions.

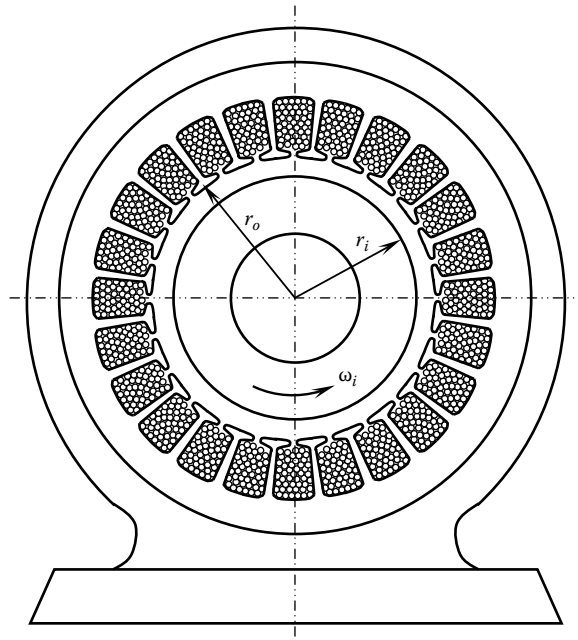
There are several approaches for modeling such rotational and interactional flows. The simplest approach is to use the single reference frame (SRF) method under the following conditions: (1) all the rotating parts rotate at the same speed with respect to a specified single axis, and (2) the problems involve moving parts and the stationary walls (as viewed from a stationary reference frame) can form surfaces of revolution with respect to the axis of rotation in the rotating reference frame [11.3]. With this approach, the entire computational domain is referred to as a single rotating reference frame. Thus, with certain restrictions, it can turn unsteady rotating flows in the stationary (inertial) reference frame into steady-state flows in the rotating (noninertial) reference frame. However, due to the restricted conditions mentioned previously, the application of this method is limited to simple problems.

When rotating parts in a system rotate about different rotating axes, or each of the rotating parts rotates about the same rotating axis but different angular speeds, or problems involve stationary components which cannot be described by surfaces of revolution, SRF is no longer applicable, and thus, the use of the multiple reference frame (MRF) method is required. With this method, the computational domain is divided into multiple domains: some are rotating and others stationary, with interfaces (which must be surfaces of revolution) separating these domains. This approach is a steady-state approximation where a rotating frame of reference is applied on the rotating domains and a stationary frame of reference is applied on the stationary domains. During the solution process, information is continually passed through the predefined interfaces between the regions. All rotating parts (e.g., a rotor) should be located inside of the rotating regions and modeled as emptiness or solid bodies. For a multiple-rotor system, the use of MRF allows individual rotors to rotate with different speeds, directions, and axes. The approaches of MRF and SRF are appropriate when the flow at the boundary between rotating and stationary regions is weakly affected by the interaction. These methods provide reasonable time-average simulation results in various applications.

The mixing plane model (MPM) is a variant of the MRF model for simulating flow through domains with one or more regions in relative motion. In this approach, each fluid region is treated as a steady-state problem. Flow field data from adjacent regions are passed as boundary conditions that are spatially averaged at the mixing plane interface.

The sliding mesh method has been developed for analyzing unsteady fluid flows. This method allows flow pattern calculations without the need of experimental data as rotor boundary conditions. As a transient approach, this method can provide the most accurate results for complex flow systems. With this method, the flow field is divided into two regions. One region, associated with the stationary components, remains stationary. Another region, associated with the rotating components, rotates relative to the stationary mesh. The two grids slide past each other in a time-dependent manner, exchanging information at the sliding interface [11.4]. At the interface, a conservative interpolation is used for both mass and momentum. The principal disadvantages for this method include the long calculation time and the required large computer resources.

The selection of appropriate model is critical for the success of computational simulation. In addition, the setup of the interface between rotating and stationary regions in MRF is also very important for the accuracy of numerical results. One example picked up here is the rotating air gap flow in a motor. As depicted in Figure 11.1, the rotor rotates counterclockwise with the rotating angular velocity ω_r , and the radii of the rotor and stator are r_i and r_o , respectively. This is a special case of the Taylor–Couette flow that has been

**FIGURE 11.1**

Schematic diagram of the formation of the air gap between the rotor and stator.

extensively studied for more than a century [11.5–11.8]. Except in the very thin boundary layers of cylinders, the tangential velocity profile of the concentric rotating flow can be theoretically derived as [11.6]

$$u_t = \frac{A}{2}r + \frac{B}{r} \quad (11.1)$$

where A and B are constants, determined from the boundary conditions at the rotor surface $u_{t,r=r_i} = r_i\omega_i$ and at the stator surface $u_{t,r=r_o} = 0$.

This problem can also be solved with a CFD model by defining multiple regions. The interface that separates the rotating and stationary regions is at $r=r_{int}$. As shown in Figure 11.2, the tangential velocity decreases in the thin boundary layer of the rotor and then increases with the radius in the rotating region until it reaches the maximum value at the interface (where $u_{t,max}=r_{int}\omega_i$). In the stationary region, the velocity is continuous at the interface and then decreases with the radius until it vanishes at the stator surface. The comparisons between the experimental solution and the results from both the small and large rotating regions are presented in Figure 11.3. The small rotating region is set up as close as possible to the rotor but without causing difficulty in model meshing. This figure shows the significant deviations between these three cases. This may suggest that for such a problem, the use of small rotating region provides better result than the large rotating region.

11.1.3 Porous Media Modeling

In some types of electric machines, the stator winding coils are made up of bundles of strands of insulated copper wires and embedded in the slots of the stator core. At the

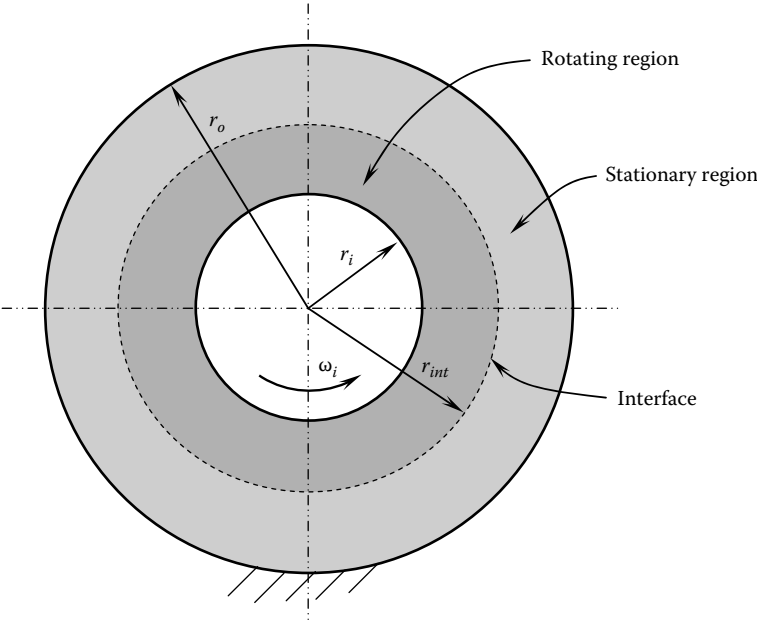


FIGURE 11.2
Defined rotating and stationary regions (as viewed from the stationary frame).

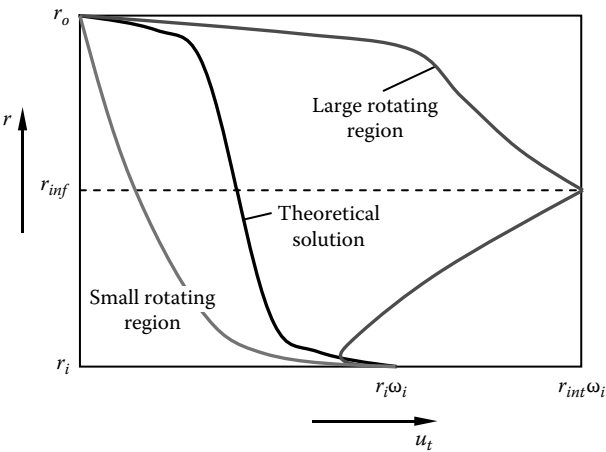


FIGURE 11.3
Comparison of theoretical solution and results from the model with multiple regions.

end-winding region, the stator coils bend to form continuous current paths. Without a resin encapsulation treatment, the stator winding contains a large number of voids in its complicated 3D structure (see Figure 4.35). Thus, to simplify the CFD analysis, the stator windings, especially at the end-winding regions, can be modeled as porous media [11.9,11.10].

Porous medium technology is defined as a utilization of specific and unique features of a highly porous medium for supporting and controlling the cooling process in electric machines. In practice, the flow velocity field and heat transportation in a 3D-structured porous medium are very complex. To overcome the difficulties, it is often required to treat the 3D structure of the porous media as a large number of hot spots homogeneously distributed throughout its volume. Moreover, the 3D model of the porous medium may be simplified into a 2D axisymmetric model.

In the porous media model, the flow passing through the stator winding with a fraction of the flow open area is equivalent to the flow passing through a porous medium with a full open area, with an identical mass flowrate and pressure drop:

$$\Delta p_a = \Delta p_p \quad (11.2)$$

$$\dot{m}_a = \dot{m}_p \quad (11.3)$$

where

Δp_a and Δp_p are pressure drops

$\dot{m}_a$ and $\dot{m}_p$ are mass flowrates from the actual and porous system, respectively

By defining the loss coefficient K ,

$$K = \frac{\Delta p}{(1/2)\rho u^2} \quad (11.4)$$

Δp_a and Δp_p can be rewritten as

$$\Delta p_a = K_a \left(\frac{1}{2} \rho u_a^2 \right) \quad (11.5)$$

$$\Delta p_p = K_p \left(\frac{1}{2} \rho u_p^2 \right) \quad (11.6)$$

Substituting Equations 11.5 and 11.6 into 11.2 yields

$$K_a \frac{\rho u_a^2}{2} = K_p \frac{\rho u_p^2}{2} \quad (11.7)$$

That is,

$$K_p = K_a \left(\frac{u_a}{u_p} \right)^2 \quad (11.8)$$

As the most important parameter in calculating porous media flows, porosity η is introduced and defined as the ratio of the actual open area to the total flow area (i.e., 100% open):

$$\eta = \frac{A_a}{A_p} \quad (11.9)$$

Rewriting Equation 11.3,

$$\rho A_a u_a = \rho A_p u_p \quad (11.10)$$

The relationship between the velocity ratio u_a/u_p and η can be determined as

$$\frac{u_a}{u_p} = \frac{A_p}{A_a} = \frac{1}{\eta} \quad (11.11)$$

Substituting (11.11) into (11.8), it yields

$$K_p = \frac{K_a}{\eta^2} \quad (11.12)$$

Thus, the porous media inertial resistance factor ξ , defined as the pressure loss factor per unit length, can be calculated as

$$\xi = \frac{K_p}{\Delta L} \quad (11.13)$$

where ΔL is the length through the media in the flow direction.

Thus, with the equations mentioned previously, the flow characteristics can be obtained at the porous flow zone. By integrating the porous media parameters into the CFD model, the velocity and pressure fields at the end-winding region are determined.

11.1.4 Numerical Simulation of Motor Cooling

Modern electric motors have more complicated mechanical and electromagnetic structures over conventional motors for gaining better performance, higher operation reliability and efficiency, and longer lifetime. For such systems with complicated geometry, conventional analytic methods are no longer competent to provide comprehensive information of fluid flows and temperature distributions of motor in a system level. Therefore, CFD techniques have been extensively employed by motor manufacturers to help optimize motor cooling systems.

As in the example shown in Figure 11.4, a large-scale elevator motor is cooled by high volumetric air driven by a blower. Unlike most conventional motors whose rotors are on the inside of wound stators, this external rotor motor has its rotor on the motor outside to directly drive the elevator car. The advantages of the external rotor motors include increased motor torque, reduced power losses, and promoted transmission efficiency [11.11]. This type of motor is especially suitable for applications such as fans, motor-wheel driving systems [11.12,11.13], and lifting systems of forklift trucks.

Due to the symmetry of the model, only one-tenth of the motor (36°) is modeled. In order to reduce the required computing resources and accelerate the computations, the model has been simplified and defeatured without loss in accuracy. As shown in Figure 11.5, the cooling air flow enters into the stationary hollow shaft at its one end and injects into the radial cooling channels located at the central and end-winding regions. Then, the cooling air flow turns to 90° when it reaches the air gap and finally exhausts from the two endbells to the environment.

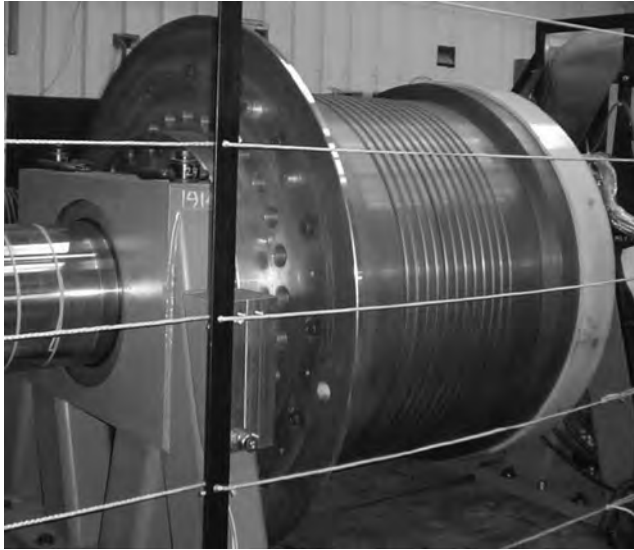


FIGURE 11.4
Large-scale elevator motor. (Courtesy of Kollmorgen Corporation, Radford, VA.)

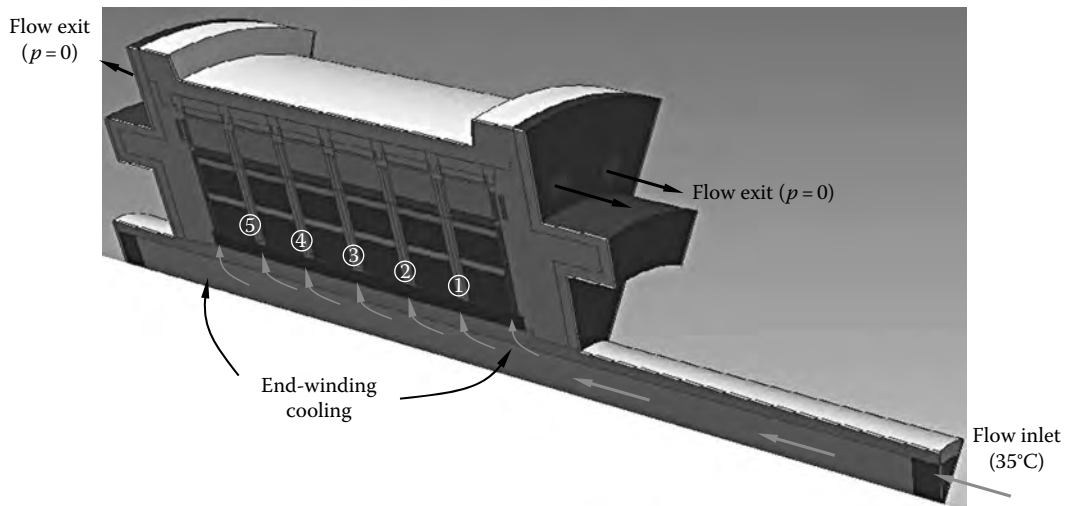


FIGURE 11.5
Forced air cooling for a large-scale elevator motor. (Courtesy of Kollmorgen Corporation.)

11.1.4.1 Mathematical Formulations

In a CFD analysis, governing equations are derived from the following fundamental laws of physics:

- Conservation of mass
- Conservation of momentum
- Conservation of energy

A time-dependent continuity equation is derived from the conservation of mass. Navier–Stokes nonlinear partial differential equations arise from applying Newton’s second law to the motion of fluid. These equations consist of three time-dependent momentum equations corresponding to three coordinates. If a problem involves heat transfer, an additional time-dependent energy equation, based on the conservation of energy, has to be coupled with Navier–Stokes equations. All equations must be solved simultaneously for getting the entire flow, pressure, and temperature fields of electric motor.

The governing equation may be used with a relatively high degree of accuracy for incompressible flows if the viscosity gradient is not too large. In majority of the problems, this assumption is adequate. The temperature-dependent properties are often used in CFD analyses.

- Continuity equation

Continuity equation is based on the conservation of mass in a fixed control volume of fluid flow and can be expressed in vector notation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{V}) = 0 \quad (11.14)$$

Equation 11.14 is the continuity equation in conservation form. In this equation, the second term can be separated into two parts as priority

$$\frac{\partial \rho}{\partial t} + \mathbf{V} \cdot \nabla \rho + \rho \nabla \cdot \mathbf{V} = 0 \quad (11.15)$$

By introducing the differential operator [11.14]

$$\frac{D(\)}{Dt} = \frac{\partial(\)}{\partial t} + \mathbf{V} \cdot \nabla(\) \quad (11.16)$$

Equation 11.15 can be rewritten in nonconservation form:

$$\frac{D\rho}{Dt} + \rho \nabla \cdot \mathbf{V} = 0 \quad (11.17)$$

- Momentum equations

Momentum equations are derived from the conservation of momentum, comprising three components, that is, x , y , and z momentum equations in a Cartesian coordinate system or r , θ , and z in a cylindrical system, in a general form of fluid motion equations:

$$\rho \frac{D\mathbf{V}}{Dt} = -\nabla p + \nabla \cdot \bar{\bar{\tau}} + \rho \mathbf{f} \quad (11.18)$$

where

$\mathbf{f}$ is the body force per unit mass

$\bar{\bar{\tau}}$ is the viscous stress tensor

For Newtonian fluids, $\bar{\tau}$ is a linear function of the velocity gradient and can be expressed as

$$\tau_{ij} = \mu \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) + \lambda (\nabla \cdot \mathbf{V}) \delta_{ij} \quad (11.19)$$

where

μ is the dynamic viscosity

λ is the coefficient of bulk viscosity

x_i and x_j denote mutually perpendicular coordinate directions

The coefficient of bulk viscosity and dynamic viscosity can be related to each other through the Stokes' hypothesis:

$$\lambda + \frac{2}{3}\mu = 0 \quad (11.20)$$

Substituting Equation 11.20 into 11.19 yields

$$\tau_{ij} = \mu \left[\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} - \frac{2}{3}(\nabla \cdot \mathbf{V}) \delta_{ij} \right] \quad (11.21)$$

where δ_{ij} is the Kronecker delta operator, which is equal to 1 if $i=j$ and it is zero otherwise.

- Energy equation

Energy equation is based on conservation of energy and expressed as

$$\rho \left[\frac{\partial h}{\partial t} + \nabla \cdot (h\mathbf{V}) \right] = -\frac{\partial p}{\partial t} + (\mathbf{V} \cdot \nabla p) + \nabla \cdot (k\nabla T) + \phi \quad (11.22)$$

where

h is the specific enthalpy that is related to specific internal energy as $h = e + p/\rho$

T is the temperature

ϕ is the dissipation function representing the work done against viscous forces and is expressed as

$$\phi = (\bar{\tau} \cdot \nabla) \mathbf{V} = \tau_{ij} \frac{\partial V_i}{\partial x_j} \quad (11.23)$$

For an incompressible fluid, the energy equation can be also expressed as

$$\rho c_p \left[\frac{\partial T}{\partial t} + \mathbf{V} \cdot \nabla T \right] = \nabla \cdot (k_{eff} \nabla T) + S_h \quad (11.24)$$

where

T is the temperature

k_{eff} is the effective thermal conductivity ($k_{eff} = k + k_t$, where k_t is the turbulent thermal conductivity)

S_h is the volumetric heat source

- Turbulence model

A majority of fluid flows encountered in motor cooling are turbulent flows. There are a number of turbulence models in CFD analysis to deal with turbulent flows, such as constant eddy viscosity, standard k - ϵ , Reynolds stress, and renormalization group (RNG) k - ϵ model, just to name a few. This study employs the RNG k - ϵ turbulence model, which was derived using a rigorous statistical technique:

k equation

$$\nabla(\rho k \mathbf{V}) = \nabla(\alpha_k \mu_{eff} \nabla k) + \mu_t \phi - \rho \epsilon \quad (11.25)$$

ϵ equation

$$\nabla(\rho \epsilon \mathbf{V}) = \nabla(\alpha_\epsilon \mu_{eff} \nabla \epsilon) + C_{1\epsilon} \mu_t \phi \frac{\epsilon}{k} - C_{2\epsilon} \rho \frac{\epsilon^2}{k} - R \quad (11.26)$$

where

$\mu_t \phi$ represents the generation of turbulent kinetic energy

$C_{1\epsilon}$ and $C_{2\epsilon}$ are constants ($C_{1\epsilon} = 1.44$ and $C_{2\epsilon} = 1.92$)

α_k and α_ϵ are the inverse effective Prandtl numbers for k and ϵ , respectively

In the aforementioned equations, the effect of buoyancy is ignored.

Because turbulence always causes mixing between the fluid layers, it results in diffusion, which is treated as an increase in viscosity. As a result, μ_{eff} is introduced as the effective viscosity:

$$\mu_{eff} = \mu + \mu_t = \mu + \rho C_\mu \frac{k^2}{\epsilon} \quad (11.27)$$

where $C_\mu = 0.0845$.

The R term in the ϵ equation is given by

$$R = \frac{C_\mu \rho \eta^3 (1 - \eta/\eta_o)}{1 + \beta \eta^3} \frac{\epsilon^2}{k} \quad (11.28)$$

where

$$\eta = Sk/\epsilon$$

$$\eta_o = 4.38$$

$$\beta = 0.012$$

11.1.4.2 Numerical Method

The problem approaches include setting up boundary and initial conditions and making necessary assumptions, including (1) assigning thermal physical properties to the fluid and solid components; (2) defining various power losses, which are obtained from a separate electromagnetic analysis, to the corresponding motor components as the heat sources; (3) applying the inlet boundary conditions such as volumetric flowrate (or velocity) and temperature; (4) applying the outlet boundary condition with zero static pressure; (5) assuming all outer walls as adiabatic and heat dissipation only through the cooling fluid; and (6) specifying the initial temperature to the computational domain.

The governing equations are solved on the discretized elements using commercial CFD software. All walls are treated as adiabatic and no slip. The flowrate is defined at the flow inlet, and zero pressure is specified as outlet.

The aims of performing the CFD analysis of the motor cooling include the following: (1) The main objective is to obtain the temperature distribution in the motor interior, especially around the stator windings, to avoid motor overheating. (2) The cooling flowrates and their distributions are important for the assessment of motor fan selection and performance. Insufficient cooling flows can result in hot spots occurring near the heat sources such as stator winding. Through the CFD analysis, the required air flowrate can be determined. (3) The 3D velocity field of the motor reveals fluid–solid interactions and flow patterns such as flow separation and attachment to solid body surfaces. A separated air flow may cause an increase in skin friction and impact on cooling efficiency. (4) The pressure drop data are used to evaluating motor windage losses. Attention should be focused on abnormal pressure drops at local structures for further minimizing windage losses.

The overall quality of meshes is of primary importance in predicting the temperature and velocity fields in this complex computational domain. In order to obtain accurate results, very fine meshes are set at the high-pressure gradient regions. Due to the large motor size and complicated geometries, the CFD model is meshed with more than 4 million elements: 2.27 million elements for the fluid and 2.07 million elements for the solid parts (Figure 11.6).

In order to achieve accurate numerical simulation results, the winding insulation is also taken into account in this analysis (Figure 11.7). Because the winding insulation is too thin to be meshed with a conventional method, it must be dealt with special approaches. In practice, such thin-walled objects such as shells are defined as surface parts. One effective way to deal with the meshing of thin-walled objects is to represent them only with surfaces and eliminate the thickness form of the model. In other words, the surface parts are meshed with 2D elements and are used to represent thin-walled objects. In such a way, surface parts can be used to conduct heat as well as obstruct flow. They exhibit the same heat transfer characteristics as 3D volumes.

To ensure proper convergence, residuals for mass, momentum, turbulent kinetic energy, and temperature were monitored until achieving minimum level of 10^{-4} . Furthermore,

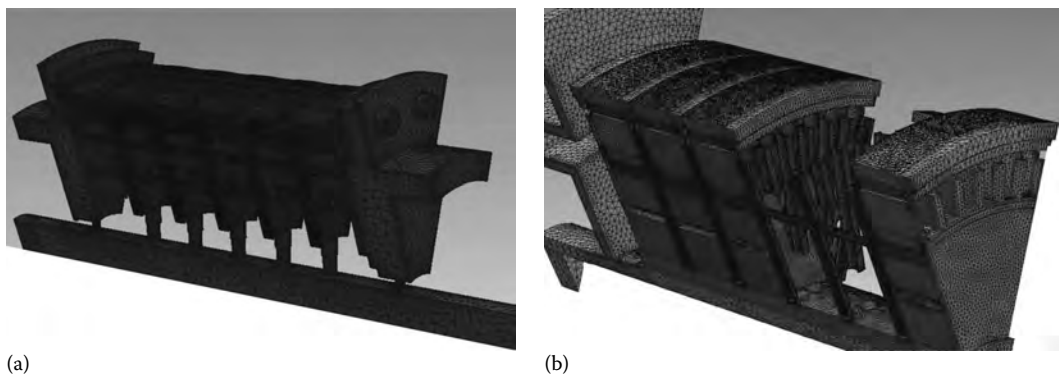
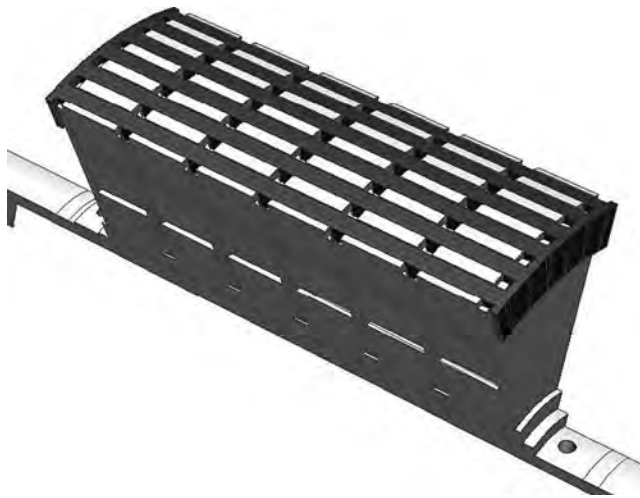


FIGURE 11.6

Model meshing: (a) fluid meshing (2.27 million elements) and (b) meshing for solid parts (2.07 million elements).

**FIGURE 11.7**

The thin insulation layer of stator winding is taken into account in the analytical model.

various monitoring planes at different sections of the motor fan assembly were created and monitored to achieve stable convergence for mass flow and total pressure without any undue fluctuations. In order to determine the required fan flowrate, a number of volumetric flowrates are used in this CFD analysis, including 300, 500, 700, 900, and 1200 CFM. All power losses, such as copper, core, magnet, and eddy current, are defined at the corresponding components.

To help achieve convergence, the analysis starts with pure fluid dynamics. As the flow field becomes fully developed, the heat transfer model is added to calculate the temperature distribution in the computational domain.

11.1.4.3 CFD Results

The velocity field of the motor is presented in Figure 11.8 for a flowrate of 1200 CFM. In order to clearly demonstrate the flow field, only the fluid domain is shown. High velocities occur at the radial spray nozzles on the hollow shaft. The flowrate distributions through five middle channels and two end-winding channels are displayed unevenly; the flowrate is inversely proportional to the distance from the flow inlet. This suggests that for receiving a uniform flowrate distribution, different nozzle diameters are required.

Figure 11.9 displays the velocity fields of middle cooling channels. The flow pattern at each channel is directly associated with the cooling of motor components. For example, at channel 1, the air cooling flow goes through the middle and left subchannels, and almost no flow goes through the right subchannel, resulting in a higher temperature (about 5°C) on the stator winding on the right subchannel.

The static pressure distribution inside the motor is given in Figure 11.10. It can be found that the highest pressure occurs near the nozzle where the dynamic pressure is converted into the static pressure. The pressure drop across the motor is a function of the flowrate, air viscosity, flow path geometries (e.g., air gap depth and nozzle diameter), and rotor rotating speed. The air flow travels through the nozzles, channels, air gap, and exhaust holes,

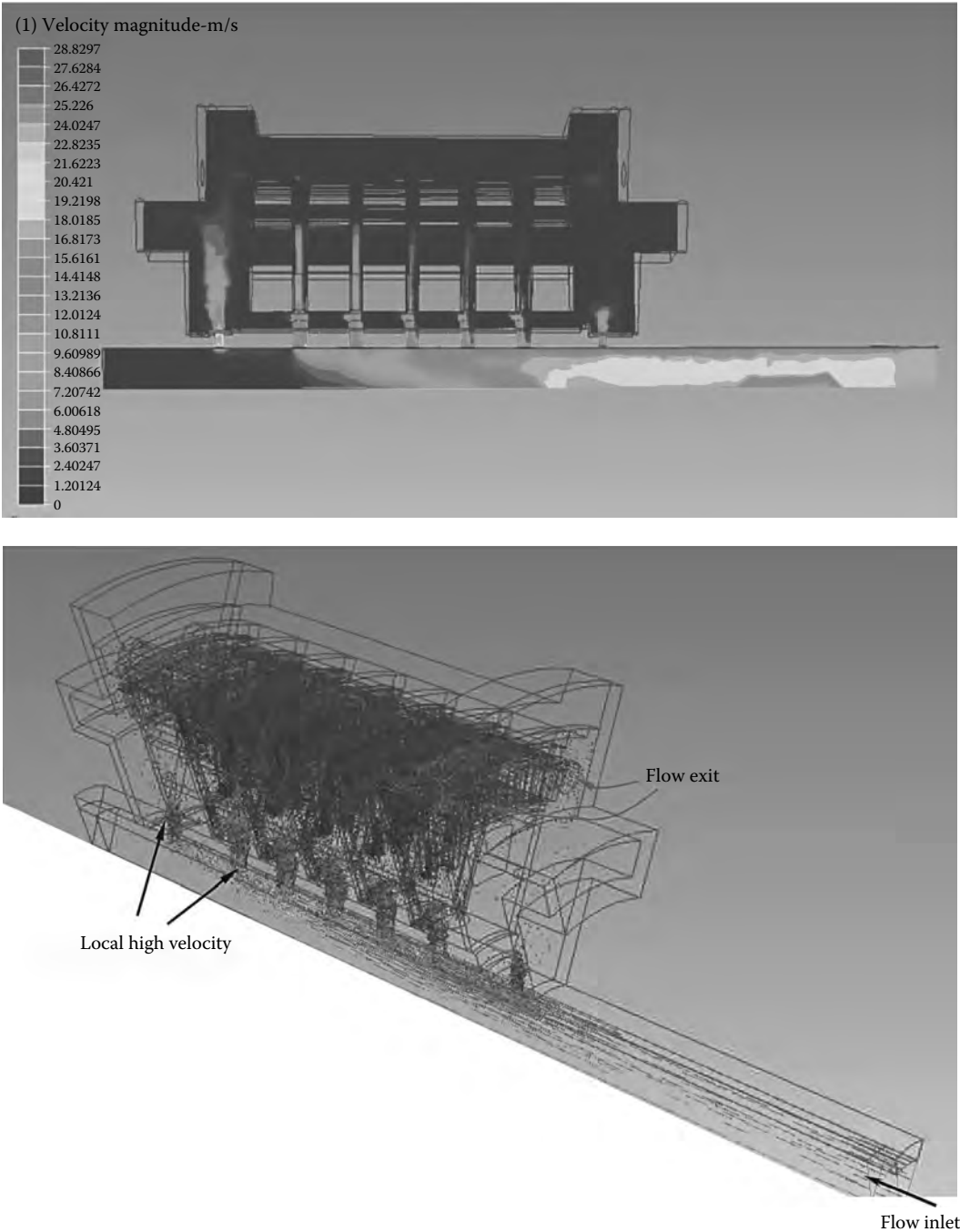


FIGURE 11.8
Velocity field through the motor interior with an intel flowrate of 1200 CFM.

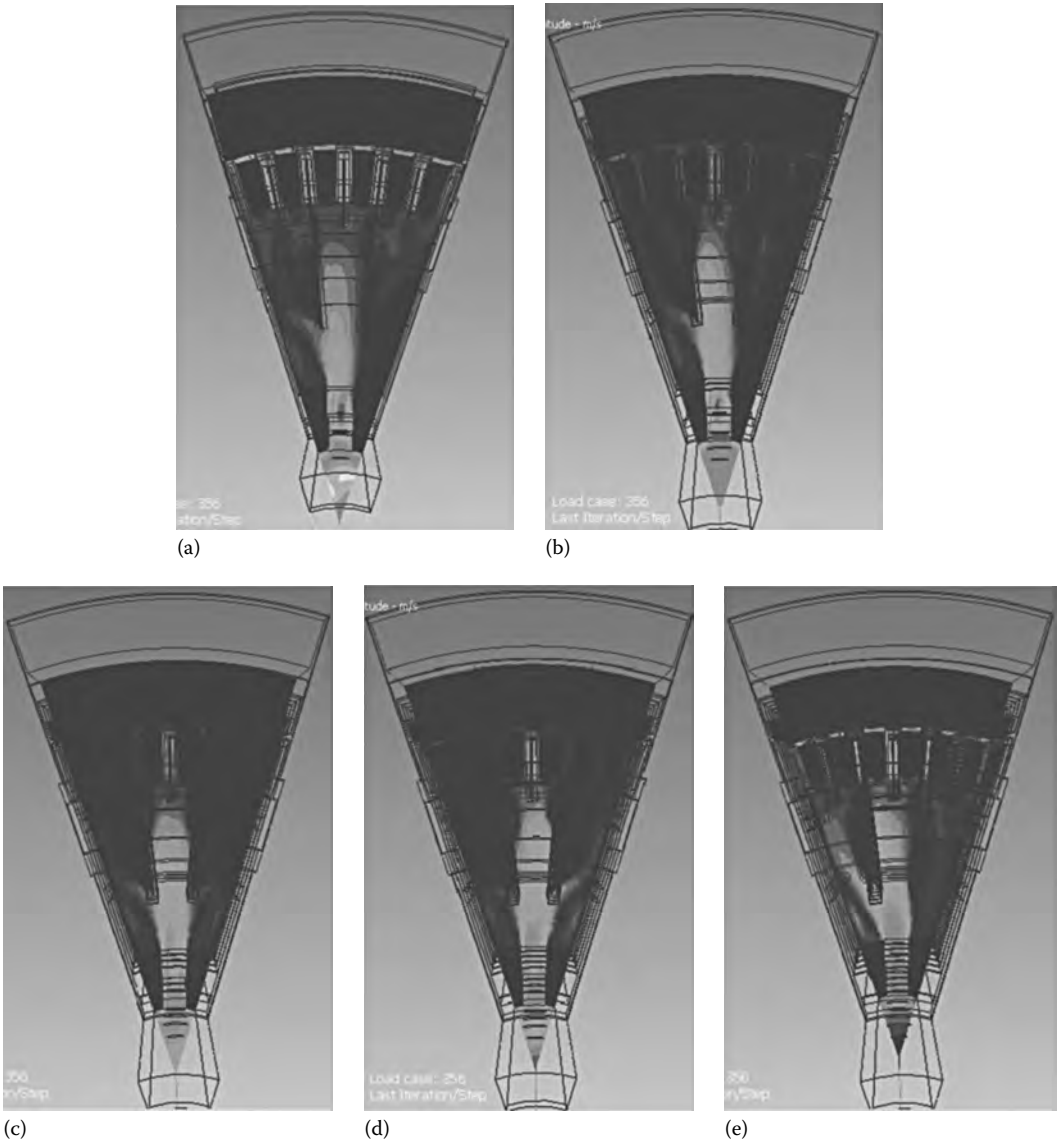


FIGURE 11.9 Velocity contours through the middle cooling channels: (a) channel 1, (b) channel 2, (c) channel 3, (d) channel 4, and (e) channel 5.

creating a pressure drop due to the restriction of flow. Based on the pressure contours in the figure, the air flow passages have been optimized for minimizing the pressure drop and reducing the windage power loss.

The temperature distribution in the motor is shown in Figure 11.11. The highest temperature for the flowrate of 1200 CFM is 93.7°C, occurring on the stator end turn near the flow inlet. This is consistent with the observation that the flowrate is inversely proportional to the distance from the flow inlet. From the calculated results, it is estimated that the temperature difference due to the nonuniform flowrate is about 10°C on the stator

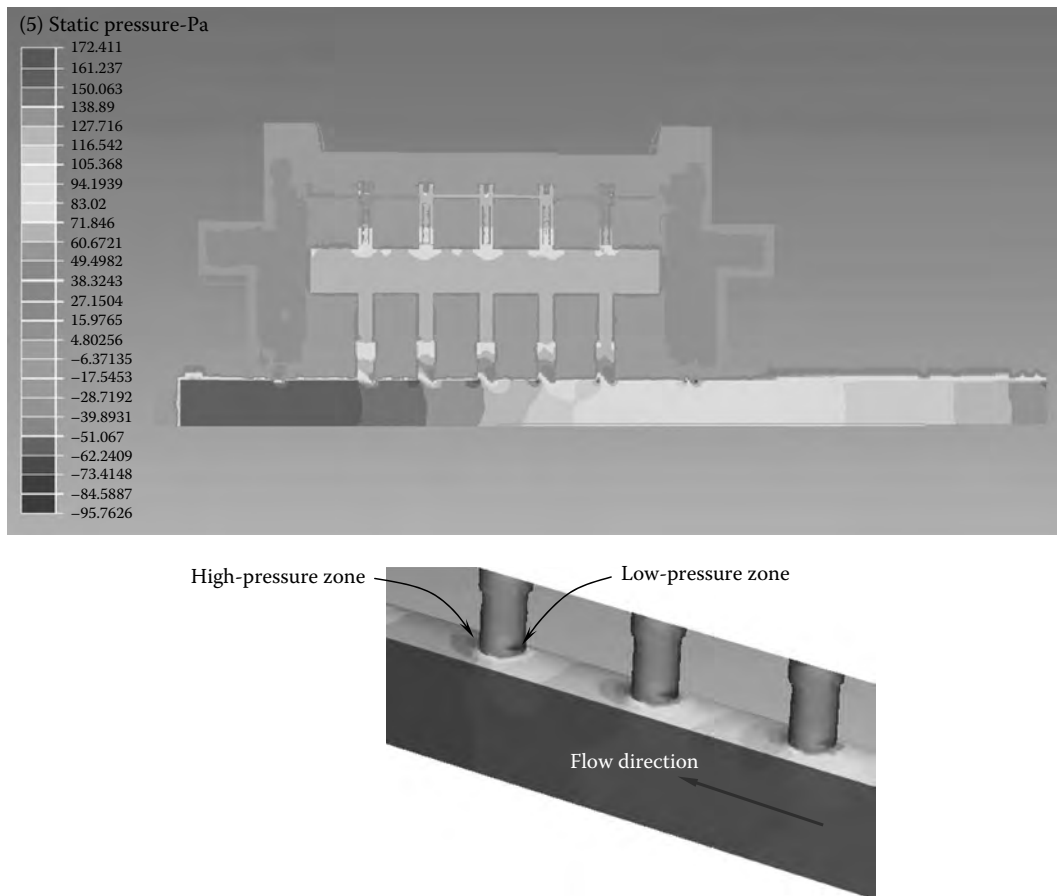


FIGURE 11.10
Pressure distribution through the motor.

winding. The temperature of magnets is shown to be 66°C . Usually, rare-earth magnets such as Nd-Fe-B are sensitive to heat. If a magnet is heated above its maximum operating temperature, it will permanently lose a fraction of its magnetic strength. If the magnet is heated about its Curie temperature, it will lose all of its magnetic properties. Therefore, it is important to always check the magnet temperatures.

Figure 11.12 shows in detail the temperature distribution in each cooling channel. It can be observed that the temperature difference on the stator winding in the same channel is usually less than 5°C .

The maximum temperature of 93.7°C indicates that the air flowrate can be further reduced, lowering not only the power consumption of the blower but also windage loss of the motor. Figure 11.13a and b presents the temperature distributions for the flowrate of 700 and 900 CFM, respectively. For the flowrate of 900 CFM, the maximum temperature is increased to 109°C (-16.3%). Continuously reducing the flowrate to 700 CFM, the maximum temperature becomes 121.7°C , which can be used for the class B insulation that has the maximum operation temperature of 130°C . For the class F insulation (maximum operation temperature of 155°C), the flowrate can be further reduced.

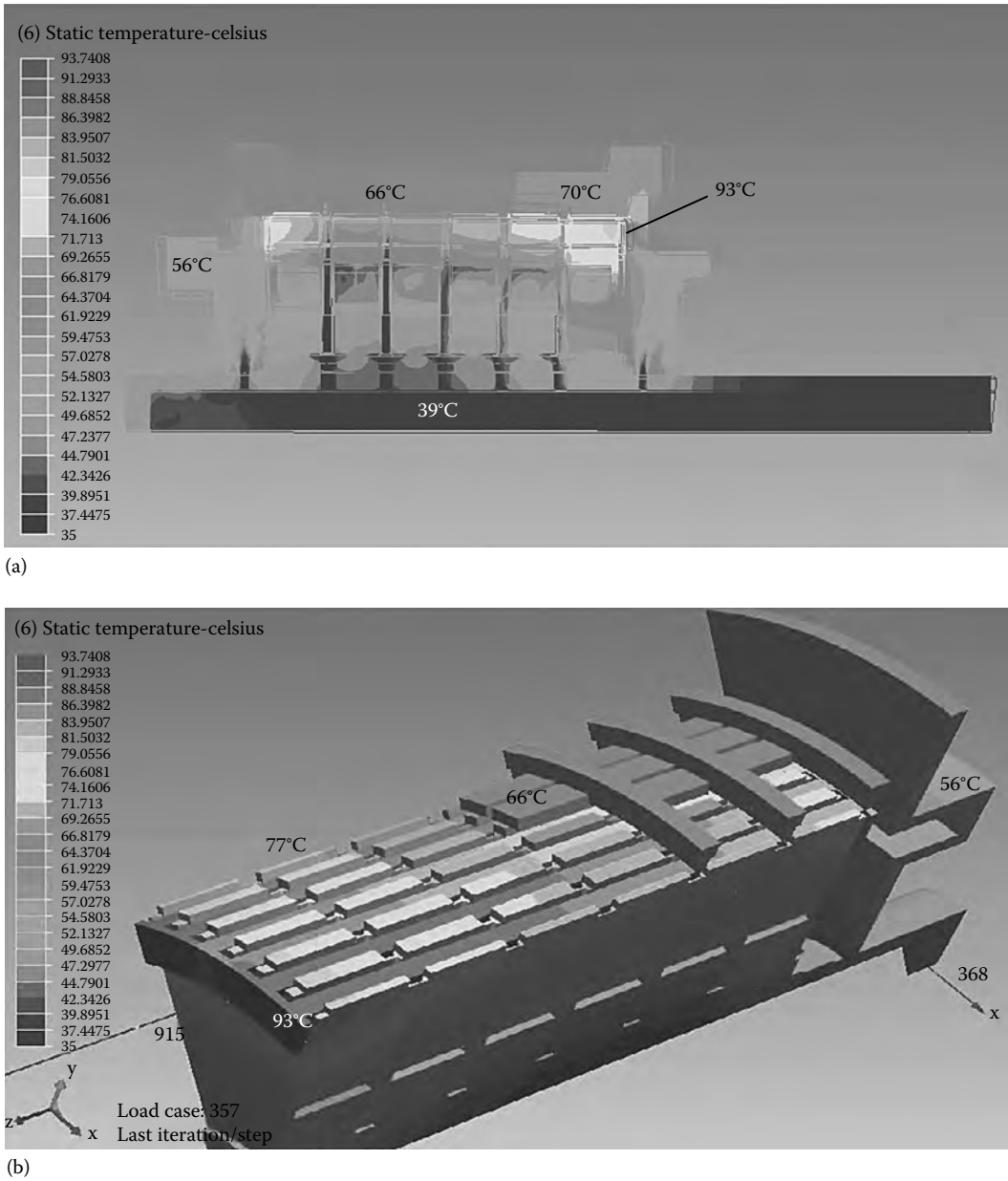


FIGURE 11.11 Temperature distribution on (a) the whole motor and (b) solid components only. The flowrate of 1200 CFM is used for the analysis.

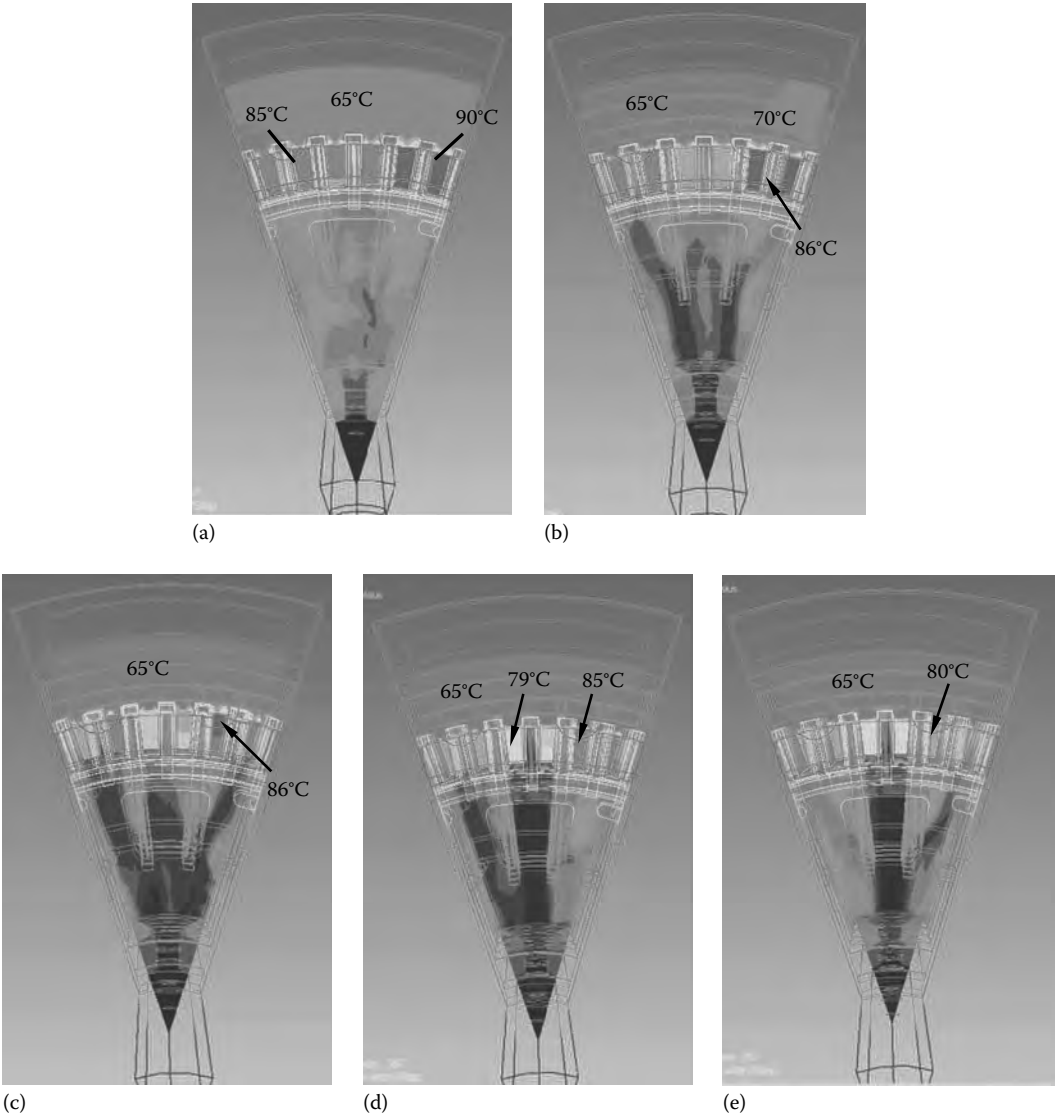


FIGURE 11.12 Temperature distributions through the cross-sectional cooling channels: (a) channel 1; (b) channel 2; (c) channel 3; (d) channel 4; and (e) channel 5.

Performing the CFD analysis with the flowrates of 300–1200 CFM (0.14–0.57 m³/s) can provide a useful relationship between the maximum temperature and flowrate, as displayed in Figure 11.14. The results indicate that the appropriate range of the volumetric flowrate for the motor cooling is 700–900 CFM (0.33–0.42 m³/s), with the corresponding temperatures in the range of 121°C–109°C.

Figure 11.15 presents the relationship between the flowrate and pressure at blower. This information helps select an appropriate blower/fan and determine the optimum operating points of the blower/fan.

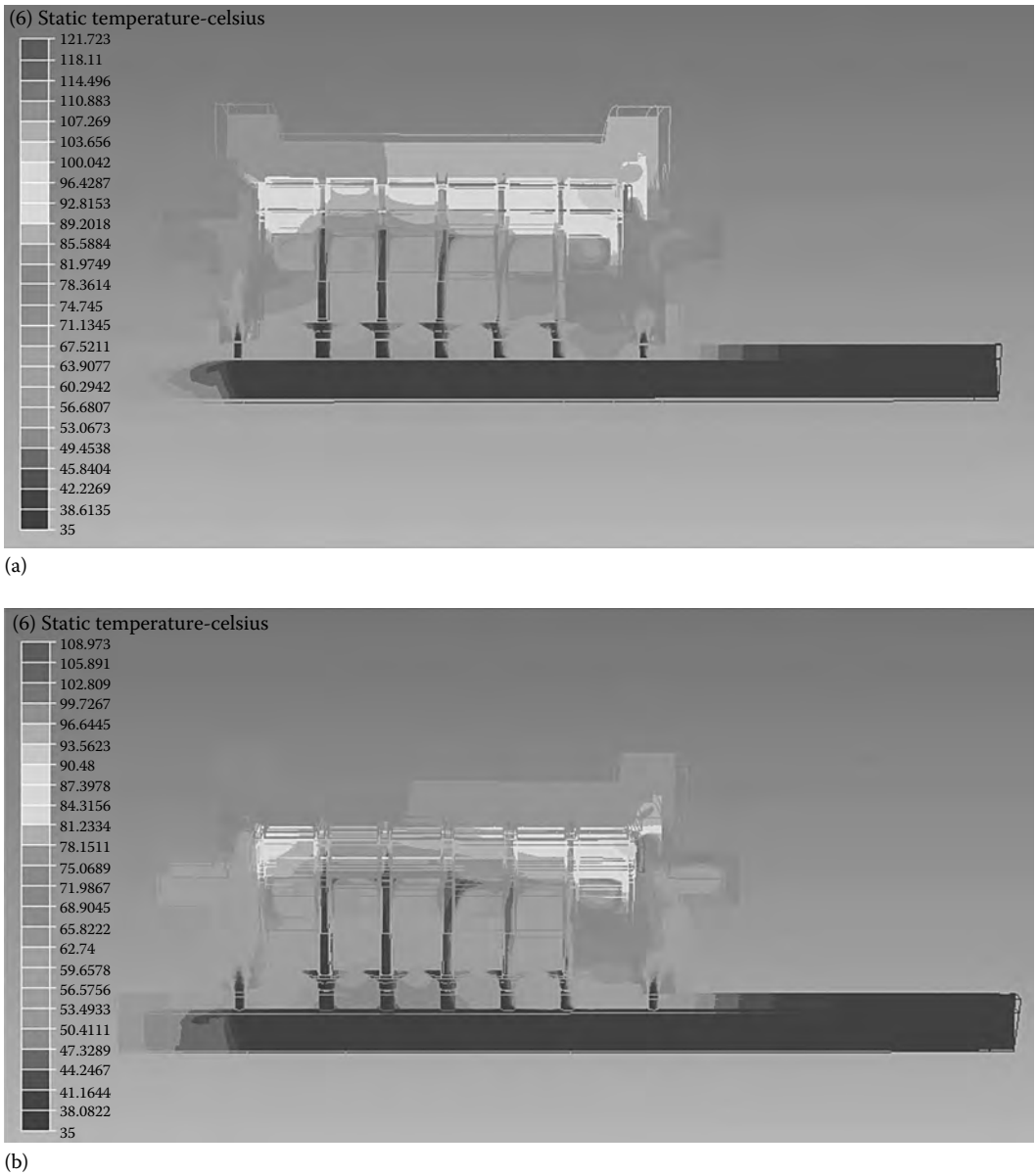


FIGURE 11.13
Temperature distributions inside the motor: (a) flowrate of 700 CFM and (b) flowrate of 900 CFM.

11.2 Thermal Simulation with Lumped-Circuit Modeling

Among various thermal analysis methods, a simple and common approach is to use lumped-circuit (or network) analogy modeling. This method is based on the analogy between electrical and thermal systems. The analogies between current I and rate of heat

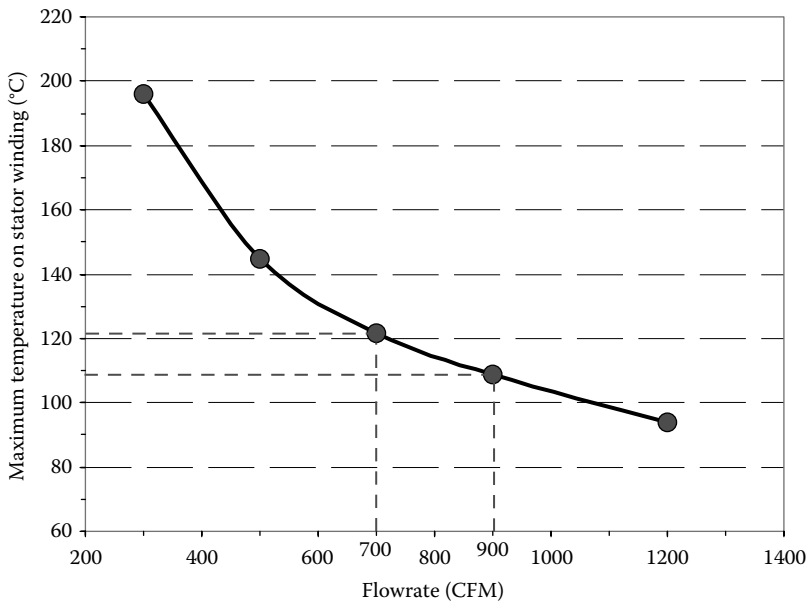


FIGURE 11.14 Maximum temperature versus flowrate. The results indicate that the required volumetric flowrate for the motor cooling is in the range of 700–900 CFM. Correspondingly, the motor maximum temperature is in the range of 109°C–122°C.

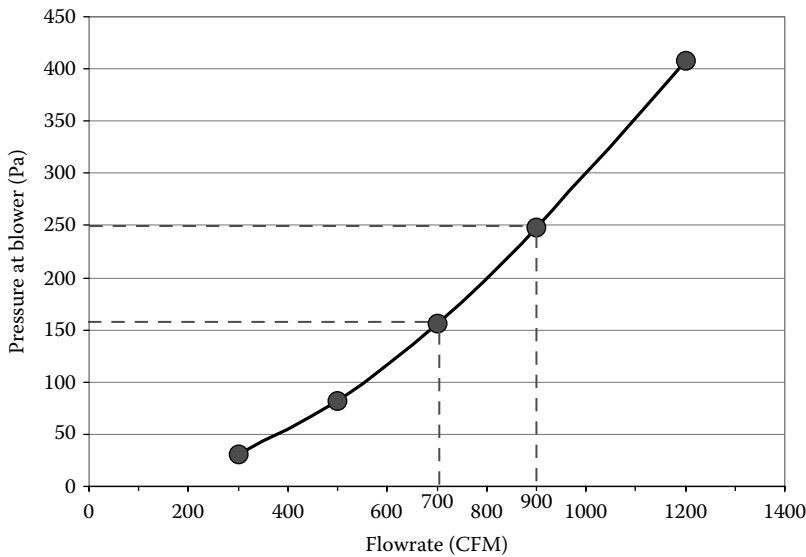


FIGURE 11.15 Blower pressure versus flowrate. The results indicate that the required volumetric flowrate for the motor cooling is in the range of 700–900 CFM.

TABLE 11.1

Analogous Parameters between Electrical and Thermal Systems

Parameter	Electrical System	Thermal System
Flow	Current I (in Ampere [A])	Heat flow q (in W)
Potential	Voltage V (in V)	Temperature differential ΔT (in °C or K)
Resistance	$R = \frac{V}{I}$ (in Ω)	$R_{th} = \frac{\Delta T}{q}$ (in °C/W)
Conductance	$S = 1/R$ (in A/V)	$G = 1/R_{th}$ (in W/°C)
Capacitance	$C = \frac{I}{dV/dt}$ (in Farad [F])	$C_{th} = \frac{q}{dT/dt}$ (in J/°C)
Ohm's law	$I = \frac{V}{R}$	$q = \frac{\Delta T}{R_{th}}$

flow q , voltage V and temperature difference between two spatial points or surfaces ΔT , etc., can be properly applied in analyzing thermal systems. In such a way, it is possible to take thermal resistances as the analog to electrical resistances. This allows thermal engineers to utilize fully developed basic laws (such as Ohm's law and Kirchhoff's law) in electrical engineering to determine heat transfer characteristics in thermal systems. The analogous parameters between the two systems are listed in Table 11.1.

In fact, this conventional approach was popular before the advanced computational techniques and CFD software have been fully developed. The lumped-circuit approach has the advantage of being very fast to calculate, but the development of the network model is time-consuming, requiring thermal engineers to spend a large amount of time in defining the circuit that accurately models all heat transfer paths and circuit elements.

By converting a complex 3D physical model into an equivalent lumped-circuit system, it greatly simplifies the problem complexity and thus reduces the simulation runtime and the memory usage. The lumped-circuit analogy method involves representation of the thermal and flow systems as a network of thermal and flow paths and components for the prediction of system-wide temperature distribution and flow field. In fact, this technique is very efficient in terms of the effort required for model definition, solution, and examination of results because it employs overall component characteristics for analyzing their systems-wide interaction. Because this technique requires much shorter time compared with CFD analysis, it is especially suitable to explore different designs at the conceptual design stage [11.15]. However, the most difficult aspect of this approach is to determine the heat transfer coefficients and the resultant thermal resistances in convective heat transfer, as well as in radiation heat transfer.

An analytical-numerical hybrid scheme for system-wide thermal modeling of electric motors has been proposed by Liu et al. [11.16]. This scheme employs finite volume method (FVM) concept to calculate heat conduction for motor components while using flow network modeling (FNM) for fluid convection calculation. This hybrid scheme not only simplifies the complicated flow simulation but also considers motor's geometry information and increases the accuracy of the simulation.

In order to develop an equivalent thermal network, it is required to subdivide the thermal system into a number of finite subvolumes called nodes. All the thermal parameters, such as temperature, thermal capacitance, and heat generation, are considered to be concentrated at the central nodal point. This point represents the mean values of the thermal

parameters in the node. The thermal models for the basic elements of electric machines have been discussed by Perez and Kassakian [11.17] in more detail.

An extended survey on the evolution and the modern approaches in thermal analysis of electrical machine has revealed that it follows the path from the lumped-parameter network, to FEA, and to today's numerical simulations using CFD techniques. It can be advantageous to use the lumped-parameter network approach for its simplicity, requirement of less computing resources, and fast calculations. FEA can be considered a convenient solution in a very complex geometry not approachable with lumped-parameter network. CFD can deal with quite complex heat transfer problems and provide very accurate results. However, it needs very knowledgeable and experienced thermal engineers to correctly set up the model and requires very high computer capabilities and powerful computing resources. As a matter of fact, each of these approaches has its advantages and disadvantages.

A comprehensive review has been made by Boglietti et al. [11.18] on the evolution and modern approaches in the thermal analysis of electrical machines. The three primary thermal analysis methods, that is, lumped-parameter thermal network, FEA, and CFD, are analyzed in depth and compared in order to highlight the qualities and defects of each. The thermal network is the most basic form to calculate conduction, convection, and radiation resistances for different parts of the motor construction. The convection heat transfer coefficient is most often based on empirical convection correlations. This is fundamentally different from CFD analysis, where the heat transfer coefficient is calculated from the CFD model itself. According to Boglietti et al., FEA can only be used to model conduction heat transfer in solid components. They expect that CFD will be more popular and widespread in thermal analysis due to the fast development in computational capability of modern computers, as well as the cost reduction of CFD software.

11.3 Thermal Analysis Using Finite Element Method

FEM has been one of the major numerical solution techniques. One of the major advantages of the FEM approach is the simplicity and ease for using FEM to solve complex geometry problems. By comparing with finite difference method (FDM) and FVM, FEM is superior in its built-in ability to handle unstructured meshes, a rich family of element choices, and natural handling of boundary conditions. As indicated by its name, FEM requires division of the solution domain into many discrete volumes or finite elements. Thus, FEM yields discretized equations that are entirely local to the elements and provides complete geometric flexibility [11.19]. Applications of FEM become more widespread for motor design engineers to analyze structural, thermal, and fluid dynamic problems and their interactions. However, it is difficult to develop computationally efficient solution methods for strongly coupled and nonequations using FEA [11.20]. In addition, FEM does not help in determining quantities such as convective heat transfer coefficients. Thus, FEM is still not comparable to professional commercial CFD codes in solving convective heat transfer and fluid dynamic problems so far.

One example of employing FEM to carry out a thermal analysis is shown in Figure 11.16. This is a frameless motor with two endbells positioned at the two sides of the stator core. The endbells are made from gray cast iron and ductile cast iron for achieving high-vibration damping and high-strength properties. The stator winding is encapsulated by epoxy resin to eliminate the voids inside the winding, improve heat transfer, and increase

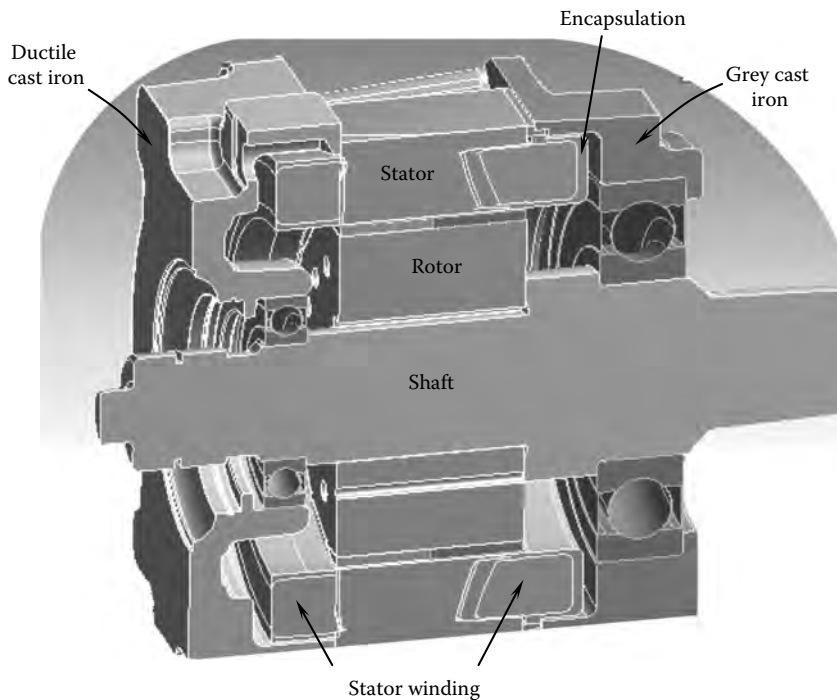


FIGURE 11.16
FEM model of an electric motor.

the winding's dynamic stiffness. The physical motor model is generated by a CAD tool and transferred into the FEM model.

This FEM analysis deals primarily with heat conduction through the motor components. The heat transfer coefficient h on the outer surfaces of the motor is obtained from the heat transfer correlations available in the literature for addressing the heat dissipation from the motor to the ambience (Figure 11.17). It shows for both horizontal and vertical surfaces, heat transfer coefficients increase exponentially with the increase in temperature. Due to the symmetry of the model, only a half of the motor is modeled. In addition to heat conduction (inside the motor) and convection (on the motor outer surfaces), the heat radiation effect is also integrated into the model. The initial temperature of the motor is set at 45°C. The thermal conductivities of copper, epoxy E88, and cast iron are functions of temperature (Figure 11.18). As can be seen from the figure, for the temperature less than 200°C, the thermal conductivity of copper decreases sharply and then becomes a linear function of temperature for $T > 200^\circ\text{C}$. By comparison, the temperature effect on k for epoxy is rather weak; when the temperature increases from 100°C to 500°C, the thermal conductivity reduces only 8%. In addition, the increase in the silicon content of the silicon iron lamination can lead to decrease in the thermal conductivity and increase in eddy-current losses [11.21].

The power losses from the stator winding, rotor and stator cores, bearings, and others are properly defined to the corresponding components as the heat sources.

Figure 11.19 shows the meshes created for the FEM model. Relatively fine meshes are used in the vicinities of stress-concentrated regions such as sharp contact areas, entrant corners, load transfer (welds, bonded joints, reinforcing bars, etc.), abrupt changes in thickness, material properties, and cross-sectional areas. It is critical to pay attention to

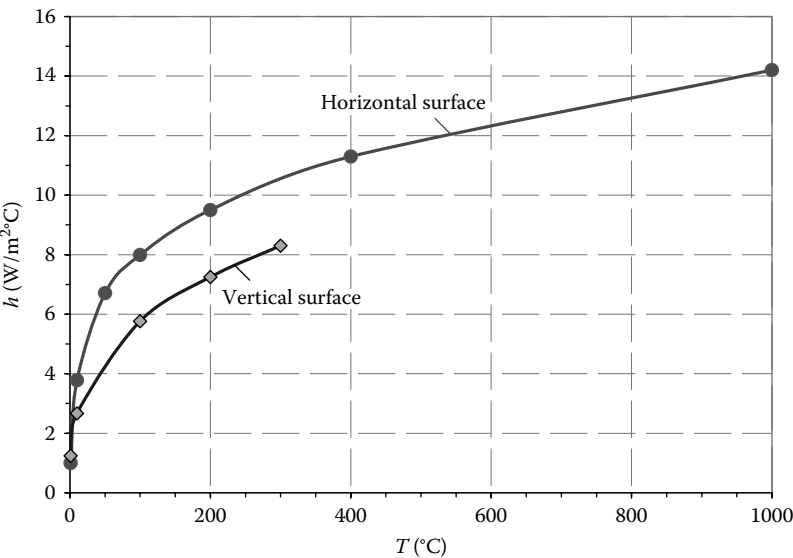


FIGURE 11.17 Heat transfer coefficients on horizontal and vertical surfaces of the motor, obtained from the thermal correlations in the literature.

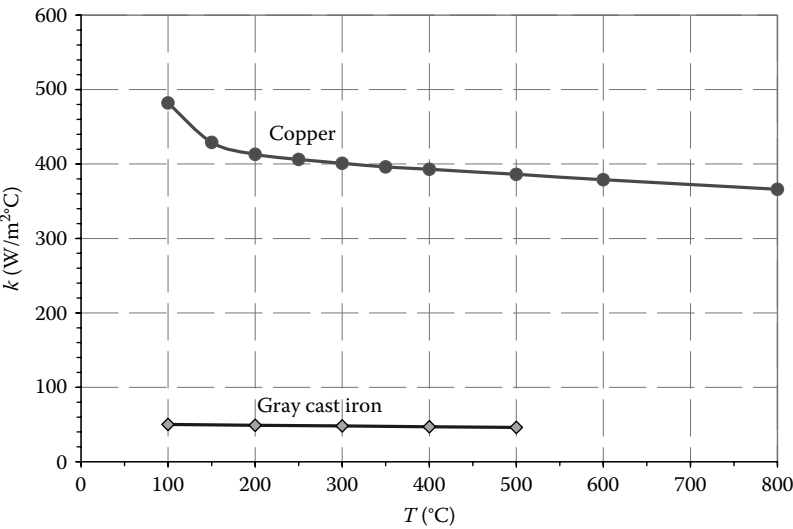


FIGURE 11.18 Thermal conductivities of copper and gray cast iron as functions of temperature.

the connections between adjacent parts and set up the same mesh pattern on them. In the mesh setting, fast transitions from small elements to large elements should always be avoided.

The calculated temperature distribution in the motor is presented in Figure 11.20. It is noted that the temperature ranges from 92.4°C to 112.7°C , with the maximum temperature occurring on the stator winding. In the rotor, temperature is quite uniform, about 95°C .

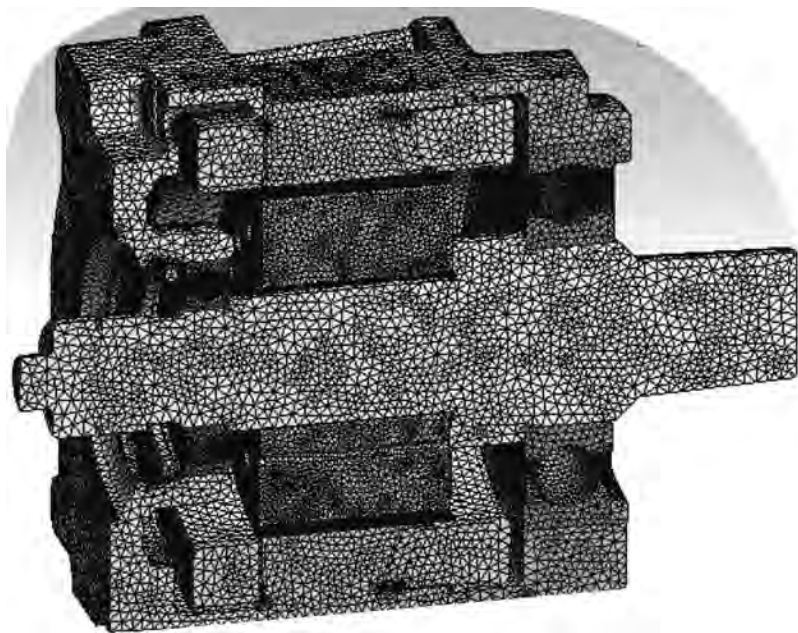


FIGURE 11.19
Meshes on the motor using FEM.

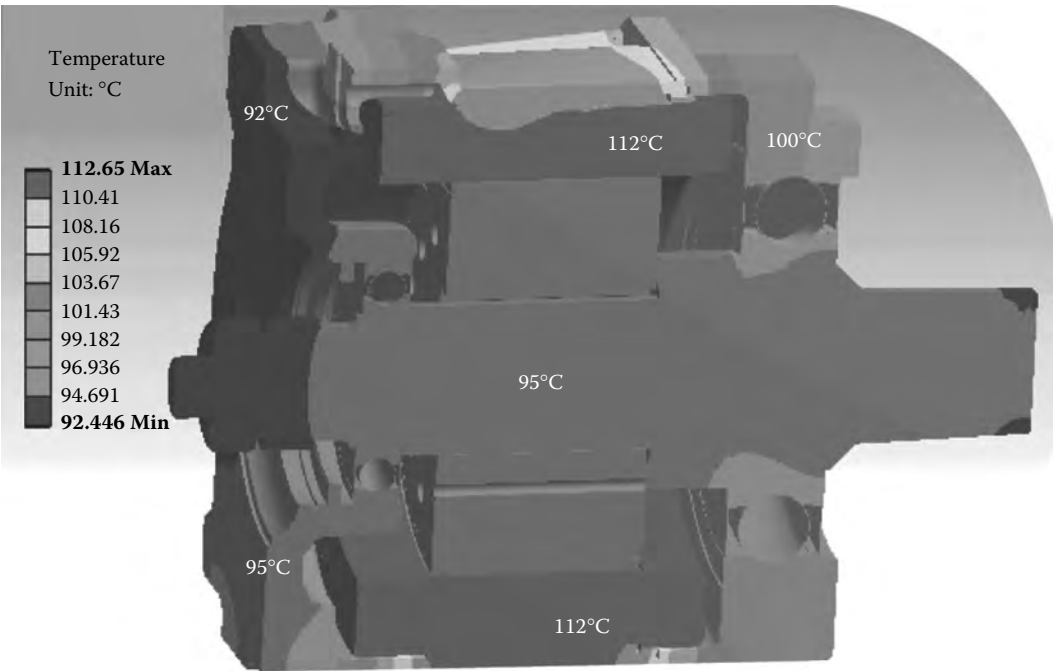


FIGURE 11.20
Temperature distribution on motor components.

The temperature at the large bearing of the drive side is about 5°C higher than that of the small bearing. Due to the gaps between the encapsulated stator winding and two endbells, the temperatures at the endbells are 10°C–18°C lower than that of the stator core. By eliminating these gaps, the temperature difference between the winding and endbells can be reduced less than 12°C.

11.4 Rotordynamic Analysis

Rotordynamic analysis is of great practical importance when designing rotating systems such as motors, generators, pumps, and compressors. It has been widely accepted that high-vibration levels of electric motors are caused by lateral critical speeds near the operating speeds. By modeling the rotating geometry and its dynamic characteristics, such as stiffness and damping, the critical speeds of rotating machinery can be predicted. As a consequence, design optimizations can be carried out to ensure that the machine operating speeds do not fall into the vicinities of its critical speeds.

The finite element rotordynamic analysis and critical-speed design sensitivity investigation are performed with a motor fan system. Results show that critical-speed separation margins of more than 30% are obtained from a rated speed of 60,000 rpm without any adverse effects from the spline shaft and that the critical-speed change rates to the support modeling of spline shaft connection points are extremely negligible. Furthermore, the critical-speed change rates to the shaft-element length changes show quantitatively that the spline shaft has some limited influence on the fourth critical speed but practically no influence on the first to third critical speeds.

11.4.1 Problem Description

Cooling fans in electric motors are typically mounted at the end of the rotor to produce high-pressure cooling air for the motor cooling. The cooling fan blows or sucks air into the motor being cooled and exhausts hot air from the interior of the motor to either an environment or a heat exchanger to be cooled before reentering the machine. A seal must be positioned between the high-pressure hot gas discharging from the fan and the low-pressure cold gas inlet to the rotor. As prevailing market trends require higher performance, high efficiency, higher reliability, lower cost, and high-power density motors, motor cooling, especially at the stator and rotor end regions, becomes a limiting factor.

Fan loss is the prime mover power transmitted through the rotor shaft to raise the static pressure of the cooling air flow through the fan. An examination of motor design data shows that fan loss could be a significant portion among the total windage losses. This loss can be effectively reduced by optimizing the fan parameters in design, including fan blade profile (for instance, 3D blade profile), number of fan blades, inlet and exit conditions, diffuser performance, and fan tip clearances.

Usually, the lack of a fan test facility causes any new fan design to rely on only numerical simulation results from CFD analysis. However, a high risk is incurred without proper validation of the numerical model through experimental investigations. The most reliable information about a physical process is given by real measurement. The motor design process can benefit from full-scale fan test data for optimizing fan performance and motor design.

With strong demands for large-size, high-power, and cost-effective electric machines, there have been growing concerns to build up the fan testing facility for optimizing machine cooling design. As the key step of a typical design process for rotating machinery, the rotordynamic analysis must be carried out prior to the construction of the fan facility.

The rotordynamic model includes the shaft–rotor assembly (shaft, couplings, and bearings), the rotor bearings, and their supports. The stiffness and mass of the bearing support play a crucial role in the calculation of the rotor critical speeds. The objective of the analysis is to determine the damped and undamped critical speeds of the fan test system with the bearing supports' stiffness, damping, and mass modeled.

11.4.2 Bearing Support's Stiffness and Damping

As demonstrated in Figure 11.21, the fan test system consists of a shaft, a forward disk and an aft disk both mounted on the shaft, journal bearings, and bearing supports. The outer ring is mounted on the disks rigidly, and fan blades are attached to the outer ring and forward disk. The system is supported with two bearings at the rotor ends with asymmetrical translational stiffness and damping values. In addition, the two bearings provide the additional damping for stabilizing the systems.

The stiffness, damping, and effective mass of the bearing supports can be determined from experimental tests. An effective approach is to use an instrumented hammer to excite the system and measure its response. Equipped with an accelerometer at one end, this hammer is used in tandem with the sensor to measure the vibration. This technique is usually called the impact input test, or impedance test, which is effective, because the impact inputs a small amount of force in the system and receives the response over a large frequency range. To properly apply this technique in this problem, it needs to select a number of locations on the bearing support for obtaining accurate results, as shown in Figure 11.22.

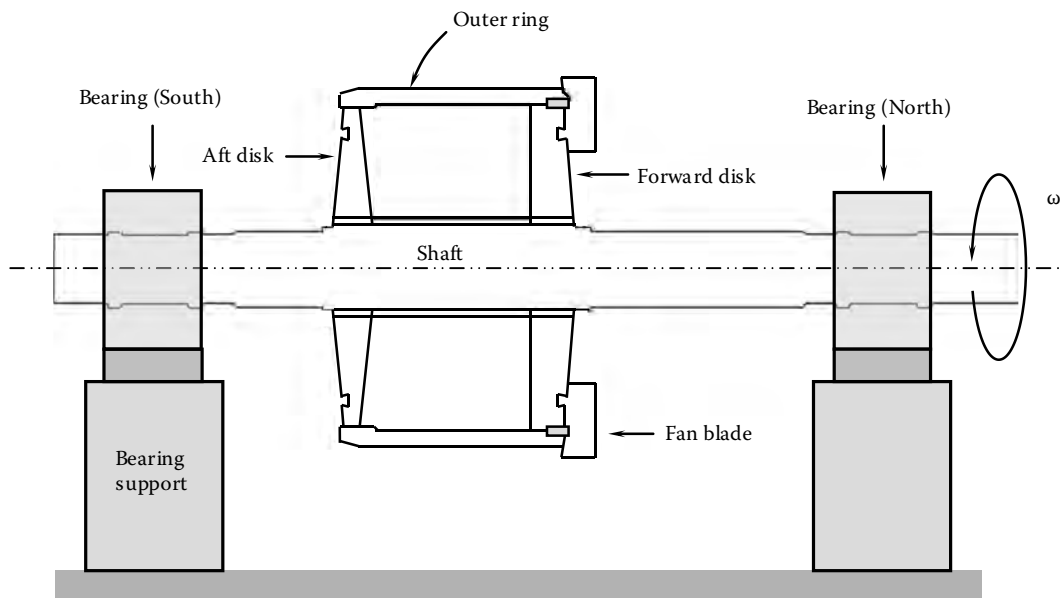
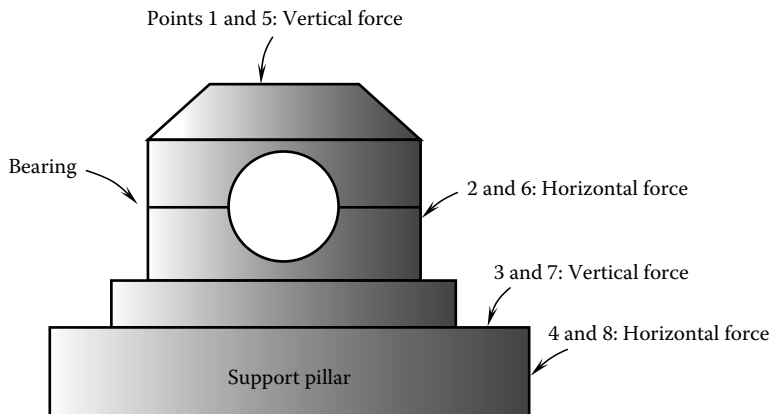


FIGURE 11.21

Configuration and components of the fan test system.

**FIGURE 11.22**

Using an instrumented hammer to excite the bearing support to obtain the natural frequency, damping, stiffness, and mass. Points 1 and 4 are taken on the south bearing, and points 5 and 8 are taken on the north bearing. Points 1, 3, 5, and 7 are vertical measurements, while points 2, 4, 6, and 8 are horizontal measurements.

TABLE 11.2

Bearing Support Stiffness, Damping, and Mass Data

Direction	Bearing Support	Point Location	Damping (N-s/m)	Natural Frequency		Stiffness (N/m)	Mass (kg)
				Hz	rpm		
Vertical	South	1	249,556	70	4200	5.15×10^8	2715
	North	5	340,271	82	4920	3.50×10^8	1319
Horizontal	South	2	51,837	37	2220	5.25×10^7	972
	North	6	14,010	19	1140	1.75×10^7	1228

The resulting test data as a function of frequency include force/acceleration ($\text{N-s}^2/\text{m}$), force/velocity (N-s/m), and force/distance (N/m). The stiffness in each direction is determined using the force/distance curve and using the value as the frequency approaches zero. The supports' damping is obtained by using the force/velocity curve and using the value at the natural frequency. The natural frequency for each bearing support direction and the resulting damping values are shown in Table 11.2.

All of the components are steel ($\rho = 7833 \text{ kg/m}^3$) except for the blades that are aluminum ($\rho = 2796 \text{ kg/m}^3$). The rotordynamic model of a fan test system is shown in Figure 11.23. The upper half of the cross section represents the geometry used to calculate the mass properties, and the lower half represents the geometry used to calculate the stiffness. For each of the two disks, the density and outer radius were solved for to exactly match the correct mass and mass polar moment of inertia. The forward disk includes the blade mass properties.

The disks' connections to the shaft are modeled as if they are continuous material with the shaft. The outer-ring press fit joints to the disks are modeled with a *rigid bearing* connection. This modeling technique couples the slope and displacement of the two components together.

For the undamped critical-speed analysis, only one bearing stiffness is needed. Therefore, the averaged stiffness is used in the undamped critical-speed calculation

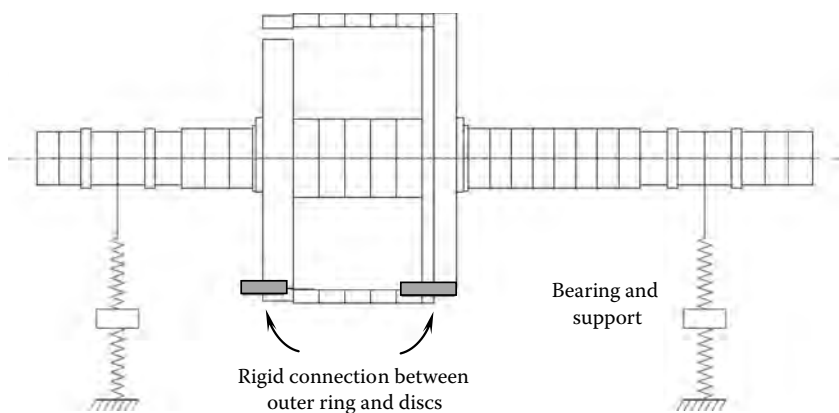


FIGURE 11.23
Rotordynamic model of fan test system.

TABLE 11.3
Translational Stiffness and Damping of the Fan Bearing Support

Translational Stiffness		Translational Damping	
k_{xx} , horizontal (N/m)	k_{yy} , vertical (N/m)	c_{xx} , horizontal (N-s/m)	c_{yy} , vertical (N-s/m)
1.93×10^8	3.85×10^8	4.48×10^5	8.50×10^5

(where $\bar{S}_b = 2.89 \times 10^8 \text{ N/m}$). For damped critical-speed analysis, both horizontal and vertical stiffness and damping are incorporated into the model (Table 11.3).

The primary advantage in using this technique is that it can monitor phase shifts, vibration force, and coherence. When the slope of the phase–frequency curve becomes infinity, the frequency at which it occurs is the natural frequency.

The mass of the bearing support m_b is determined with the bearing support’s stiffness k_b and natural frequency ω_n :

$$m_b = \frac{k_b}{\omega_n^2} \tag{11.29}$$

11.4.3 Rotordynamic Modeling

A commercial rotordynamic software has been applied for this analysis. This software is a rotordynamic program based on an FEA method and takes into account rotary inertia, shear deformation, and gyroscopic effects.

The geometry of the fan test system is shown in Figure 11.23. The mass and mass moments of inertia of the disks, outer ring, and blades are obtained from the 3D solid model. All components are made of steel except the fan blades that are made of die casting aluminum. In the rotordynamic model, the bearing spring is modeled in series with the bearing support spring.

The undamped and damped critical speeds are evaluated for synchronous forward and backward whirl. The stability of the damped critical speeds is also evaluated. The test rig design speed is 3000 and 3600 rpm with a 1.2× overspeed condition of 4320 rpm.

11.4.4 Results of Rotordynamic Analysis

The undamped and damped critical speeds are evaluated for synchronous forward and backward rotations. The stability of the damped critical speeds is also evaluated. Modern high-performance motors normally operate above the first critical speed, which is considered as the most important mode in the system, and avoid continuously operating at or near the critical speeds. The typical evaluation criterion is to maintain a critical-speed margin of $\pm 10\%$ between the operating speed and the nearest critical speed. Otherwise, redesign is required. The undamped critical speeds are shown in Table 11.4.

In practice, the mode shape corresponding to a critical speed is important in determining how the rotor system might vibrate when the critical speed is excited. The undamped mode shapes associated with critical speeds for the forward synchronous rotation are presented in Figure 11.24. The first mode at 2096 rpm shows that the rotor displacement at the bearings is relatively small while the vibration amplitude at the rotor center is much larger. The second mode has a conical mode shape with substantial motion at the bearings at the critical speed of 3422 rpm. This mode type is usually sensitive to rotor speed. The third mode occurring at 4847 rpm is similar to the first mode but has the larger rotor motion at the bearings. The fourth mode is similar to the second mode with a very small displacement at one rotor end.

Under the backward synchronous rotation condition, first three undamped modes keep the same shapes as the forward synchronous rotation case. The difference of the corresponding critical speeds between the two cases is less than 3%. However, the critical speed of the fourth mode becomes 4,568 rpm, which is much lower than 11,459 rpm in the previous case. Correspondingly, the mode has the second bending shape (Figure 11.25).

The evaluation of rotor stability is an important aspect of motor rotor design. Rotor system instabilities are normally associated with poor designs and inadequate selection of bearings. In practice, rotor stability is normally evaluated by the amount of damping on the first mode. The standard measure of mechanical damping is the logarithmic decrement, which is computed as the natural logarithm of the ratio between the amplitudes of two successive peaks. The relation between the mode logarithmic decrement δ and the corresponding damping ratio ζ can be found to be [11.22]

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \tag{11.30}$$

TABLE 11.4
Undamped Critical Speeds

Mode	Forward Synchronous Rotation				Backward Synchronous Rotation			
	Critical Speed (rpm)	Strain Energy (%)			Critical Speed (rpm)	Strain Energy (%)		
		Rotor	Supports	Bearings		Rotor	Supports	Bearings
1	2,096	10	22	68	2081	11	22	67
2	3,422	0	38	62	3033	1	33	66
3	4,847	39	50	11	4700	41	42	17
4	11,459	9	2	89	6568	59	21	20

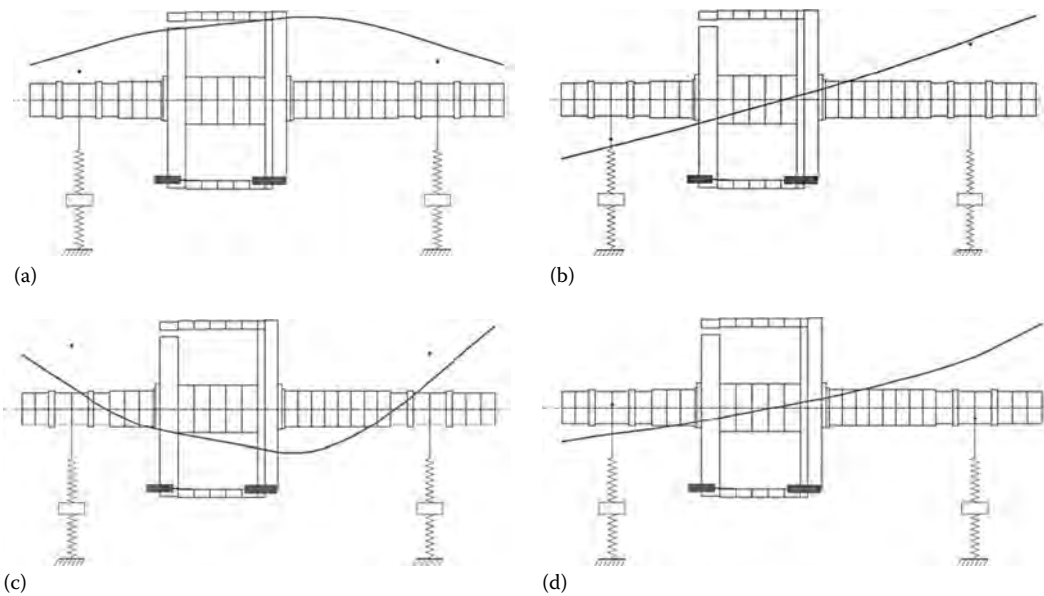


FIGURE 11.24 Undamped critical speeds and mode shapes of forward synchronous rotation: (a) mode 1: 2,096 rpm; (b) mode 2: 3,422 rpm; (c) mode 3: 4,847 rpm; (d) mode 4: 11,459 rpm.

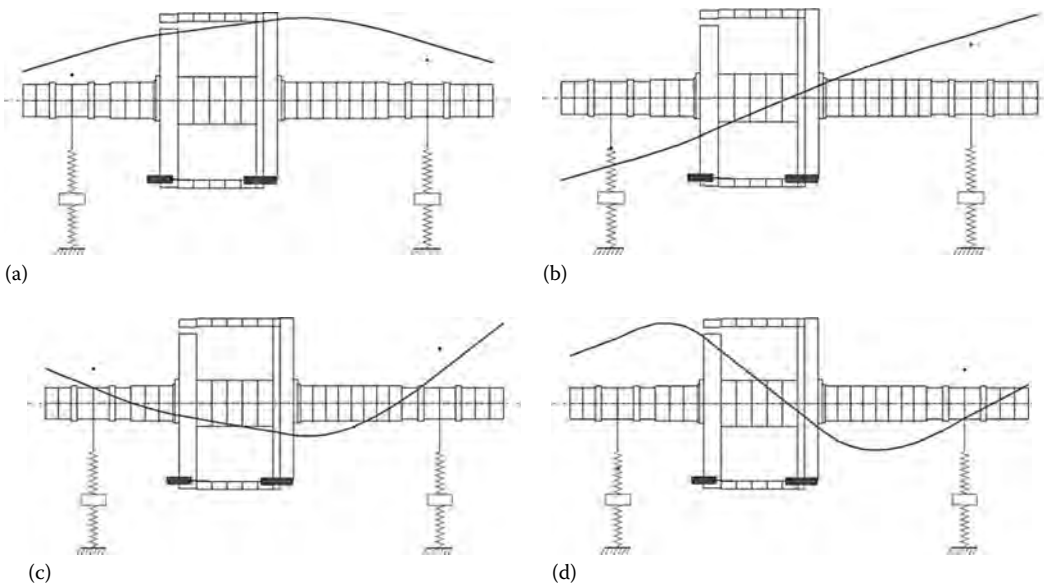
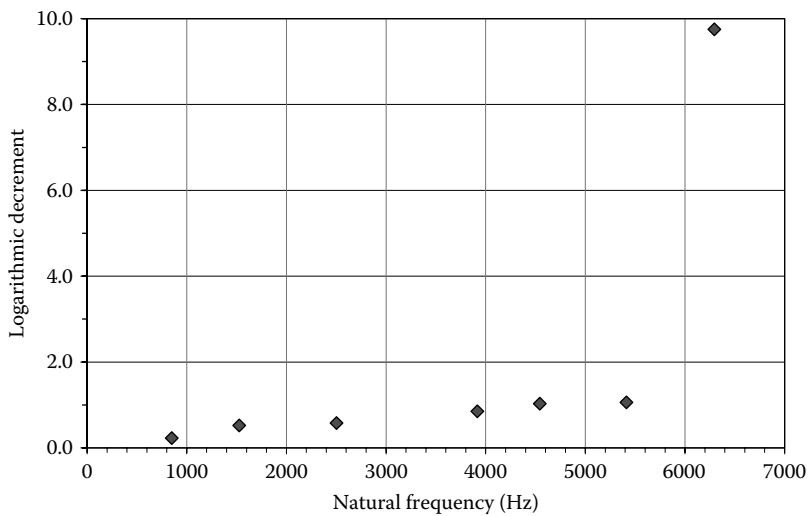


FIGURE 11.25 Undamped critical speeds and mode shapes of backward synchronous rotation: (a) mode 1: 2081 rpm; (b) mode 2: 3033 rpm; (c) mode 3: 4700 rpm; (d) mode 4: 6568 rpm.

**FIGURE 11.26**

Damped natural frequencies versus damped ratios for determining the system stability.

TABLE 11.5

Damped Critical Speeds

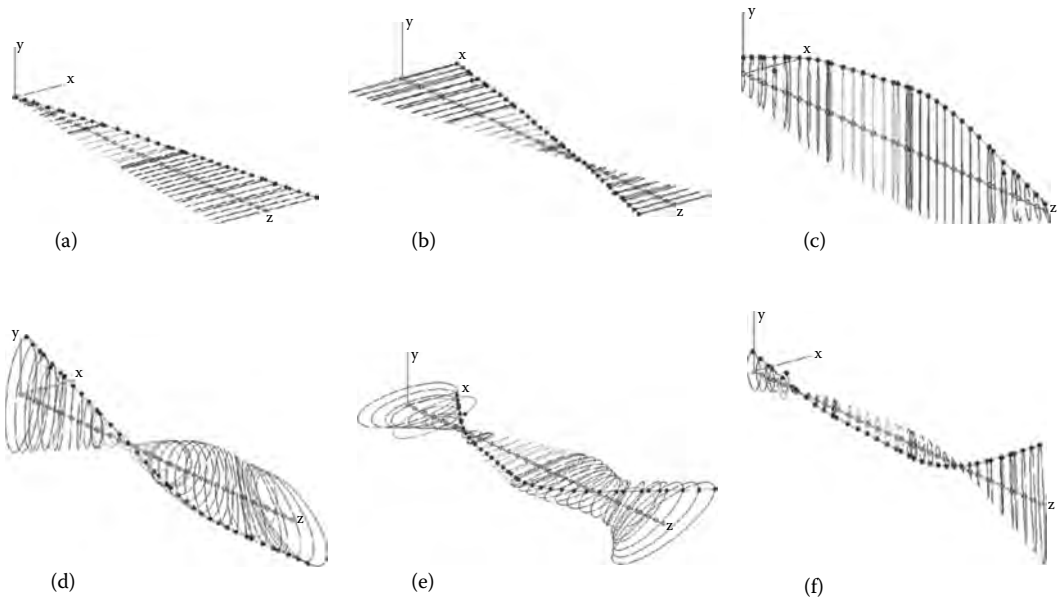
Mode No.	Critical Speed (rpm)	Logarithmic Decrement at Critical Speed	Stability	% Near 3000 rpm	% Near 3600 rpm	% Near 4320 rpm
1	850	0.23	Stable	-71.7	-76.4	-80.3
2	1527	0.52	Stable	-49.1	-57.6	-64.7
3	2502	0.58	Stable	-16.0	-30.0	-41.7
4	3915	0.85	Stable	30.5	+8.8	-9.4
5	4543	1.03	Stable	51.4	+26.2	+5.2
6	5413	1.06	Stable	80.4	+50.4	+25.3
7	6295	9.75	Stable	109.8	+74.9	+45.7

Theoretically, for $\delta > 0$, the system vibration will be damped out with time, and the system is considered stable. On the contrary, for $\delta < 0$, the vibration will increase with time, and the system is considered unstable. However, per API Standard 617 [11.23], for a rotor system to be stable, a minimum logarithmic decrement of 0.1 is required.

The damped natural frequencies ω_n versus logarithmic decrement δ are plotted in Figure 11.26. It can be seen from the figure that all logarithmic decrement values larger than 0.1 and thus all modes pass the stability evaluation criteria.

Table 11.5 presents the predicted damped critical speeds. The data shown in this table indicate that mode 4 fails the $\pm 10\%$ avoidance criteria for the 3600 rpm operating speed. Modes 4 and 5 fail the $\pm 10\%$ avoidance criteria for the 4320 rpm overspeed condition.

The mode shapes of damped critical speeds are plotted in Figures 11.27 and 11.28 for detail descriptions. The operating speed ranges of 50 and 60 Hz machines, as well as the overspeed range, are plotted to compare with the damped critical speeds.

**FIGURE 11.27**

Damped critical-speed mode shapes: (a) mode 1 at 850 rpm; (b) mode 2 at 1527 rpm; (c) mode 3 at 2502 rpm; (d) mode 4 at 3915 rpm; (e) mode 5 at 4543 rpm; and (f) mode 6 at 5413 rpm.

11.5 Static and Dynamic Stress/Strain Analysis

In the design of electric motors, static or dynamic analysis must be performed to determine the stresses, strains, displacements, and factor of safety in motor components and structures. Stress and strain define the intensity of internal reactions of deformed solid components with the changes of dimension and shape by externally applied forces. Stresses can be further divided as tensile, compressive, or shearing stress, according to the type of loads and straining action of the loading parts. Load-deformation data obtained from tensile or compressive tests can be used to calculate the stress of the specimen:

$$\sigma = \frac{F}{A} \quad (11.31)$$

where

F is the applied force on the specimen

A is the original cross-sectional area of the specimen

Strain is a measure of the change in the specimen's length Δl to its original length l_o , defined as

$$\varepsilon = \frac{l - l_o}{l_o} = \frac{\Delta l}{l_o} \quad (11.32)$$

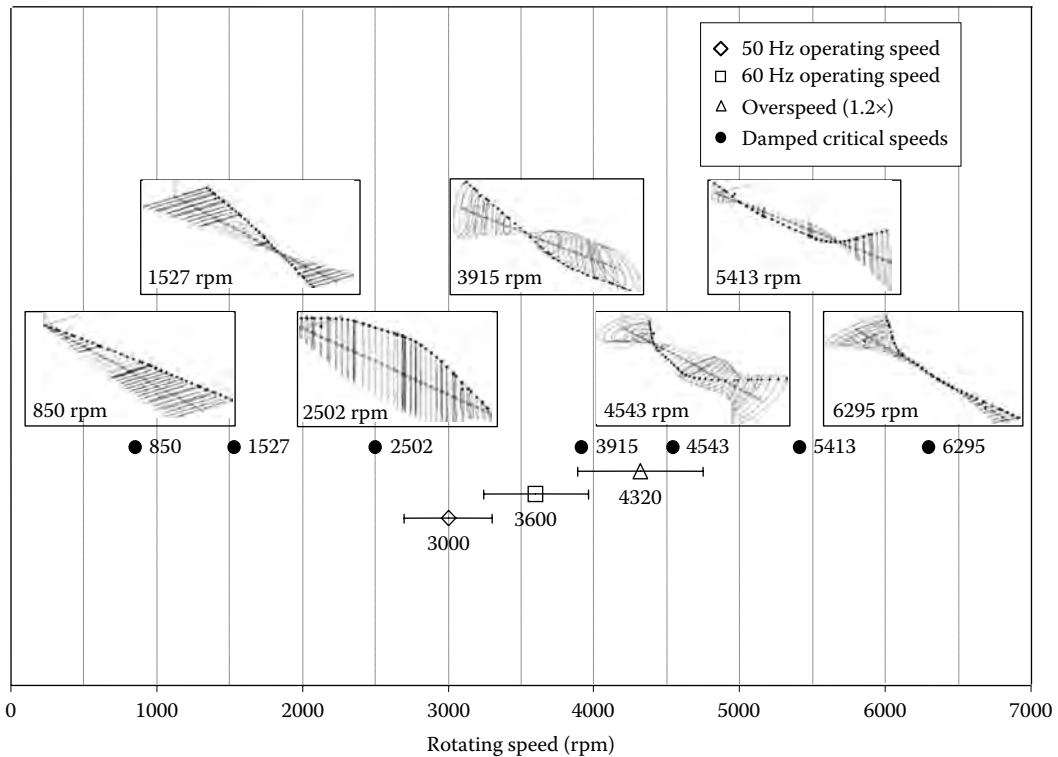


FIGURE 11.28
Predicted damped critical speeds and corresponding mode shapes.

For most solid materials, the tensile/compressive stress is directly proportional to the strain, following the Hooke's law:

$$\sigma = E\varepsilon \quad (11.33)$$

where E is the modulus of elasticity or Young's modulus. It is noted that Hooke's law describes only the initial linear portion of the stress-strain curve.

Similarly, the relationship of the shear stress τ and shear strain γ is given as

$$\tau = G\gamma \quad (11.34)$$

where G is the shear modulus of elasticity. For steel and aluminum, the shear modulus of elasticity G is approximately 40% of the modulus of elasticity E .

Unlike stress, strain is a dimensionless quantity and is often expressed in units of mm/mm or in./in. In practice, because the magnitude of measured strain is very small, it is sometimes expressed in microstrain units (e.g., $\mu\text{m}/\text{m}$).

11.5.1 Static Analysis

A static analysis is an essential part of the design process that enables the study of stress, strain, displacement, and shear and axial forces that result from static loads. The word

“static” indicates that the loads maintain invariable in their magnitudes, directions, and acting locations all the time, such as the gravitational force.

Two equilibrium equations are often used in static analysis: force balance and momentum balance with the expressions

$$\sum_i F_i = 0 \quad (11.35)$$

$$\sum_i M_i = 0 \quad (11.36)$$

11.5.2 Dynamic Analysis

A dynamic analysis determines the motor structural response based on the characteristics of the structure and the dynamic loads acting on the motor. Unlike a static loading, a dynamic loading is the forces/loads that move or change when acting on a structure. In normal operation, a rotor is subject to various loads such as electromagnetic force, external bending moment, axial force, friction force, and windage drag force. Due to the rotor rotation, these applied forces always change the directions on the rotor.

11.5.2.1 Centrifugal Force–Induced Stress on PMs

Centrifugal force refers to the force that tends to pull an object toward the axis around which it rotates and can be calculated with the formula

$$f_c = \frac{mu^2}{r} = \frac{m(r\omega)^2}{r} = mr\omega^2 \quad (11.37)$$

where

m is the mass of the object

r is the radius

ω is the rotating speed

Equation 11.37 can be used to calculate the centrifugal force acting on a PM and the average stress on the magnet. Assuming that magnet mass $m = 0.0328$ kg, the magnet contact area $A = 744.5$ mm², and the radius $r = 62.8$ mm, the centrifugal force acting on the PM is an exponential function of the rotating speed, as shown in Figure 11.29.

With the calculated centrifugal force, the averaged tensile stress acting on the magnet is determined as

$$\sigma = \frac{f_c}{A} \quad (11.38)$$

The stress–speed curve is presented in Figure 11.30, showing that the magnet stress increases exponentially with the rotating speed.

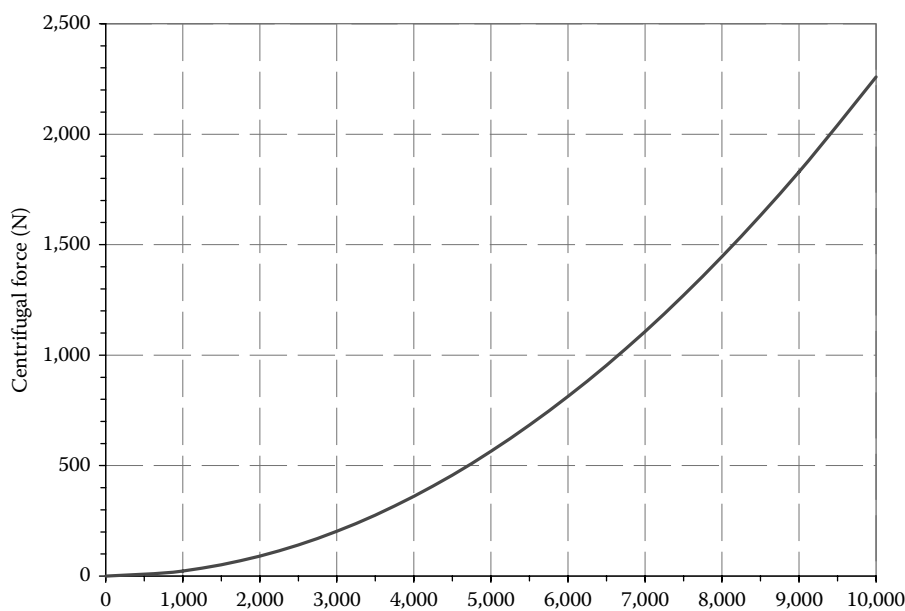


FIGURE 11.29
Centrifugal force acting on a PM under different rotating speeds.

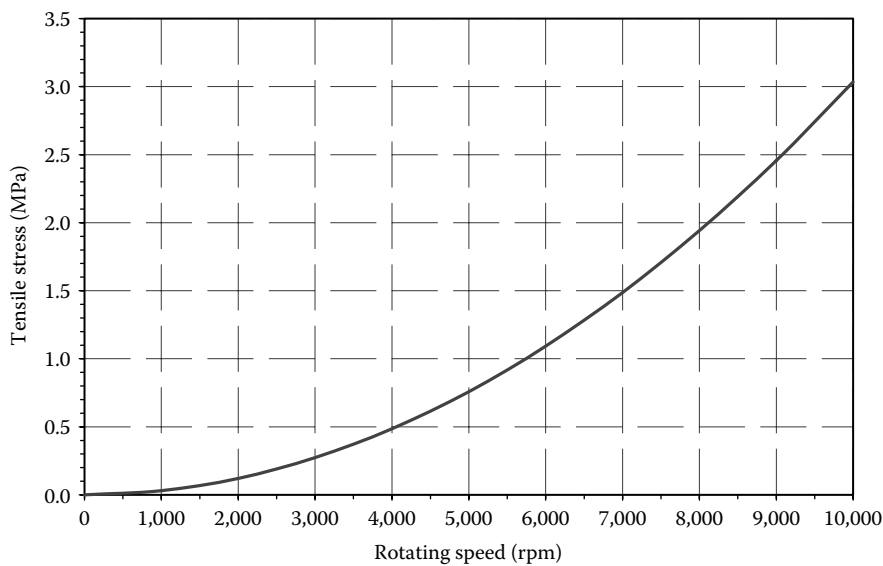


FIGURE 11.30
Average tensile stress of a PM resulted from centrifugal force under different rotating speeds.

11.5.2.2 Structural Analysis Using Finite Element Method

Analytical solutions for modern electric motors under dynamic loading conditions are unlikely to be available due to highly irregular motor geometry, complex loading condition, heterogeneous material properties, and nonlinear dynamic response of structure to a variety of loads. In practice, structural analysis is usually performed using an FEM.

Taking the fan test system as an example, due to the system symmetry, only a small portion of the system is modeled, as illustrated in Figure 11.31. For this problem, the primary consideration is the safety of the rotating system, particularly focusing on two important aspects:

- At high rotating speeds, the stress produced in each rotating component must be lower than its yield strength.
- During operation, all attached components such as blades must be held firmly against centrifugal force and other loads.

The comparison of various stresses (von Mises, radial, Hook, and axial) of the blade is graphically displayed in Figure 11.32. The stress data are well below the material yield strength, indicating that the blades can safely operate at the rotating speed of 4320 rpm.

For the safety consideration, a severe failure scenario is simulated. It assumes one blade, and its cap screws are missed from the system during operation. The aim of the analysis is to find out that under such a severe circumstance whether the whole system collapses.

A structural analysis is carried out for this scenario. The results are illustrated in Figure 11.33 for both maximum principal stress and radial displacement. These results

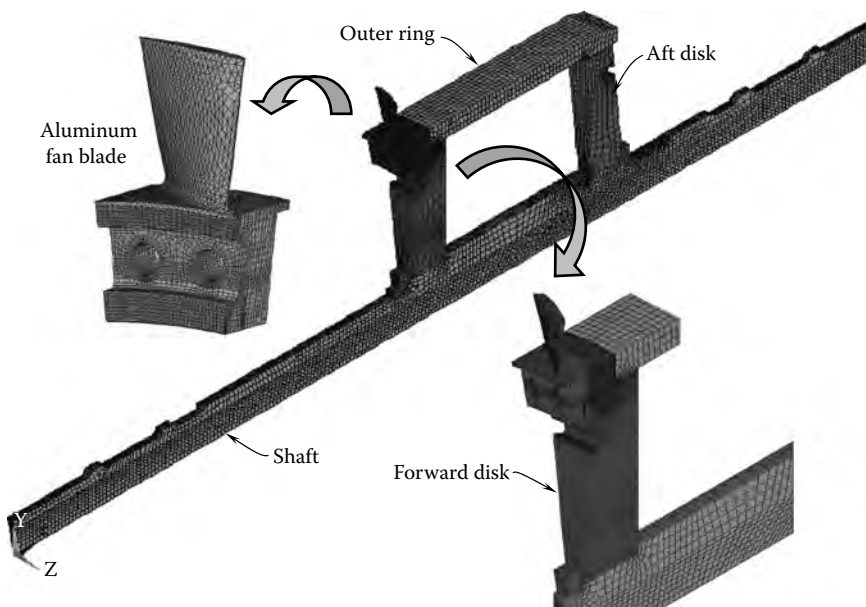


FIGURE 11.31
Mesh of the sector model for structure analysis.

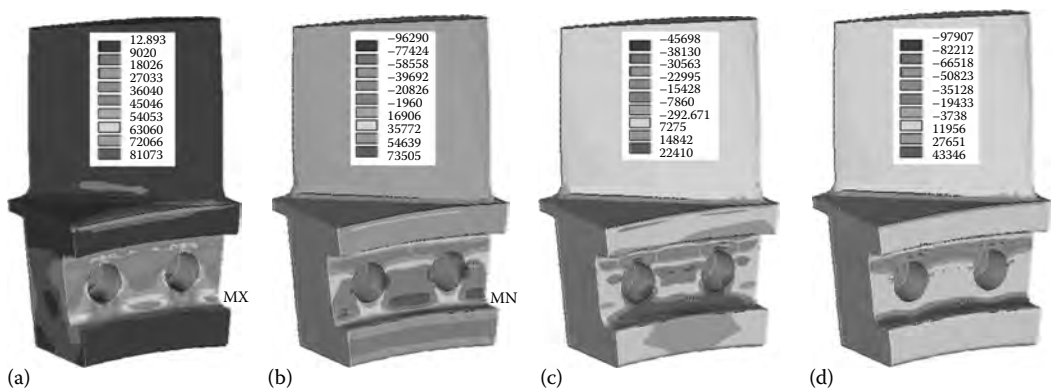


FIGURE 11.32 Comparison of various stresses of a fan blade at 4320 rpm: (a) von Mises stress, (b) radial stress, (c) hoop stress, and (d) axial stress.

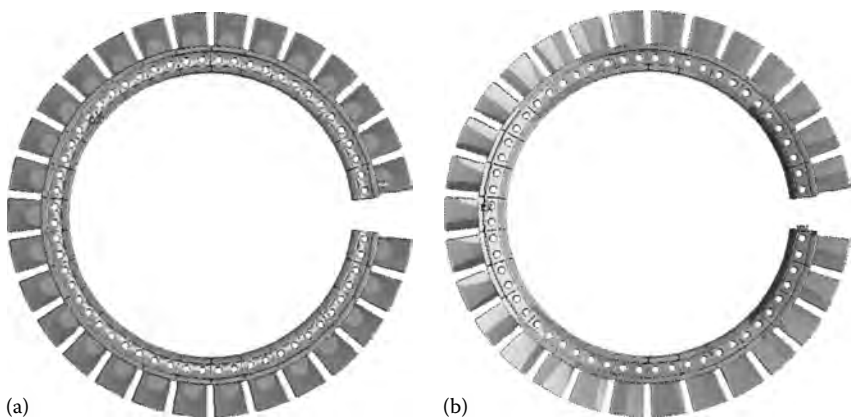


FIGURE 11.33 Results of structural analysis for missing a blade and its cap screws at 4320 rpm: (a) maximum principal stress and (b) radial displacement.

clearly show that the system unbalance due to the blade missing has a significant effect on the blade displacement. However, it does not lead to the system failure.

11.5.3 Shock Load

Electric motors may also experience shock loads caused by some accidental events such as earthquake, free fall, and sudden bump. Motors on fast moving objects (satellites, missiles, rockets, etc.) have to withstand high shock loads during acceleration and deceleration periods. According to Newton’s second law, $F = ma$, the force F is proportional to the mass of the object m and the acceleration a (or deceleration if a is negative). Thus, the force F can be considerably high under high acceleration/deceleration conditions.

When shock loads occur during motor operation in a very short period of time, if the shock loads are high enough, failures of motor components can take place immediately or at late time. To prevent motor failure, a common approach to deal with shock loads at

the motor design stage is to define an artificial gravity (i.e., g value, which is a measure of acceleration) in equations at horizontal and vertical directions. For instance, 5–10 g is commonly applied for moving vehicles, and 30–100 g for missiles.

11.6 Fatigue Analysis

It is estimated that about 60% of permanent damages are caused by fatigue failure of materials and elements [11.24]. One of the main concerns in the motor design is to extend the motor's life. Factors that influence the motor fatigue life include load history, geometry, relevant materials, and manufacturing processes.

Fatigue in materials is the process of initiation and growth cracks under alternative tensile loading conditions. Fracture occurs when the effects of total stress and flaw size exceed a critical value commonly referred to as the fracture toughness. The fracture toughness depends upon a number of factors, such as microstructure and composition of the material, service temperature, loading rate, plate thickness, and fabrication processes [11.25]. However, an accurate determination of the fracture toughness is complicated.

In structural component design of electric motors, the appropriate criterion for fatigue failure should be based on consideration of failure modes of the component being designed. The fatigue damage caused by repeated dynamic loads depends on the number of cycles and the frequency of the occurrence of significant stresses. A number of motor parts are subject to cyclic loads during motor operation, leading to fatigue of materials. The material fatigue initiates from the crystalline structure and becomes visible in a later stage by plastic deformation, formation of microcracks on slip bands, coalescence of microcracks, and finally propagation of a main crack.

Several approaches are available in the fatigue analysis, from the traditional time-domain S-N analysis to the frequency-domain approach. The time-domain approach is satisfactory for periodic loading but requires large time records to accurately describe random loading processes. As a contrast, compact frequency-domain fatigue calculations can be utilized where the random loading and response are categorized using power spectral density (PSD) functions and the dynamic structure is modeled as a linear transfer function [11.26].

3D fatigue analysis can be performed using FEM. This method predicts the behavior that is otherwise difficult to find out by theoretical calculations due to the results of large number of degree of freedom involved in it. As a matter of fact, FEM is an excellent tool to calculate the fatigue life of motor components.

11.7 Torsional Resonance Analysis

Torsional resonance is one of the main causes of motor vibration that can lead to fatigue failures of the motor. As illustrated in Figure 11.34, a simple torsional resonance test facility consists of a motor and a load wheel. The motor shaft is connected directly with the load shaft. As the rotor rotates, the inertia of the load wheel provides the torsional load to the rotor, and thus, the torsional resonance of the motor can be determined.

Torsional resonance is affected on both the driving device and rotor inertias, the shaft stiffness that relies on shaft material properties, shaft size, and shaft length.

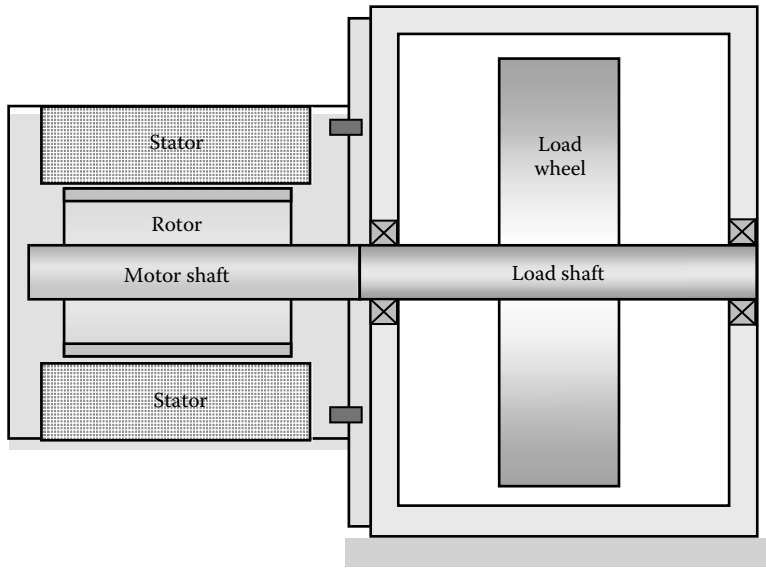


FIGURE 11.34
Torsional resonance measurement of electric motor.

The inertias of the load and motor are calculated in the following equations, respectively:

$$J_l = J_{wheel} + \sum J_{shaft} + \sum J_{i,bearing} \quad (11.39)$$

$$J_m = J_{rotor} + J_{PM} + \sum J_{shaft} \quad (11.40)$$

The torsional stiffness of each shaft segment (both load and rotor) must be determined using the formula

$$S_i = \frac{\pi(d_{out,i}^4 - d_{in,i}^4)G}{32l_i} \quad (11.41)$$

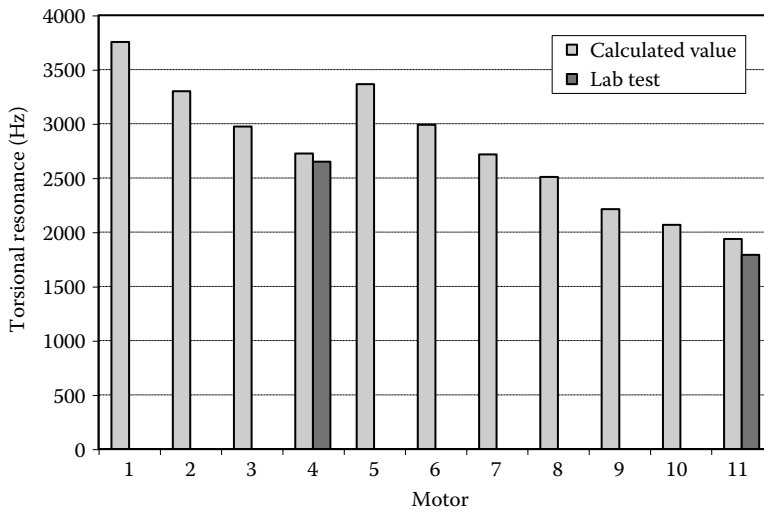
Thus, the total torsional stiffness is

$$S_t = \frac{1}{\sum (1/S_{i,load\ shaft}) + \sum (1/S_{i,rotor\ shaft})} \quad (11.42)$$

Torsional resonant frequency (Hz) is thus calculated as

$$f_{tr} = \frac{1}{2\pi} \sqrt{S_t \left(\frac{1}{J_m} + \frac{1}{J_l} \right)} = \frac{1}{2\pi} \sqrt{\frac{S_t (J_m + J_l)}{J_m J_l}} \quad (11.43)$$

The comparison of calculated and tested torsional stiffness is presented in Figure 11.35.

**FIGURE 11.35**

Comparison of calculated and tested torsional resonances for different types of motor.

11.8 Motor Noise Prediction

As a form of energy, sound is the propagation of low-amplitude pressure waves traveling with the speed of sound. In practice, noise typically refers to undesired or unpleasant sound. Acoustic noise from an electric motor may consist of a number of components: mechanical noise, which is associated with the relevant motion (e.g., rotation and vibration) between motor components; electromagnetic noise, which is activated by the fast change in the electric and magnetic field; and aerodynamic noise (or windage noise), which is introduced by flowing fluid inside the motor.

The main challenge in numerically predicting sound waves stems from the fact that sounds have much lower energy than fluid flows, typically by several orders of magnitude. This poses a great challenge to the computation of sounds in terms of difficulty of numerically resolving sound waves, especially in predicting sound propagation to the far field. Another challenge comes from the difficulty of predicting turbulent flow fields in the near field of the sound source [11.27]. In this field, the interferences between contributing waves from different parts of the source lead to the interference effect, which is greater for pure tones than for bands of noise. The prediction of motor noise level can be thus performed according to the type of noise using available commercial software.

Vibration is generally resulted from the unbalanced dynamic system. In a motor, noise is created by unequal weight distributions of the rotor assembly around its axis of rotation. In addition, noise may be generated by nonuniformly distributed rotor winding in some types of induction motors. It has been found that vacuum impregnation and/or encapsulation of stator has helped in reducing stator winding vibration and noise.

The noise reduction is always one of the design targets for motor engineers and designers. There are a number of engineering approaches available, including

- Carefully balancing the rotor to reduce motor vibration
- Increasing the thickness of motor frame wall
- Improving noise sealing by adopting acoustic blankets [11.28] or by coating/gluing a thin layer of noise-absorbing material on the motor surfaces

11.9 Buckling Analysis

In engineering practices, there are two major categories of failures in mechanical systems: material failure and structural instability. Buckling refers to a phenomenon that a part or component that is subjected to compression suddenly becomes unstable. As shown in Figure 11.36, a thin-walled sheet is subjected to compressive force F on its two ends. By increasing F until a certain value, the sheet will suddenly bend, indicating that it is no longer to withstand any loads. In this figure, δ_l and δ_o are load displacement and out-of-plane displacement, respectively. This phenomenon can also occur for other geometries such as columns, flanges, and shells. Because of its suddenness, buckling failure may cause catastrophic consequences.

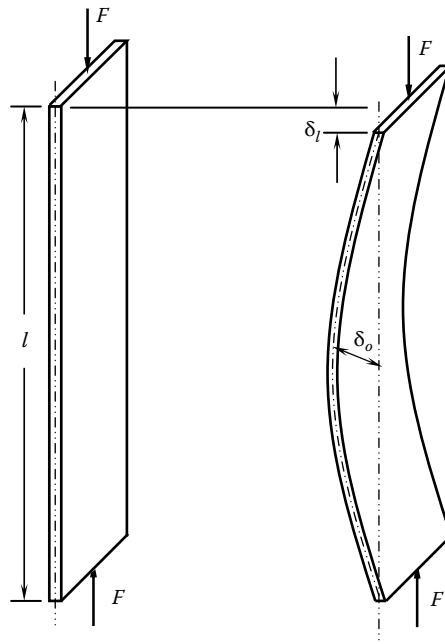


FIGURE 11.36
Buckling of a thin-walled sheet.

The key factor in buckling is the slenderness ratio of the member, defined as the ratio of the member length l and the radius of gyration k :

$$S = \frac{l}{k} = l \sqrt{\frac{A}{I}} \quad (11.44)$$

where

A is the cross-sectional area

I is the area moment of inertia

The stress at failure is called the critical buckling stress. With central loading, there are two models to cover the entire range of compression problems: Euler model and Johnson model [11.29]. The selection of an appropriate model depends on the value of slenderness ratio. The point S_1 is introduced to separate the two models:

$$S_1 = \pi \sqrt{\frac{2CE}{S_y}} \quad (11.45)$$

where

E is Young's modulus

S_y is the yield strength of the material

C is a constant depending on the column end condition

Thus, for long columns where $S > S_1$, the Euler model is used to calculate the critical buckling stress:

$$\sigma_{cr} = \frac{CE\pi^2}{S^2} \quad (11.46)$$

For intermediate-length columns where $S \leq S_1$, the Johnson model is applied as

$$\sigma_{cr} = S_y - \frac{1}{CE} \left(\frac{S_y S}{2\pi} \right)^2 \quad (11.47)$$

It is noted that critical buckling stress can be well below the material yield point.

In many practical problems, loads acting on columns are often away from their centroidal axes by the eccentricity e . Correspondingly, the critical buckling stress is calculated by superposing the axial component and the bending component:

$$\sigma_c = \frac{F}{A} \left[1 + \frac{ec}{k^2} \sec \left(\frac{S}{2} \sqrt{\frac{F}{EA}} \right) \right] \quad (11.48)$$

where

F is the force

c is the distance from the neutral axis to the bending surface

ec/k^2 is defined as the eccentricity ratio

11.10 Thermally Induced Stress Analysis

Thermally induced stresses in a motor occur as some or all parts are not free to expand or contract in response to changes in temperature due to the geometric and/or external constraints. Some examples encountered in electric motors are:

- For induction motors, conductor bars and end rings are fabricated by casting aluminum or copper into rotor slots to provide the induction current paths. Due to the different thermal expansions between the casted material and silicon steel laminations, shear stresses can be developed right away at the completion of the casting process due to the different cooling rates of the two materials. This is also true during a motor heating process or motor operation at high temperatures. In fact, frequent starting often imposes thermal stresses on electric motors.
- In a shrink-fitting or press-fitting assembly, the contact pressure between two contacting members changes as the temperature changes from its original fitting temperature, resulting in the variation of contact condition and nonuniform distribution of contact pressure on the contact surface.
- For a PMM, a number of PMs are typically mounted on a rotor. Because these magnets have twice the thermal expansion coefficient over silicon steel, large shear stresses can be produced on the magnets during motor normal operation, leading to uneven axial and radial forces across the magnets and thus possibly developing micro-scale cracks on the magnets.
- When the resin is encapsulated into a stator and cured in an oven for a period of time following a certain temperature profile, the thermal stress is developed in the resin correspondingly with the change of the operation temperature.
- The shaft and housing/endbells are usually made of different materials. A fast variation in temperature may introduce thermal stress or thermal interference problems.

11.11 Thermal Expansion and Contraction Analysis

Materials change their size with the changes in temperature and pressure at different rates. All gases and most liquids and solids expand when heated and contract when cooled, referred to thermal expansion and thermal contraction, respectively. The rate of thermal expansion is characterized by the CTE β for a specific material.

CTE is the measure of how much that material will expand with the change in temperature of 1°. Three types of CTE, that is, the linear, area, and volumetric thermal expansions, are defined as

$$\beta_L = \frac{1}{L} \left(\frac{\partial L}{\partial T} \right)_p \quad (11.49a)$$

$$\beta_A = \frac{1}{A} \left(\frac{\partial A}{\partial T} \right)_p \quad (11.49b)$$

$$\beta_V = \frac{1}{V} \left(\frac{\partial V}{\partial T} \right)_p \quad (11.49c)$$

For anisotropic materials such as crystals and many composites, β_L has different values in different directions. For isotropic materials, there is only one β_L in all directions. For these materials, it is easy to find that

$$\beta_A = 2\beta_L \quad \text{and} \quad \beta_V = 3\beta_L \quad (11.50)$$

Thus, as the temperature changes from T_o to T_f the variations in the material length, area, and volume can be calculated as

$$\Delta L = \beta_L L_o (T_f - T_o) \quad (11.51a)$$

$$\Delta A = \beta_A A_o (T_f - T_o) \quad (11.51b)$$

$$\Delta V = \beta_V V_o (T_f - T_o) \quad (11.51c)$$

where

L_o , A_o , and V_o represent the initial length, area, and volume of the material

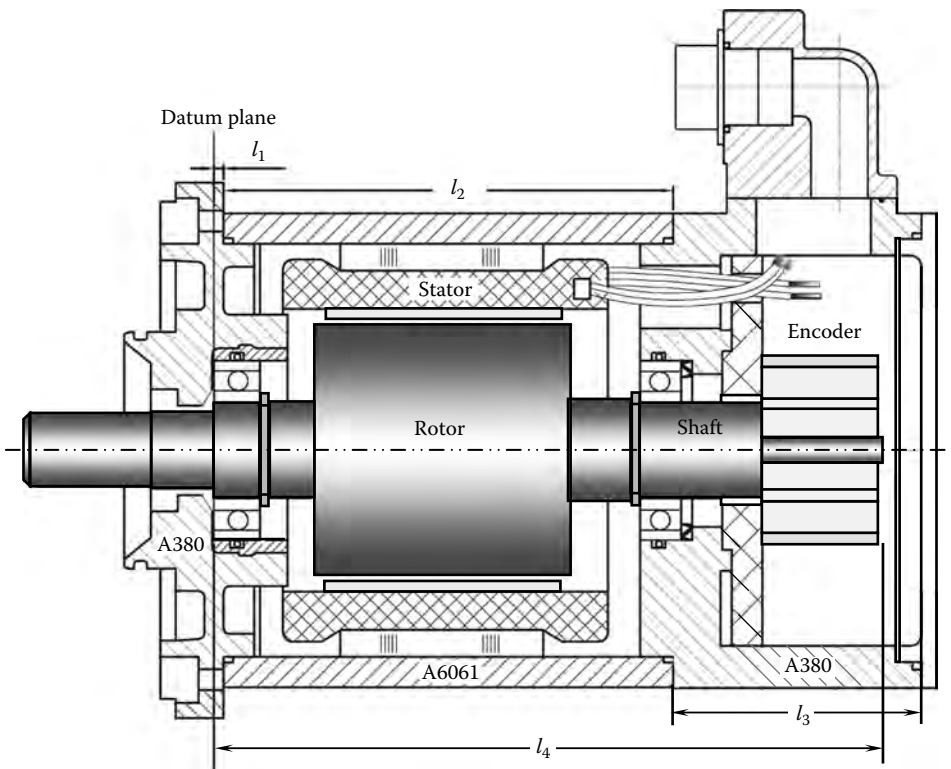
ΔL , ΔA , and ΔV are the variations in length, area, and volume due to the thermal expansion, respectively

The linear thermal expansion coefficients of some solid materials are listed in Table 11.6.

TABLE 11.6

Linear Thermal Expansion Coefficients of Some Solid Materials

Material	Linear CTE at 20°C (μm/m-°C)	Note
ABS—glass fiber	30.4	
Aluminum 356-T6	21.4	For sand casting
Aluminum 383	21.1	For die casting
Aluminum 6061-T6	23.6	
Carbon steel 1008	12.6	
Carbon steel 1045	11.5	Cold drawn
Gray cast iron	10.8	Averaged value for the material class
Copper	16.6	
Copper casting alloy	16.9	UNS C80100
Ductile iron A536	11.6	For casting
Epoxy—cast resins and compounds, unfilled	55	
Epoxy—cast, unreinforced	100	
Epoxy/glass SMC	3.6	
Fiberglass—polyester	25	
Stainless steel 304	16.9	Averaged value for 0°C–100°C

**FIGURE 11.37**

Calculations of displacement of the shaft and housing/endbell, respectively, for avoiding part interference during thermal expansion and contraction processes.

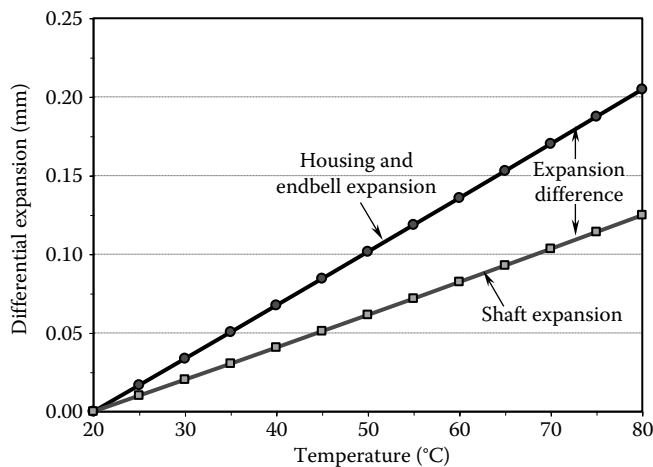
Thermal expansion and contraction analysis is to address the dimensional changes of motor components due to the variations in temperature during motor operation and storage processes. As a motor is heated or cooled, the dimensions of its components always vary corresponding to the temperature on them. The induced differential dimensions between these components may cause the parts to contact each other or, in more severe conditions, interferences between the parts. Therefore, to avoid undesired thermal-induced interferences, it is important to leave enough allowances for the expansion of hot components.

Figure 11.37 illustrates a regular PMM, consisting of two endbells made of die-casting aluminum A380 and a housing made from extruded aluminum A6061. The shaft is made of carbon steel AISI 1045. Because these materials have different thermal expansion rates, it is important to determine the displacements of the housing/endbell and the shaft to avoid any interference between these parts during thermal expansion.

Referring to Figure 11.37, the differential displacement of the housing and endbells Δl_h during the thermal expansion can be calculated as

$$\Delta l_h = \beta_{A380} l_1 \Delta T_1 + \beta_{A6061} l_2 \Delta T_2 + \beta_{A380} l_3 \Delta T_3 \quad (11.52)$$

where ΔT_i is the temperature difference between the part temperature T_i and the reference temperature T_{ref} (usually $T_{ref}=20^\circ\text{C}$). Similarly, the differential displacement of the shaft Δl_s is

**FIGURE 11.38**

Comparison of thermal expansions of housing/endbell and shaft, showing that the expansion of the housing/endbell always exceeds that of the shaft.

$$\Delta l_s = \beta_{\text{steel}} l_4 \Delta T_4 \quad (11.53)$$

With the real dimensions of the motor component, the calculated results are plotted in Figure 11.38 under different temperatures. It can be observed that at the reference temperature of 20°C, $\Delta l_h = \Delta l_s = 0$. For temperatures larger than 20°C, Δl_h is always larger than Δl_s , and the difference between Δl_h and Δl_s is proportional to the temperature. The results indicate that no interference will occur during the thermal expansion process between these parts.

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Innovative and Advanced Motor Design

The electric motors consume a significant amount of electricity in the industrial and tertiary sectors. In today's highly competitive global motor market, the demands for higher motor rates, higher efficiency, lower costs, and increased reliability are fueling the need for increased automation and control products with reduced energy consumption and carbon emissions. Many motor manufacturers have committed to develop their next-generation products for capturing the maximum market share and boosting profit margins. For these purposes, motor manufacturers not only fully utilize their own technical resources but also strengthen the cooperation with universities and research institutions for implementing technological breakthrough. However, there are still a variety of challenges that motor manufacturers have to face, for example, how to shorten the product development cycle time, how to apply new technologies in both existing and new products, how to differentiate their products and adapt to global markets, and how to effectively reduce the production costs. All these challenges must be appropriately considered and addressed to ensure successful outcomes.

Over the past few decades, the increasing demands for high efficiency, high torque density, low cost, and low noise electric motors have driven much R&D of innovative motor technologies. At the same time, rapid technological developments, such as 3D printing, digital technology, nanotechnology, and new composite materials, have made it possible to design and manufacture next-generation electric motors.

12.1 High-Temperature Superconducting Motor

The discovery of high-temperature superconductors (HTSs) in 1986 has strongly stimulated the R&D on superconducting materials and their industrial and military applications. In the following years, the potential economic benefits of HTS had initiated an international race among the United States, Europe, Japan, and China to develop the advanced superconductivity technology and its commercialization. Large efforts have been made to apply HTS toward power equipment such as electric motors, generators, energy storage systems, transformers, and power cables. HTS materials have much higher critical temperatures (above the liquid nitrogen temperature of 77 K or -196°C) than conventional low-temperature superconductors (LTSs) cooled with liquid helium (4.2 K or -269°C). The use of liquid nitrogen to cool HTS can lead to a reduced complexity of cryogenic system, improved system reliability and performance, and reduced total cost.

The advent of HTS has driven a quantum leap forward in advanced technologies of electric machines. Superconducting motors based on HTS are much more efficient than conventional electric motors due to zero resistance in superconducting windings. Because HTS can carry much higher currents than regular copper wires, these HTS windings are

capable of generating much more powerful magnetic fields with minimized power losses. For large synchronous superconducting motors, 50% volume and loss reduction can be achieved compared to conventional motors [12.1,12.2]. In addition, HTS motors have lower sound emissions than conventional motors. These features are especially attractive for some military customers. For instance, the US Navy is interested in applying HTS motors to its new generation of surface ships and submarines.

According to the use of HTS in motor windings, three types of HTS motors can be classified: (1) superconducting rotor type that uses HTS in the rotor winding, (2) superconducting stator type that uses HTS in the stator winding, and (3) full superconducting motor type that uses HTS in both rotor and stator windings.

Founded by the US Department of Energy (DoE), Reliance Electric Corporation demonstrated the world's first DC HTS motor in 1990 and AC synchronous HTS motor in 1993 [12.3]. The first HTS motor in Europe was made and tested at Siemens' Research Center in 2001. During experimental operation in motor and generator modes, the trial motor reached a continuous power output of 400 kW [12.4]. In 2005, IHI in Japan released the world's first full superconducting motor cooled with liquid nitrogen [12.5].

In the United States, American Superconductor has designed, built, tested, and delivered a 5 MW, 230 rpm, 6-pole HTS ship propulsion motor [12.6]. The motor uses an air core armature winding and first-generation HTS wire field winding. The aim of the project is to validate the technologies required to design and build larger HTS ship propulsion motors, as well as to develop a motor production process that streamlines development time and minimizes cost. A commercial variable frequency drive is used to power the motor. The HTS field winding uses gaseous helium as the cooling medium in a closed cycle.

The world's largest HTS ship propulsion motor of 36.5 MW (49,000 hp) was successfully tested at full power [12.7]. This motor can produce 2.9 million newton-meters of torque at a nominal rotating speed of 120 rpm. Incorporating coils of HTS wire that are able to carry 150 times the current of similar-sized copper wire, the motor is less than half the size of conventional motors and can reduce the ship weight by nearly 200 tons. With up to three times higher torque density than conventional motors, the HTS motor is more fuel-efficient. The size and weight benefits make HTS machines less expensive and easier to transport and install, as well as allow for arrangement flexibility in the ship. In addition, the absence of iron stator teeth significantly reduces the structure-borne noise.

In 2007, engineers in Sumitomo Electric designed a liquid nitrogen-cooled HTS motor of 365 kW. To simplify the cooling system and achieve high operating reliability, this motor adopts axial gap PM type. The motor armature composes of six iron-cored HTS windings and forms three phases. Heat generated by AC losses is efficiently dissipated into the liquid nitrogen through the cooling channels. This 365 kW HTS motor was successfully used to directly drive a contrarotating propeller in tandem with a 50 kW HTS motor. Due to the reduction of transmission losses, the HTS motor could provide the maximum torque of 120 N-m (at 1500 rpm) and the maximum speed of 85 km/h to electric vehicles [12.8].

A schematic diagram of an HTS motor is shown in Figure 12.1. As can be seen in the figure, while the stator winding is made of regular copper wires, the rotor winding is fabricated by HTS as superconducting magnets, which have higher energy density than PMs. The rotor winding is thermally insulated from the rest of the machine using multilayered insulation materials and cooled with a cryogenic coolant, which is introduced from the cryocooler and passes through the cooling loop in the support structure adjacent to the

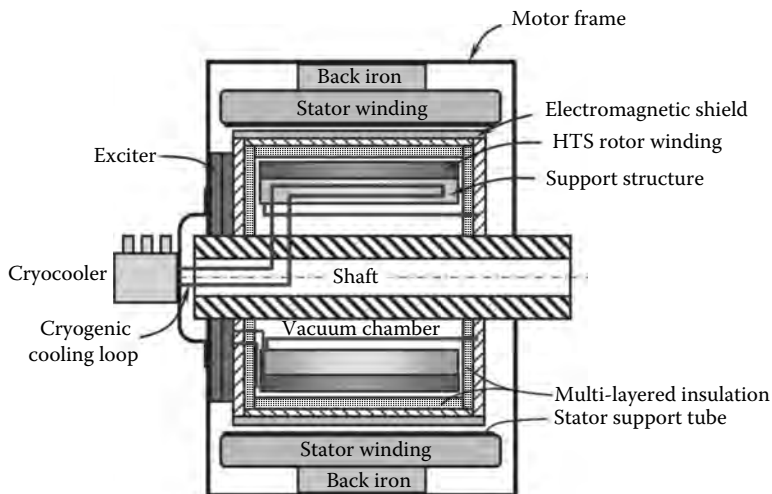


FIGURE 12.1
Schematic diagram of HTS motor.

rotor. In order to reduce radiation heat loads onto the cryogenically cooled components, the rotor winding is enclosed within a sealed vacuum chamber that rotates with the shaft. Unlike in a conventional motor, the stator winding is not housed in iron core teeth because they are saturated due to the high magnetic field imposed by the HTS windings [12.9].

Depending on the direction of the main air gap magnetic flux and the configuration of the rotor and stator against the rotating shaft, an electric motor can be classified into either RF, AF, or TF motors. Unlike the more commonly available cylinder-shaped rotor and stator in RF motors, AF motors are disk shaped, as the result of the stator and rotor being placed adjacent to each other.

The comparison of torque density of RF, AF, and TF PM motors has been performed by Pippuri et al. [12.10]. The work was carried out using a 10 kW, 200 rpm three-phase motor as the test case. The RF design is of the conventional inner rotor type. The AF design has a single-rotor, single-stator construction. The TF design has a U- and I-core stator layout with an inner rotor of two separate yokes. The magnets are mounted on the rotor surface and each of them covers 70% of the pole pitch. As shown in Table 12.1, the comparisons are made for two cases: most compact and best efficiency. In both cases, the RF topology yields the highest torque density with the given initial conditions and

TABLE 12.1

Comparison of Torque Density for Three Types of Motors

Motor Type	Torque Density for Most Compact Case (N-m/m ³)	Torque Density for Best Efficiency Case (N-m/m ³)
RF motor	16,877	14,380
AF motor	13,226	10,023
TF motor	11,504	11,504

Source: Data obtained from Pippuri, J. et al., *IEEE Trans. Magn.*, 49(5), 2339, 2013.

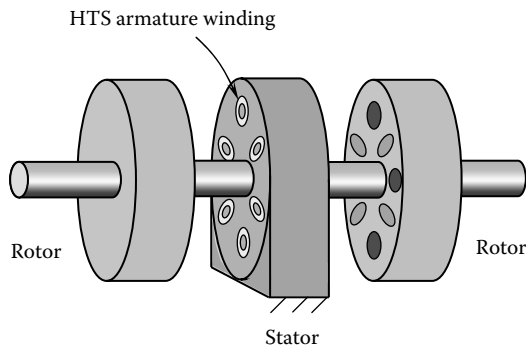


FIGURE 12.2
AF, high-temperature superconducting motor.

constraints. When comparing the AF and TF machines (TFM), the AF machine outperforms the TF design in the most compact case, and it is opposite in the best efficient case. It is interesting to note that the torque density for the TF motor remains unchanged in the two cases.

The flux of the AF motor is in the same direction with the motor shaft. Because of the large area for magnetic flux, the AF motor provides the desired performance. A schematic diagram of an AF, high-temperature superconducting motor is shown in Figure 12.2. The stator is in the middle of the motor and the two rotors are on the two sides of the stator. The stator winding is made by high-temperature superconducting materials and is cooled by a specially designed cooling system (not shown).

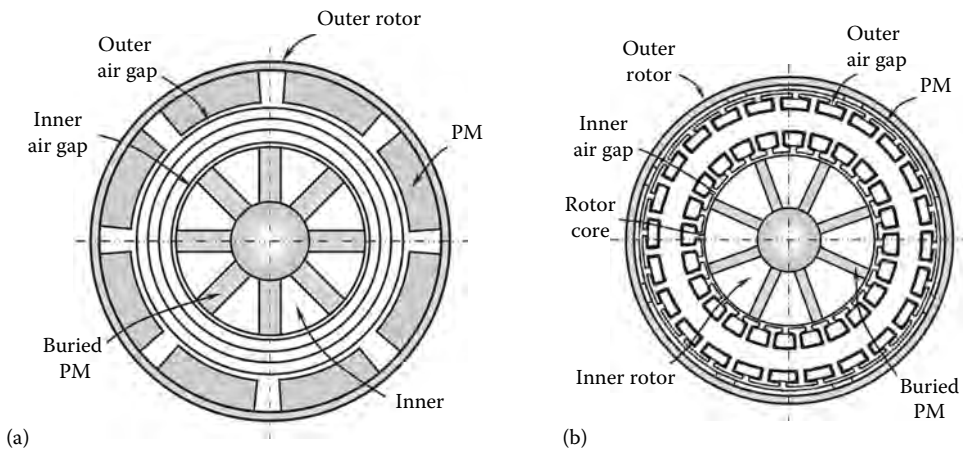
12.2 Radial-Flux Multirotor or Multistator Motor

RF multirotor and multistator motors have a number of distinctive advantages over conventional single-rotor and single-stator motors. By integrating multirotor or multistator into the same motor housings, the machines offer higher torque density and are usually more efficient.

12.2.1 Radial-Flux Multirotor Motor

A novel dual-rotor, toroidally wound PM motor was proposed by Qu and Lipo [12.11], as shown in Figure 12.3. There are some unique features in their design, including (1) a rotor–stator–rotor structure, (2) back-to-back windings, (3) very short end windings, (4) a high diameter-to-length ratio, (5) low air gap inductance, (6) high efficiency, (7) high torque density, (8) high overload capability, (9) balanced radial forces, (10) suitability for moderately high-speed performance, (11) low cogging torque, and (12) low material costs.

The motor illustrated in Figure 12.3a is constructed with a plurality of PMs mounted on the inner surface of the outer rotor and buried within a core of the inner rotor. The PMs buried within the inner rotor core preferably extend radially from a central opening of the inner rotor to the outer surface of the inner rotor. The inner and outer rotors

**FIGURE 12.3**

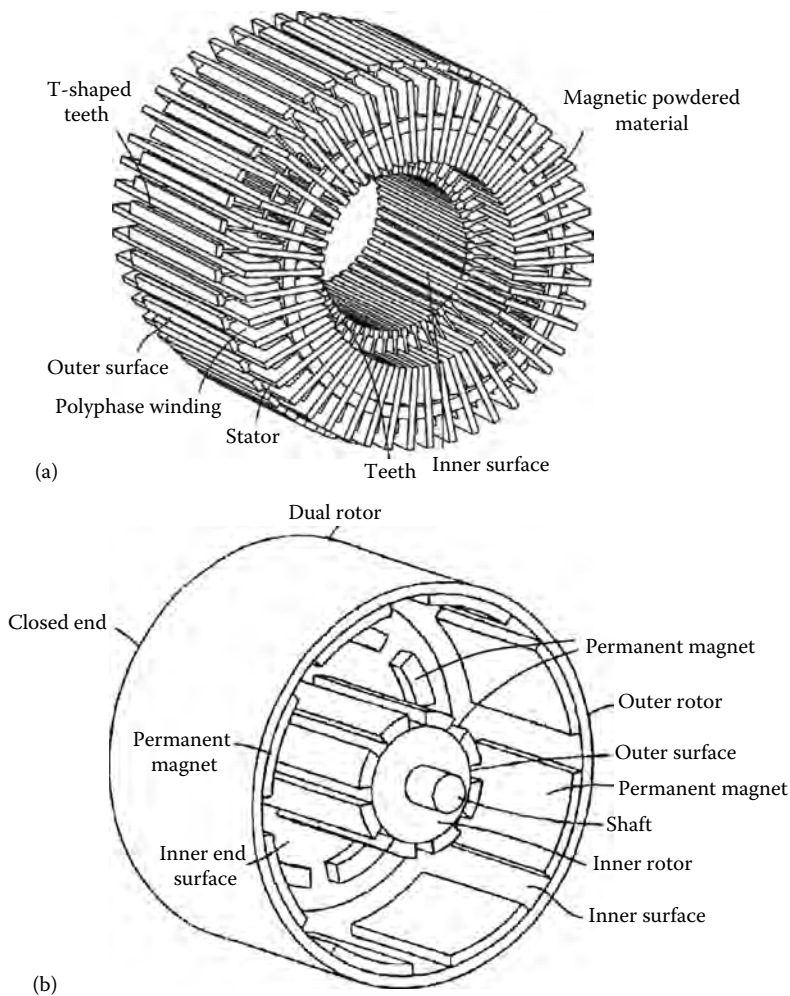
Structures of dual-rotor, toroidally round hybrid PM motors: (a) nonslotted machine and (b) slotted machine (U.S. Patent 6,924,574) [12.11]. (Courtesy of the US Patent and Trademark Office, Alexandria, VA.)

comprise a single integral rotor so that the inner and outer rotors rotate at the same speed. The torus-shaped stator has a hollow cylindrical shape and is preferably nested between the inner and outer rotors. A plurality of polyphase windings are toroidally wound around the stator. It is noted that the stator is nonslotted, meaning the stator does not include any radially extending teeth. A benefit of the nonslotted structure is that it avoids cogging torque. There are two air gaps in the motor. The inner air gap is formed between the inner rotor core and the stator, and the outer air gap is formed between the outer PMs and the stator.

The motor illustrated in Figure 12.3b has a similar configuration to the motor of Figure 12.3a, except for the stator structure. This motor employs a slotted stator rather than nonslotted in Figure 12.3a. In fact, the more the stator windings used in a motor, the higher the torque generated and the higher the motor efficiency. A higher torque density is thus achieved by slotting, but the slotted structure will introduce cogging torque.

The perspective view of the dual rotor and torus-shaped stator is shown in Figure 12.4. The dual rotor (Figure 12.4a) consists of an inner rotor, an outer rotor, and a closed end. The dual rotor magnetically interacts with the stator, whereby the PMs drive a magnetic field within the stator, causing a back electromagnetic force to be induced in the polyphase windings that are wound around the stator. The stator includes a plurality of T-shaped teeth or slots extending radially inwardly from the inner surface and outwardly from the outer surface of the stator (Figure 12.4b). The windings wound around the stator may comprise toroidally wound windings, lap windings, wave windings, or other types of windings. Since almost all of the windings on the inner, outer, and end surfaces of the stator are used for torque production, this motor creates higher efficiency and higher torque density than the conventional motors. The stator is formed by a plurality of stacked laminations that are connected together with a magnetic powdered material.

Figure 12.5 is a cutaway perspective view of the motor assembly. This figure clearly shows how the stator is nested between the inner and outer rotors. An additional gap (end air gap) is formed at the motor end, between the windings and PMs, which attached on the inner surface of the closed end (see Figure 12.4b).

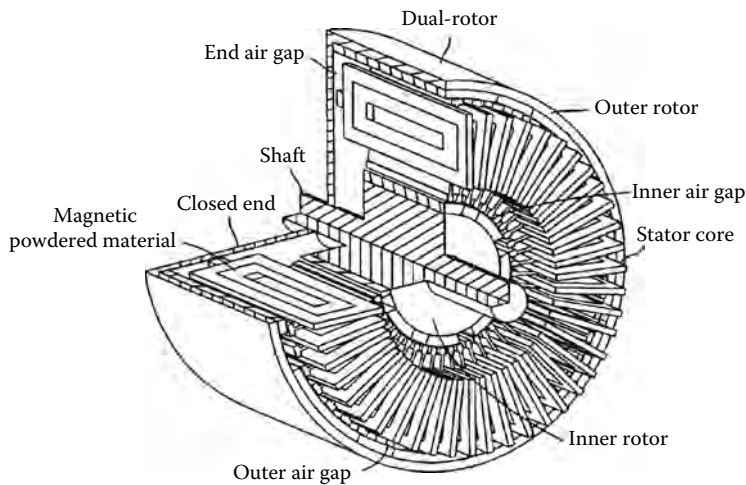
**FIGURE 12.4**

Dual-rotor, toroidally wound SPM motors: (a) stator and (b) dual rotor (U.S. Patent 6,924,574) [12.11]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

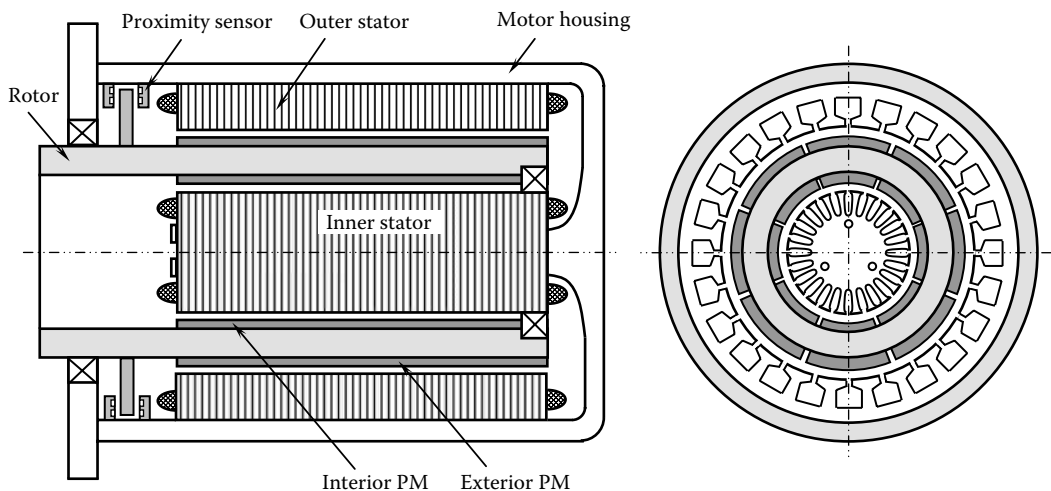
12.2.2 Radial-Flux Multistator Motor

The structure of an RF dual-stator motor is presented in Figure 12.6. The motor consists of an inner stator, an outer stator, and a rotor, which inserts between the two stators via two bearings. PMs are attached to the inner and outer surfaces of the rotor. In order to achieve the concentricity of the inner and outer stators, the two stators are integrated as one part. If the two stators are made separately, an accurate position control is required. Stator windings are wound through the inner and outer slots.

An alternative design is illustrated in Figure 12.7. In this design, PMs are mounted on the surfaces of the inner and outer stators. Windings are made through the slots of the rotor. The system is supported with three bearings.

**FIGURE 12.5**

A cutaway perspective view of the motor assembly (U.S. Patent 6,924,574) [12.11]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

**FIGURE 12.6**

RF, dual-stator, PM motor with two bearings.

12.2.3 Radial-Flux Brushless Double-Rotor Machine

In recent years, a new power-split hybrid system, called compound-structure PM synchronous machine (CS-PMSM) system, has been developed for replacing the mechanical planetary gear, flywheel, clutch, starting motor, and generator used in conventional automobiles. This is a promising technology for developing future hybrid vehicles with high efficiency and reliability. As part of the efforts, a novel RF brushless double-rotor machine (BDRM) has been proposed by Zheng et al. recently [12.12].

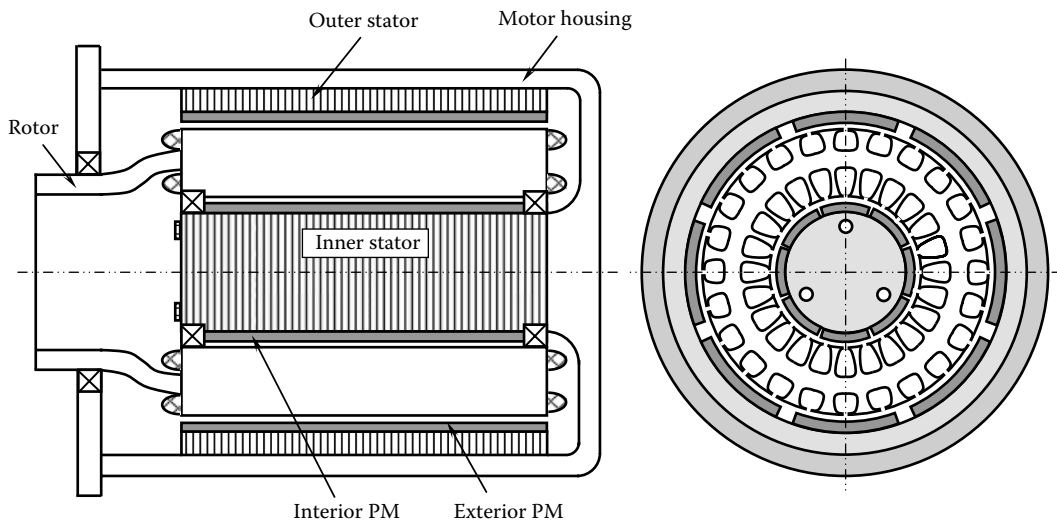


FIGURE 12.7
RF, dual-stator, PM motor with three bearings.

The CS-PMSM system is illustrated in Figure 12.8. It consists of a BDRM and a conventional machine. There are some design characteristics associated with the BDRM:

- The windings of the BDRM are mounted on the stator close to the motor case. This configuration helps dissipate generated heat from the stator windings to the environment.
- Stator phases are mutually independent with no couplings existing in-between.

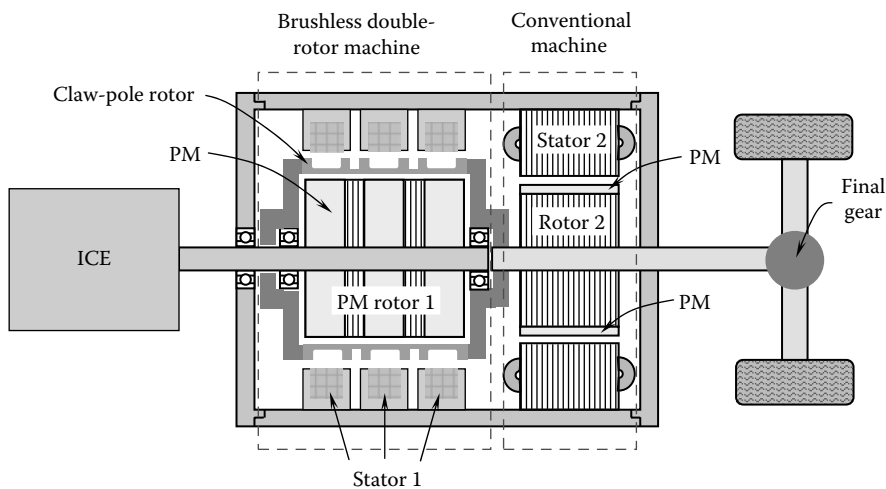
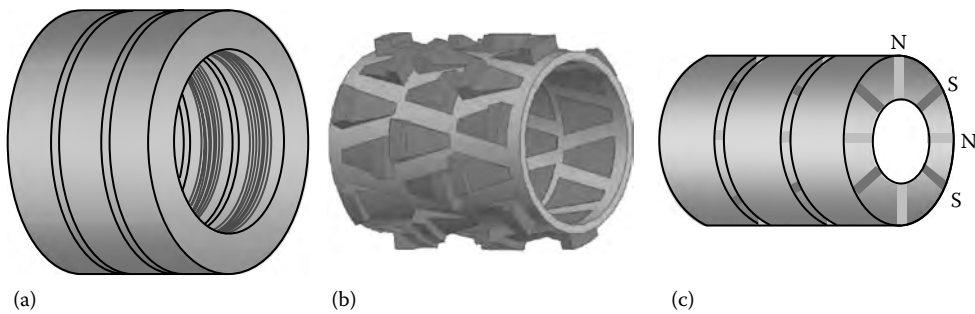


FIGURE 12.8
A CS-PMSM consisting of a BDRM and a conventional machine for driving hybrid electric vehicle.

**FIGURE 12.9**

Perspective view of the BDRM: (a) stator, (b) claw-pole rotor, and (c) PM rotor with buried radial magnets for azimuthal magnetization.

- The BDRM is designed as a multipole machine. This is especially suitable for intermediate or high-frequency operation.
- The ring-shaped stator windings of the BDRM simplify the winding fabrication and assembly process.

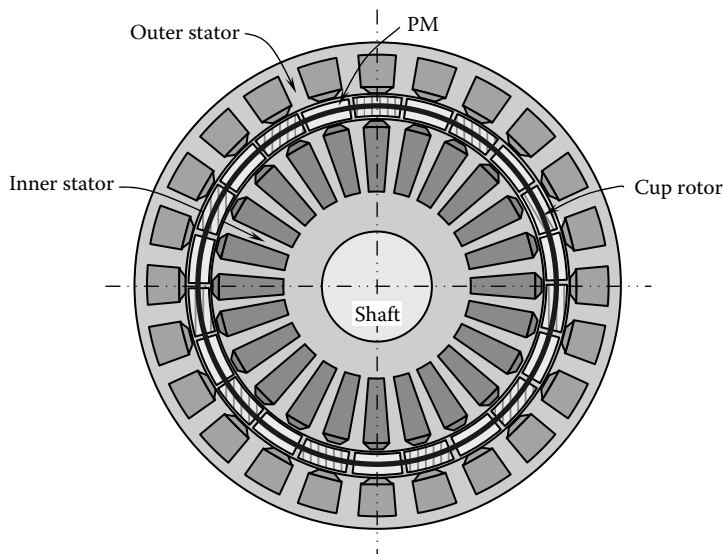
The structure of the BDRM is demonstrated in Figure 12.9. The stator shown in Figure 12.9a comprises laminated iron cores and ring-shaped stator windings. Figure 12.9b shows the claw-pole rotor that has three arrays of claws mounted on a nonmagnetic bracket. While the claw-pole rotor is connected to the final gear for driving wheels, the PM rotor 1 is connected to the crank shaft of the ICE. The function of the PM rotor is to transmit the ICE torque to claw-pole rotor. The speed adjustment between the claw-pole rotor and PM rotor 1 can be achieved by energizing the stator windings with different frequencies via an inverter. The PMs are burned radially through the rotor cores with the N-S-N-S arrangement, as presented in Figure 12.9c.

The claw-pole rotor is connected to the PM rotor 2 together with the conventional machine. Thus, the total output torque of the shaft is collectively contributed by the BDRM and conventional machine.

12.2.4 Radial-Flux Double-Stator PM Machine

The structure of an RF double-stator PM machine is presented in Figure 12.10 [12.13]. The cup rotor is nested between the inner and outer stators. This machine has 24 slots wound with three-phase windings in both the concentric outer and inner stators. The 22 pieces of PMs are mounted on the inside and outside surfaces of the rotor, where N-poles and S-poles PMs are arranged alternately along the rotor circumference. Both winding methods, that is, three-phase single-layer and three-phase double layer, can be applied to both the inner and outer stators.

This machine has the advantage that currents in both the inner and outer stators produce electromagnetic torque and both air gaps can deliver the output torque, thus improving the torque density and providing a high starting torque for hybrid electric vehicles. In addition, because of the nature of double-stator windings, the machine can flexibly change its connections, hence providing a constant output voltage for battery charging over a wide range of speed.

**FIGURE 12.10**

RF double-stator PM machine, where S-poles (shaded) and N-poles (unshaded) are arranged alternately along the rotor circumference.

12.3 Axial-Flux Multirotor or Multistator Motor

While RF motors dominate the motor market as mainstream for more than a century, AF motors have been developed in the last several decades and are attracting growing interests in recent years. AF motors may benefit from the increased efficiency, lowered axial loads acting on the rotor and bearings, and reduced motor vibration. Unlike in an RF motor in which the air gap is fixed after the rotor and stator are manufactured, the planar air gap in an AF motor is adjustable by changing the relative position of the rotor and stator.

AF motors are designed to have a higher power-to-weight ratio than RF motors, resulting in less core material and higher efficiency [12.14]. For small motors, the motor efficiency for an AF motor is about 15% higher than an RF motor with a similar size. This is attributed to the fact that the AF motor utilizes both sides of the rotor, while the RF motor utilizes only one side. Other advantages include simple cooling system, high reliability, and motor power being obtained from both ends of the spindle. Thus, the AF motors have been widely used in applications such as robots, wheel drives, hard disk drives, wind turbines, and fans.

There are many topologies of AF machines for various applications. For example, AF machines may be single-sided or double-sided, with or without slots, with or without armature core, with internal or external PM rotors, with surface-mounted or interior PM, and single stage or multistage [12.15]. Some of these topologies are addressed in the following sections.

12.3.1 Single-Sided and Double-Sided Axial-Flux Motors

A single-sided AF motor has the simplest structure with only single disk-shaped rotor and stator. In this configuration, PMs are mounted on one side of the rotor. The cross section of

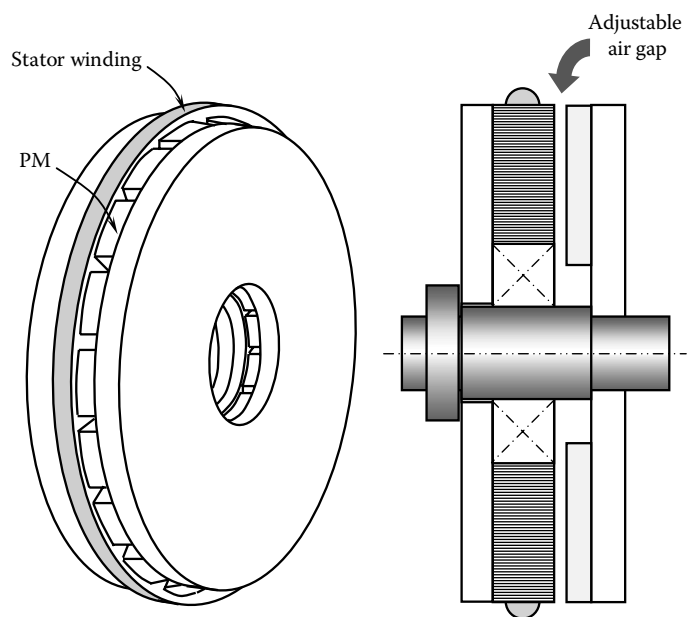


FIGURE 12.11
Single-sided AF PM motor, showing a single rotor and stator.

the rotor and stator is illustrated in Figure 12.11. The topology of AF motor allows to adjust the motor torque by adjusting the air gap, as shown in Figure 12.12.

A typical double-sided AF motor is shown in Figure 12.13. As illustrated in Figure 12.13a, the motor consists of two identical stators and one rotor, where the rotor is nested in the two stators axially. Each stator has a number of slots for receiving stator windings. The three-phase windings are uniformly distributed in the circumference of the stator and wound radially (Figure 12.13b). The PMs and iron poles are alternately embedded or glued in a disk-shaped

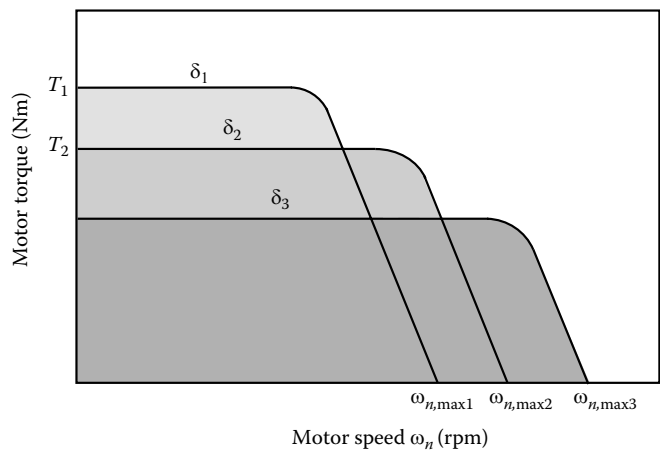
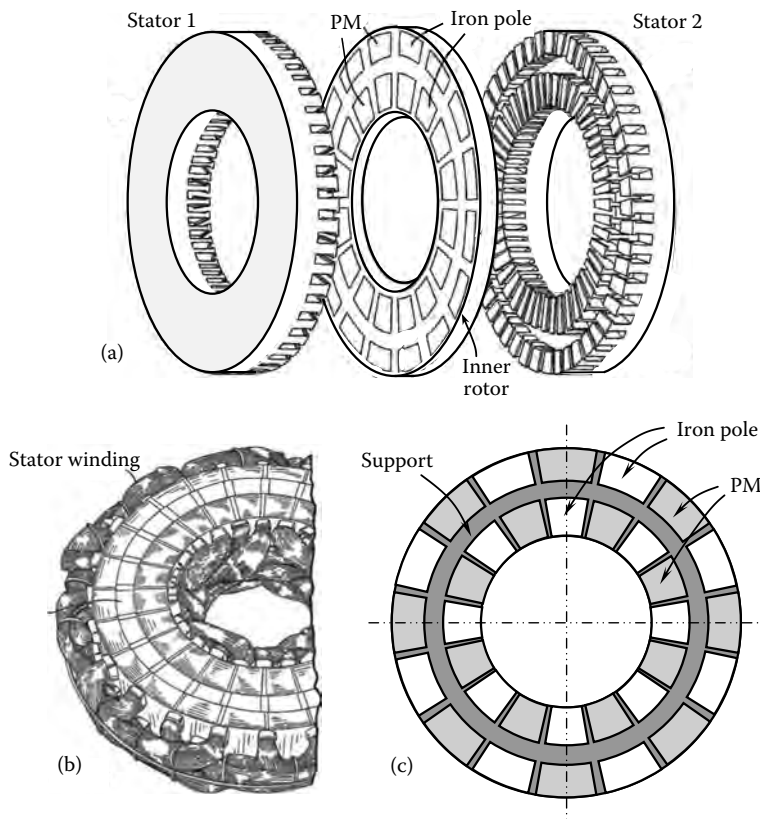


FIGURE 12.12
Adjustable motor torque by varying air gap δ in an AF motor.

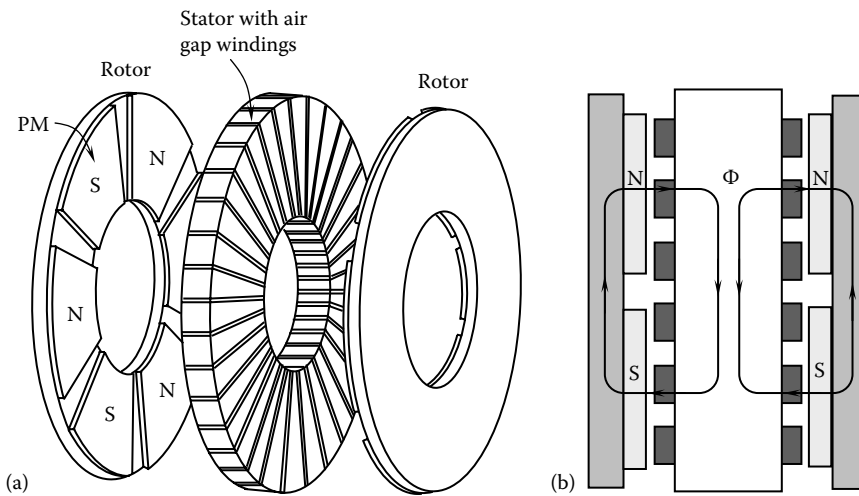
**FIGURE 12.13**

A stator-rotor-stator AF PM motor: (a) the motor contracture, (b) stator assembly, and (c) PM and iron pole arrangement on the rotor (U.S. Patent 7,608,965) [12.16]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)

support in two rows (Figure 12.13c). The support is made of a nonferromagnetic material such as aluminum and epoxy. [12.16]. The stators can be connected in series or parallel. A serial connection is usually preferred since it can provide equal but opposite axial forces.

The double-sided AF motors can have different configurations such as a stator-rotor-stator structure as addressed previously or a rotor-stator-rotor structure as shown in Figure 12.14. In the rotor-stator-rotor design, a single nonslotted stator is sandwiched between two rotor disks, forming axially two planar air gaps. The stator is realized by tape wound core with polyphase AC air gap windings that are wrapped around the stator core with a back-to-back connection. The active conductor portion is the radial portion of the toroidal windings facing the two-rotor structure (Figure 12.14a). Arch-shaped PMs are mounted axially on the inner surface of each rotor.

The magnetic flux path is presented in Figure 12.14b. An N-pole magnet on one rotor is placed directly opposite to its counterpart on the other rotor. It is noted that there are two identical magnetic loops. For each loop, the magnetic flux is driven by the N-pole magnet to go across the air gap into the stator core, travels circumferentially along the strip-wound stator core, returns across the air gap, and then enters the rotor core through the opposite S-pole of the PMs to complete the magnetic flux loop. Hence, it can be expected that the

**FIGURE 12.14**

An AF torus-type slotless rotor-stator-rotor PM motor: (a) motor structure and (b) magnetic flux direction.

axial length of the stator core would be quite long because of the summation of the flux entering the stator from both rotors. As a matter of fact, this machine can be viewed as a combination of two independent halves due to the direction of the magnetic flux.

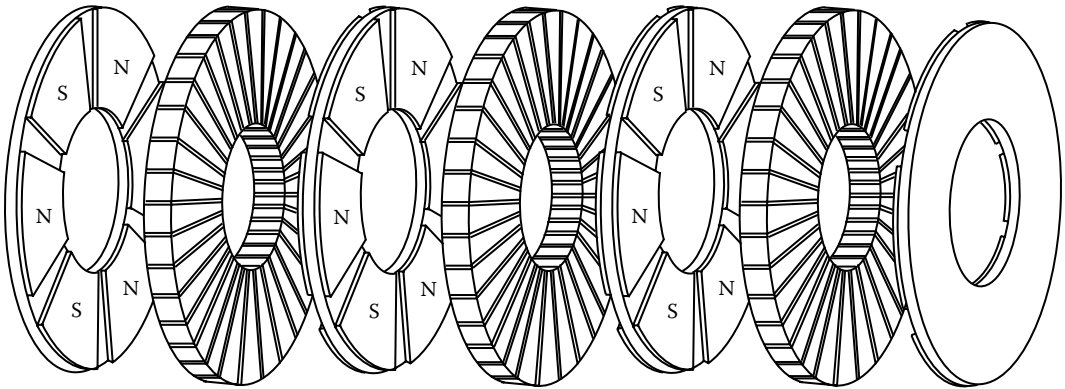
The slotless torus topology shown in Figure 12.14 has a high torque density and high torque-to-weight ratio. The portions between the air gap windings are assumed to be filled with epoxy resin as in all nonslotted structures in order to increase the mechanical robustness and provide better conductor heat dissipation. In addition, in the torus topology, the windings in the air gap are used for the torque production. Besides, due to the absence of stator slots, the cogging torque, magnetic flux ripple, pulsation losses in the rotor, and acoustic noise can be greatly reduced.

12.3.2 Multistage Axial-Flux Motor

For an AF motor, the torque is proportional to the OD of the motor. However, the centrifugal force acting on the rotating components significantly increases with the OD. Therefore, a common way to increase the motor torque is to design a motor with multistage arrangements. Usually, for a rotor-stator-rotor structure, a multistage machine has N stators and $N + 1$ rotors, where N is the number of stages. All rotors share the same shaft.

The double-sided AF motor in Figure 12.14 can be extended into a multiple stage machine as illustrated in Figure 12.15. The stators are formed by wrapping magnet wires radially around a nonslotted stage core. The rotors are formed by both rotor cores and axially magnetized SPM. Except for the end rotors, PMs are mounted on both sides of the rotor cores. Theoretically, there is no limit on the number of motor stage in multistage motors. This indicates that a desired high torque can be achieved by adjusting the number of motor stages.

The multistage AF machines are very competitive with respect to conventional AF and RF machines for their compact size, simple structure, and easy assembly. Because each air gap can be adjusted independently, the air gap depths in the motor can be made either identical or distinctive for obtaining the optimized motor performance for a specific application.

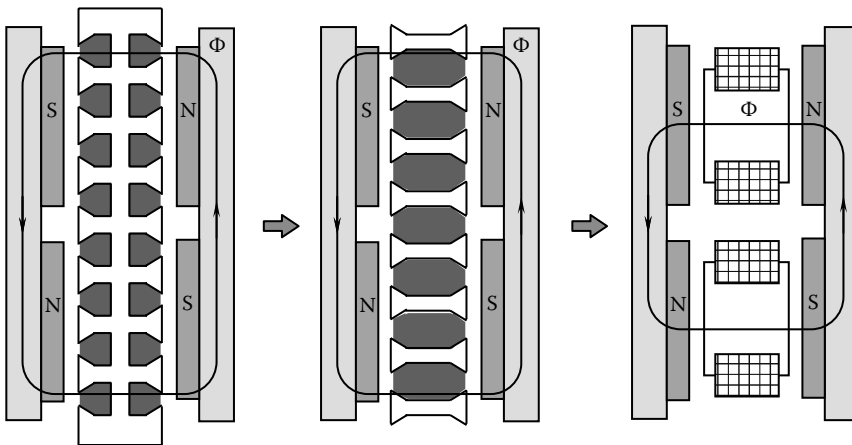
**FIGURE 12.15**

Perspective view of multistage of AF torus-type slotless PM motor.

12.3.3 Yokeless and Segmented Armature Motors

The yokeless and segmented armature (YASA) topology is a new type of AF motor. This type of motor has shown an improved torque density and efficiency compared to other AF motors [12.17]. The iron in the stator of the YASA motors is dramatically reduced, typically by 50%, causing an overall increase in torque density of about 20%. In fact, the elimination of yoke, elevated fill factor, and shortened end windings all contribute to the increase of torque density and efficiency of the electric machine. Therefore, this topology is highly suited for high-performance applications.

The YASA motors are primarily designed for vehicles. The segmented stator is made of powdered iron materials. The topology of a YASA motor combines two of the best-performing AF machines, the NS torus-S and the NN torus-S topologies, as shown in Figure 12.16.

**FIGURE 12.16**

The topology of YASA motor, which combines two best-performing AF motors.

12.4 Hybrid Motor

A hybrid motor is the motor that combines two or more types of motor structures together so as to integrate their strengths for a better performance and operation.

12.4.1 Hybrid Excitation Synchronous Machine

Hybrid excitation synchronous machines (HESMs) have been developed in recent years for traction applications such as hybrid electric and fuel cell vehicles [12.18]. The HESMs have two excitation sources: one is the PM source that provides the air gap with constant flux and the other is the DC excitation coil source that acts as the flux regulator to adjust the air gap flux distribution. These two excitation sources can be connected either in series or in parallel.

As can be seen in Figure 12.17, a HESM consists of powdered iron cores, PMs, stator windings, and DC excitation coils. PMs are placed at the center of the rotor and the excitation coils located at the two ends of the motor. Thanks to the excitation coils, it is possible to modulate the total excitation flux in the air gap region of the machine. In addition, these additional excitation coils make it possible to improve the effectiveness of the machine in different operating regions.

12.4.2 Hybrid Hysteresis PM Synchronous Motor

A hysteresis synchronous motor (HSM) is extensively adopted in small motor applications. The combination of PMs and hysteresis materials in the rotor of an HSM has many distinct advantages over the conventional PM or hysteresis motors. The hybrid motor in which

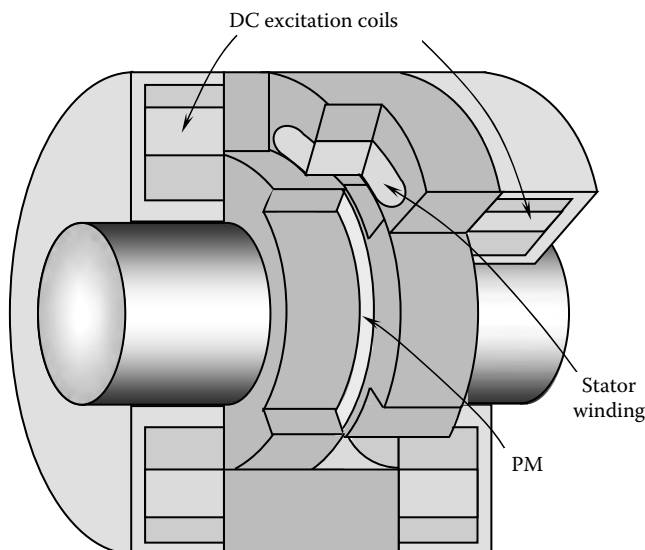
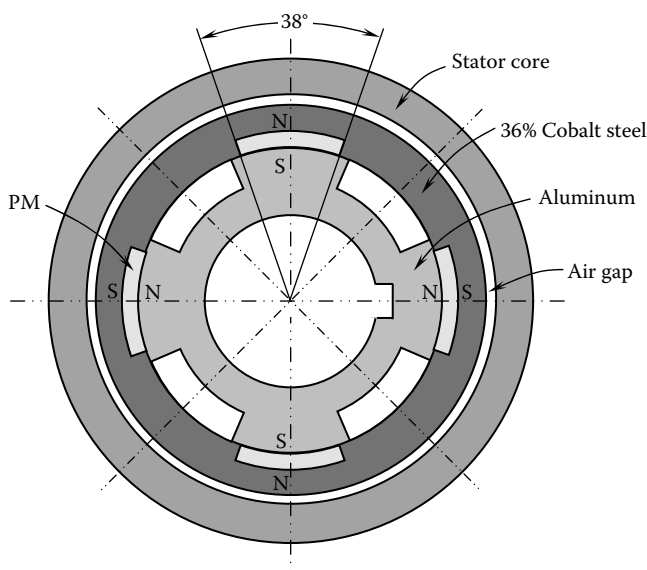


FIGURE 12.17

HESM with powdered iron core and global excitation coils.

**FIGURE 12.18**

The structure of a hybrid PM HSM. The combination of PMs and hysteresis materials in the rotor makes the motor suitable for constant-torque and constant-speed applications.

the PMs are inserted into the slots at the inner surface of the hysteresis ring is called PM hysteresis synchronous (PMHS) motor. In fact, this type of motor combines the advantageous feature of both the hysteresis and PM motors [12.19].

Qin and Rahman [12.20] have presented a hybrid hysteresis PM synchronous motor, as illustrated in Figure 12.18. The rotor of the motor consists of a 36% cobalt-steel hysteresis alloy with rare-earth PMs to improve the overall performances of such a motor. At asynchronous speeds, the motor torque contains the hysteresis torque, eddy-current torque, and PM brake torque. At the synchronous speed, the motor torque comprises the hysteresis and PM torques. Experimental results have confirmed the validity of the motor design and shown that the hybrid motor exhibits improved speed stability over a wide range. This type of motor is especially suitable for constant torque, constant speed, and quiet applications such as gyros, time and recording equipment, and pumps.

12.4.3 Hybrid Motor Integrating Radial-Flux Motor and Axial-Flux Motor

The motor shown in Figure 12.19 integrates an RF and an AF motor together to achieve high torque density and efficiency. This motor has three sets of PMs and three stators, including one regular radial stator and two axial disk-shaped stators. The two sets of PMs are mounted on the rotor's two sides and polarized in the axial direction. Thus, the north/south polarity of the PMs alternates as the rotor rotates. The third set of PMs is mounted on the curved radial outward surface and polarized in the radial direction with alternating orientation of their N and S poles. The rotor is disposed around the shaft. Thus, the regular radial stator and the third set of PMs form a regular RF motor, and the two axial stators and the two sets of PMs mounted on the sides of the rotor form an AF motor.

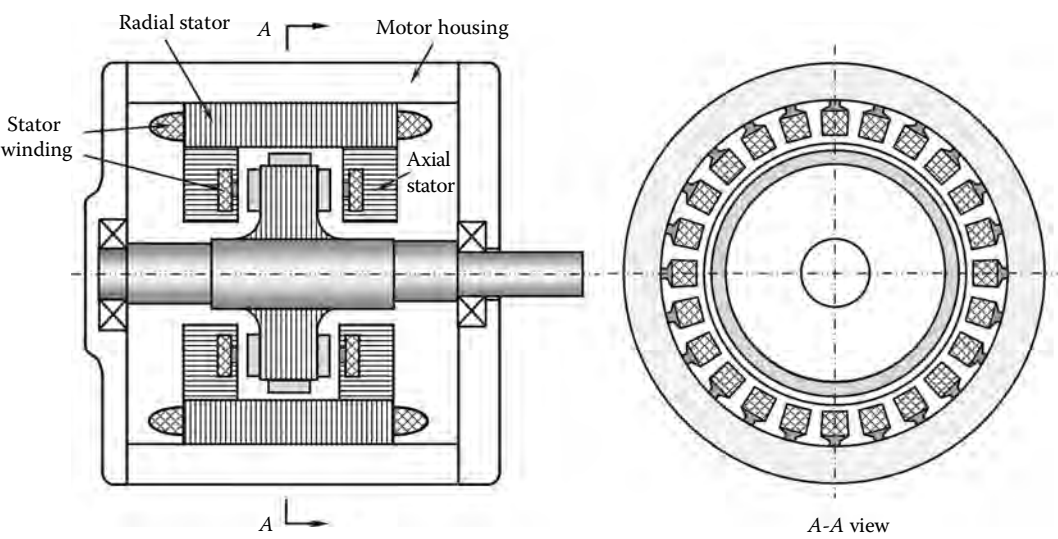


FIGURE 12.19
The new motor design that combines the RF and AF motor.

Electric currents applied to the three stators produce rotating magnetic fields that interact with the fields produced by the corresponding sets of PMs to generate a desired torque that rotates the motor shaft. For a given motor volume, this motor offers high mechanical power and torque since the entire rotor surface is used in the torque production. Typically, the motor rating can be increased one-third or greater with respect to a comparably sized RF motor [12.21].

The motor with a single-unit structure in Figure 12.19 can be expanded into a multistage form, as shown in Figure 12.20. Generally, the output power is proportional to the motor

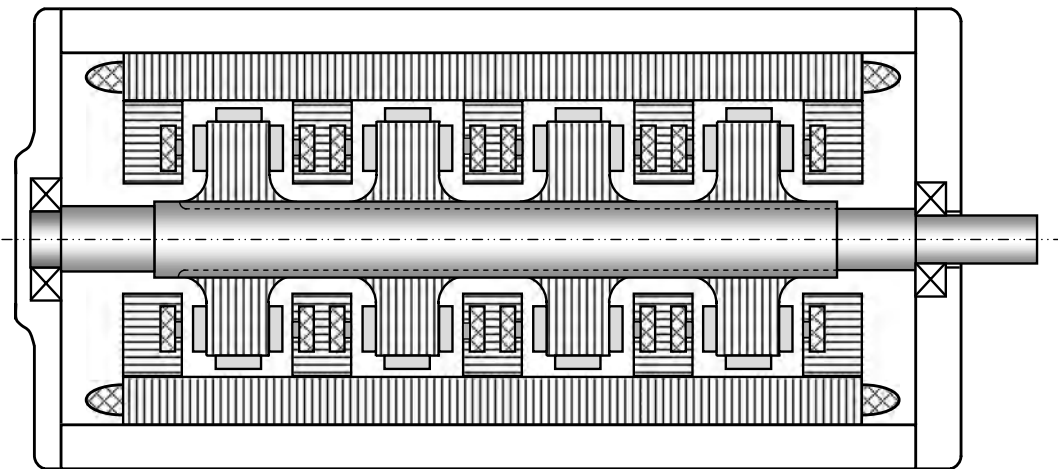


FIGURE 12.20
The motor with multistages as the integration of RF and AF motor.

OD and the stage number. Increasing the stage number can thus increase the motor torque to drive an external loading machine.

12.4.4 Hybrid-Field Flux-Controllable PM Motor

The topology of a three-phase 6/4-pole hybrid-field flux-controllable PM (FCPM) motor is presented in Figure 12.21. The main advantage of this type of motor is that the air gap flux density is directly controllable. The stator contains two types of windings, that is, the three-phase AC armature windings and the DC field windings. The key of the motor is to incorporate both PMs and DC field windings in the stator, hence offering a compact arrangement of hybrid-field excitations, while the rotor is simply composed of salient poles without winding or PMs [12.22]. PMs are closely positioned adjacent to the field windings. In such a way, the field windings serve to regulate the PM flux, offering either flux weakening or flux strengthening.

It is important to note that there is an extra air bridge in shunt with each PM. If the field winding MMF reinforces the PM MMF, the air bridge will amplify the effect of flux strengthening. On the contrary, if the field winding MMF is opposite to the PM MMF, it will increase the PM flux leakage, thus amplifying the effect of flux weakening. By choosing an appropriate width of the air bridge, a wide flux regulating range can be obtained by using a small DC field excitation.

12.4.5 Hybrid Linear Motor

Jeon et al. [12.23] proposed a novel hybrid linear motor (HLM) by both induction and synchronous operations. The motor consists of a pair of linear synchronous motors (LSMs) and a linear induction motor (LIM). The primary cores of both motors share a common ring winding, and the secondary solid conductor is arranged in both LIM and the inter-pole space of LSM. From the 3D FEA and experiment, the motor is verified to be effective for practical use.

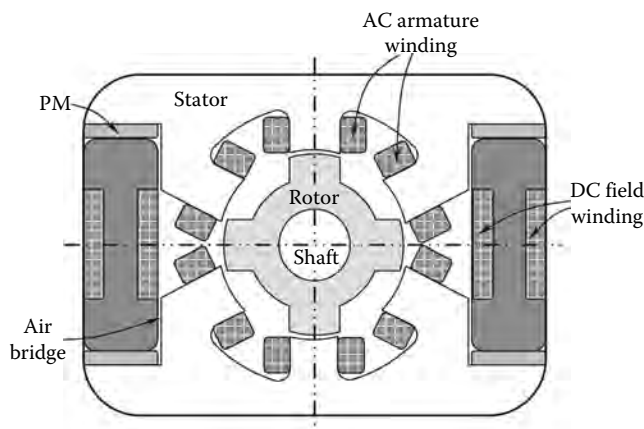


FIGURE 12.21

Topology of a three-phase 6/4-pole hybrid-field flux-controllable PM motor.

12.5 Conical Rotor Motor

In the present motor industry, almost all rotary motors are designed with a cylindrically shaped rotor and stator, so that a constant air gap is maintained in both the circumferential and axial directions. In a conical rotor motor, the rotor is in a conical shape. According to the stator shape, two groups are identified as follows:

1. The stator is in a regular shape with a constant internal diameter. Therefore, the air gap between the rotor and stator is no longer a constant; rather, it varies along the motor axial direction (Figure 12.22). It can be seen in the figure that the air gap increases linearly along the motor axis away from the bearing. This configuration may improve the motor vibrational characteristics and prevent the interference between the rotor and stator [12.24].
2. The stator is also in a conical shape, with the same conical angle as the rotor. Thus, the air gap remains a constant along the rotor axis (Figure 12.23). Sorros et al. [12.25] have designed, analyzed, and tested two micromotors, with a conventional cylindrical design and a conical design, respectively. They found that the conical motor offers comparable performance to the cylindrical motor but with a much more compact mechanical arrangement. One of the advantages of the conical design is that it eliminates the need for a long pivot bearing and hence substantially reduces the size of the motor. The high starting torque and low inertia of the conical motor have proven to be ideal for the demands of high cycle dynamic drives in various applications.

A new type of conical rotors with a variety of rotor–stator structures was proposed by Burch et al. [12.26]. As an example shown in Figure 12.24, unlike the conical rotor motors discussed previously, the rotor assembly consists of two conical rotor cores that are mounted on the motor shaft. Trapezoid magnets are attached on the conical surface of each conical core for forming the magnet field.

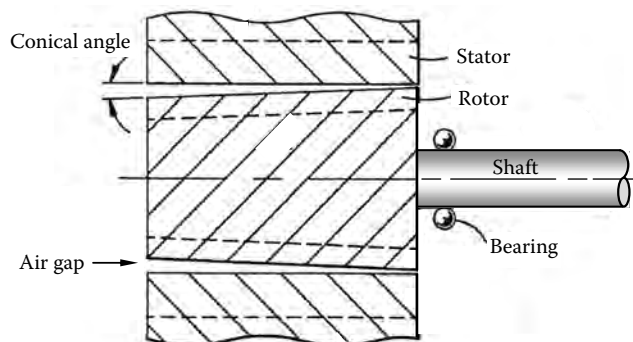


FIGURE 12.22

Conical rotor with a nonuniform air gap (U.S. Patent 5,233,254) [12.24]. (Courtesy of the U.S. Patent and Trademark Office, Alexandria, VA.)