# A MOTOR PRIMER
## Part 4


Paper No. PCIC2003-16

**Bryan Oakes**
**Member IEEE**
**Reliance Electric**
**101 Reliance Road**
**Kings Mountain, NC 28086**
**USA**

**Gary Donner**
**Senior Member IEEE**
**Shell - Los Angeles Refining Co**
**2101 E Pacific Coast Hwy**
**Wilmington, CA 90748**
**USA**

**Steve Evon**
**Member IEEE**
**Reliance Electric**
**6040 Ponders Ct.**
**Greenville, SC 29615**
**USA**

***Abstract:*** In recent years much has been written about motors on Variable Speed Drives, high speed rigid shaft motors, impact of the American Petroleum Institutes (API) Standard 541 and motor diagnostics. Most of these papers and articles assume that the reader has significant knowledge of motor theory and operation. However, this assumption is overly optimistic, considering that only a few colleges teach motor theory today and that application experience at motor user locations has been reduced in recent years.

**Index Terms** – AC Induction Motors, Sleeve Bearings, Lateral Critical Speed, Ambient Temperature, Altitude, Starting, Amplification Factor

### INTRODUCTION

This paper is the fourth in a series of papers where the authors provide answers to "Frequently asked Questions" that are routinely asked by working engineers in industry. The authors will present motor theory and application information with an extensive reference list that will help working engineers increase their general understanding and knowledge of motors. This series of papers also serves as a valuable reference for those who apply and specify motors. This paper focuses on sleeve bearing operation, lateral critical speeds, ambient conditions and motor starting. The first three papers in this series can be found in the IEEE Conference Records and Transactions.

### I. WHAT ARE SLEEVE BEARINGS?

The term "sleeve bearing" is typically used to describe hydrodynamic journal bearings. The basic purpose of a hydrodynamic journal bearing is to provide a near frictionless environment to support and guide a rotating shaft. In this type of bearing, the relative motion between the rotating shaft "journal" and stationary bearing bore, with adequate lubrication delivered by oil rings or forced lubrication, will generate a film of lubricant (lubricant wedge) between the rotating and stationary elements of the bearing. A pressure distribution is developed in this film that separates the rotating and stationary elements of the bearing. Starts and stops will produce some wear, as the motor requires a certain speed to develop a proper oil film. Properly installed and maintained, journal bearings should have essentially infinite life.

Hydrodynamic radial bearings are typically designed to be split for ease of assembly, precision bored, and of the sleeve or pad type, with steel or bronze backing and babbitted replaceable liners or pads.

Fig. 1 - Hydrodynamic Oil Film Bearing

Applications where a hydrodynamic film cannot be developed due to speed or load conditions a hydrostatic bearing design may be used. Hydrostatic journal bearings are a special type of "sleeve bearing". Hydrostatic bearings require the lubricant pressure to be provided from an external source. The lubricant is supplied to the bearing under pressure and is injected into the clearance between the rotating and stationary elements to lift the rotating element from the stationary bearing bore. The lubricant injection port is situated in the gravity-loaded region of the bearing. Hydrostatic bearings can support very large loads even at zero rotational speed, as there is always full film lubrication. This type of bearing has been limited to specialized applications such as very slow speed and reversing applications but is now finding its way into everyday usage.

## HYDRODYNAMIC vs. HYDROSTATIC ADVANTAGES/DISADVANTAGES

The advantages of using a hydrodynamic bearing include the following:

- Split for ease of inspection and replacement.
- Can be re-babbitted.
- Simple system.

The advantages of using a hydrostatic bearing include the following:

- Split for ease of inspection and replacement.
- Can be re-babbitted.
- Can have high damping and stiffness parameters.
- Can operate at extremely low speed and through reversals.

The disadvantages of using a hydrodynamic bearing include the following:

- No interchangeability from brand to brand.
- No warning of impending failure unless bearing temperatures and vibration are monitored.
- Not adequate for extremely low speed and reversing applications.
- Generally not designed to handle radial loads.

The disadvantages of using a hydrostatic bearing include the following:

- Requires high-pressure lube system.
- Expensive.
- Complex system.

## BASIC OPERATING CHARACTERISTICS

There are five basic operating characteristics that need consideration during the initial design of a bearing system. These are:

a. Minimum oil film thickness
b. Maximum bearing temperatures
c. Speed range
d. Loading
e. Stiffness and damping characteristics

Depending on the operating speed of the motor one of these characteristics may play a more important role over the others.

## MINIMUM OIL FILM THICKNESS

The magnitude of the minimum film thickness is one of prime importance in the design and operation of a hydrodynamic oil film bearing. A fully separating oil film is required for satisfactory operation.

Hydrodynamic principles show the oils adhesion to the journal and its resistance to flow, is dragged by the rotation of the journal so as to create a wedge-shaped film between the journal and bearing bore surface. This shearing action sets up the pressure in the oil film that supports the load within the bearing clearances. In a horizontally split bearing the oil wedge will lift and support the shaft, relocating the centerline slightly up and to one side into a normal attitude position in a lower quadrant of the bearing.

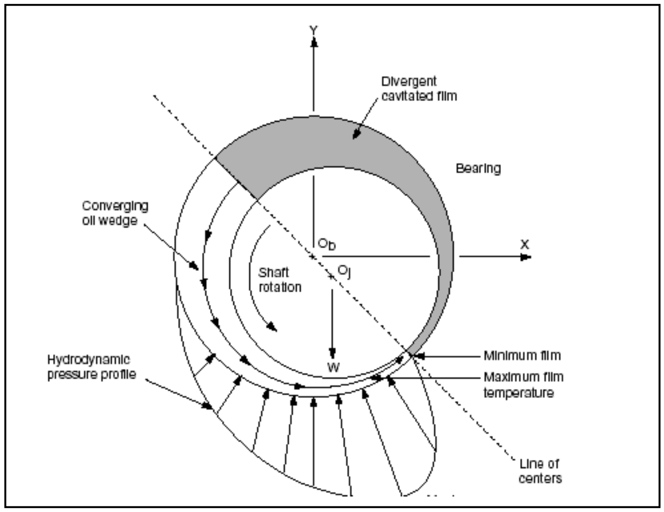



Fig. 2 - Oil Wedge Formation

The hydrodynamic oil film bearing passes through three lubrication conditions during starting and shutdown. The first condition is known as boundary lubrication. Under boundary conditions the bearing and shaft surfaces rub together with only a thin oil film separating the two. As the velocity of the shaft increases mixed film lubrication develops where the bearing supports part of the load on a boundary film where the shaft is closest to the bearing. The remainder of the load is supported by hydrodynamic lubrication. Under hydrodynamic lubrication a full film wedge develops that separates the shaft from the bearing. The full film carries the entire load of the rotating assembly with no metal to metal contact.

The initial boundary conditions are the driving components for selecting the bearing face material. The materials are chosen so that under initial starting conditions or a shutdown there is no surface damage to the load carrying area of the bearing or shaft. One of the more commonly used materials for bearing liners is a tin-based babbitt. The softness of the babbitt provides outstanding embeddability and resistance to seizure. The property of embeddability relates to the ability of the babbitt material to hold contaminants in the bearing thus preventing damage to the rotating journal surface.

The minimum oil film thickness is one operating characteristic that is predicted by calculation and is then limited to not fall below a certain value. The minimum film thickness depends on the surface finish of the journal and the bearing, as the film must be thick enough to eliminate contamination. The recommended surface finish for a ground journal is 16 to 32 microinches. For ground journals the minimum thickness of this film can be estimated by the following relationship.

$h_o = 0.00025 \times D$

| | | |
|---|---|---|
| Where: | $h_o$ | Minimum film thickness (in). |
| | D | Bearing Bore Diameter (in). |

## MAXIMUM BEARING TEMPERATURES

The temperature of the bearing needs to be considered from two standpoints:

a. Maximum temperature of the lubrication
b. Maximum temperature of the bearing babbitt

The friction generated in an oil film bearing application is important because the higher the coefficient of friction the higher the heat generation. Excessive heat reduces the oil life. Excessive heat may also result in expansion of the shaft, housing or bearing. This expansion reduces the clearance between the shaft and bearing, further increasing the operating temperature, resulting in premature bearing failure. The friction developed within the oil film also affects the efficiency of the machine. The heat generated due to friction must be properly transferred from the bearing and dissipated into an oil sump.

The highest temperature region is along the axial centerline of the bearing offset slightly from the load line. API-541 limits the total overall bearing temperature to 93°C. Built in temperature sensors are recommended to monitor babbitt temperatures under all operating conditions. API-670 states the resistance temperature detectors (RTDs) should be installed. The detector tips should be mounted within the bearing backing material and be positioned as near to the center of the bearing as possible to obtain the most accurate measurements. A detector mounted near the outside edge of the bearing does not provide a true representation of the overall bearing temperature during operating conditions.

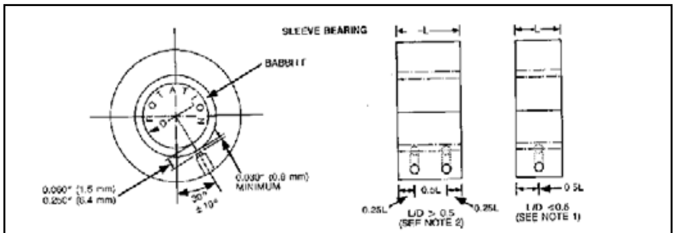



Fig. 3 - RTD Location within Bearing

## OPERATING SPEED AND LOAD

Hydrodynamic bearings rely on the creation of a pressurized film of oil sufficient to carry the load of the bearing with the oil film being generated by the motion of the rotating assembly within the bearing. In some applications motors operating on reciprocating and variable frequency drive conditions may operate at very slow speeds. Under these conditions due to very low rotational speeds an adequate oil film cannot be developed and maintained. If the bearing loading is high this too may prevent the formation of an adequate oil film to support the load. In these cases the hydrostatic bearing design would be a more suitable bearing choice.

## STIFFNESS AND DAMPING CHARACTERISTICS

Hydrodynamic oil film bearings can have a significant effect of the machinery's dynamic characteristics, as the oil film acts as a complicated set of springs and dampers and is a major factor in reducing vibration.

The hydrodynamic oil film bearing provides much more damping and less stiffness than a rolling element bearing because of the lubricant present. More viscous and thicker lubricant films provide higher damping properties and lower stiffness. The damping properties of the lubricant provide an excellent medium for limiting vibration transmission. Vibration measurements taken at the bearing housing will not represent the actual vibration on the rotating assembly within its bearing clearances. The recommended practice to monitor overall vibration conditions for this bearing type is to use proximity probes. Proximity probes measure the relative motion between the shaft and the housing and provide a better picture of overall motor vibration performance.

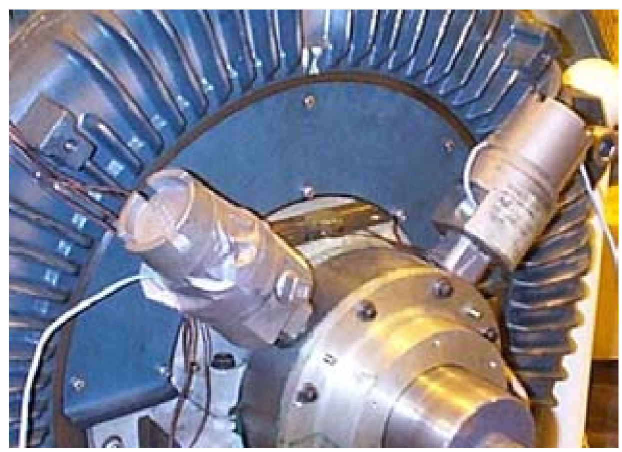


Fig. 4 - TEFC Motor with Proximity Probes Mounted 45 degrees from Vertical

## II. WHAT IS A LATERAL CRITICAL SPEED?

Rotors have natural frequencies of vibration. When the rotor is spun at this frequency, it begins to flex and go into a condition of resonance. These rotor resonances occur at what's called lateral critical speeds. A significant number of motors are designed to operate above their first bending lateral critical speed (Fig. 5) especially with the gaining popularity of variable frequency drive machines. There are other instances where (due to low base support stiffness) machines are operating with a lateral critical speed near operating speed.

``````````
``````````

The definition of a lateral critical speed differs between that defined for induction motors and the one defined for pumps. The primary specification for induction motors in the petroleum industry API-541 defines a lateral critical speed as a speed corresponding to resonant frequencies of the complete rotor, bearings, and bearing support system. The basic identification of critical speeds is made from the natural frequencies of the system and of the forcing phenomena. If the frequency of any harmonic component of a periodic forcing phenomenon is equal to or approximates the frequency of any mode of rotor vibration, a condition of resonance may exist. If a resonance condition exists at a finite speed, that speed is called a critical speed. API-541 requires that such a frequency response be removed from running speed by at least 15%.

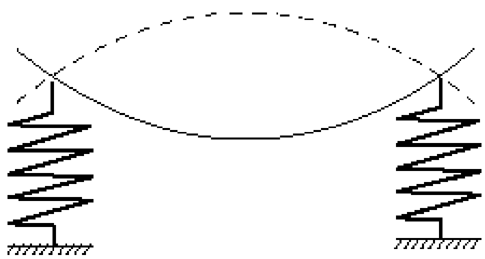


Fig. 5 - First Mode - Bending Mode

Pump manufactures have adopted the definition of a lateral critical speed defined in API-684. Here a lateral critical speed is defined as a shaft rotational speed that corresponds to a non-critically damped (amplification factor greater than 2.5) rotor system resonance frequency, see Figures 6 and 7. The frequency and amplitude are determined through damped unbalance response analysis and testing.

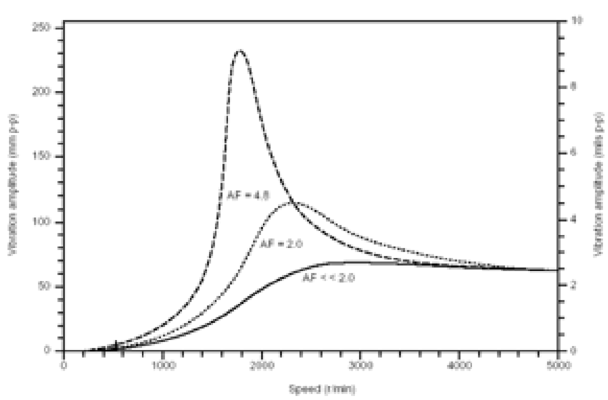
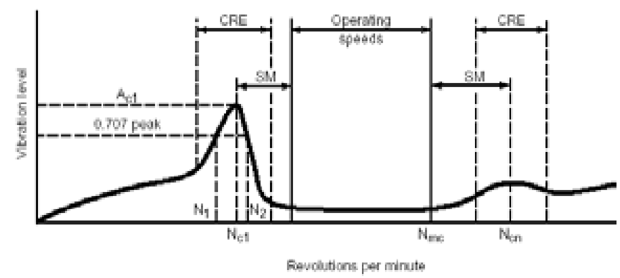



Fig. 6 Evaluating Amplification Factors from Speed-Amplitude Plots

### III. WHAT IS AMPLIFICATION FACTOR?

Amplification factor is the measure of a rotor bearing system's vibration sensitivity to unbalance when operated within the vicinity of a lateral critical speed. The amplification factor is measured by the ratio of the vibration amplitude at a resonance to the amplitude at speeds above the resonance. Amplification factors greater than ten indicate that rotor peak vibration will be approximately ten times higher than the vibration level above the resonance. An amplification factor less than five indicates the system is not sensitive to unbalance when operating near a critical speed. The effect of the amplification factor on rotor response near the associated critical is presented in Figure 7.

A method for determining the amplification factor of a machine is to use the amplitude part of a 1x-filtered Bode plot. The Bode plot shown in Figure 7, is a graph of 1x amplitude versus speed and phase versus speed.




$N_{c1}$ = Rotor first critical, center frequency, cycles per minute.
$N_{cn}$ = Critical speed, nth.
$N_{mc}$ = Maximum continuous speed, 105 percent.
$N_1$ = Initial (lesser) speed at 0.707 × peak amplitude (critical).
$N_2$ = Final (greater) speed at 0.707 × peak amplitude (critical)
$N_2 - N_1$ = Peak width at the half-power point.
AF = Amplification factor:

$$= \frac{N_{c1}}{N2 - N1}$$

SM = Separation margin.
CRE = Critical response envelope.
$A_{c1}$ = Amplitude at $N_{c1}$.
$A_{cn}$ = Amplitude at $N_{cn}$.

Fig. 7 - Bode Plot

API-541 has established vibration limits for a rotating assembly passing through a critical speed during an overspeed coastdown with a defined unbalance weight attached to the rotating assembly. API-541 states that the machine shall be run to 120 percent of its rated speed with the unbalance weights attached and then allowed to coast to rest. The shaft vibration relative to the bearing housing shall be observed. Machines with a defined 15% separation margin shall meet the following criteria:

The shaft displacement relative to the bearing housing at any speed within the operating speed range or separation-margin limits shall not exceed the smaller of the following value or 55 percent of the shaft-to-bearing and seal diametrical running clearances:

$D_s = 1.5*(12000 / N)^{1/2}$

Where: $D_s$ shaft displacement, (mils peak to peak).
N operating speed nearest the resonant speed of concern (rpm).

API-684 utilizes the amplification factor as a basis for pass/fail criteria for an unbalanced coastdown test.

$AF = N_{C1} / (N_2 - N_1)$

Where: AF Amplification Factor
$N_{C1}$ Rotor 1st Critical, Center Frequency (rpm)
$N_2 - N_1$ Peak Width at the Half Power Point

The actual proximity probe locations relative to the displacement of the rotating assembly may provide inadequate information regarding the true resonance amplitude response.

Depending on the probe location within the bearing housing the measured amplitudes or amplification factor may or may not meet the specified requirements. How can this be? The bearing acts as a node point with shaft deflection at the node point being zero. If the proximity probes are mounted inboard of the bearing the deflection value will be greater than probes mounted outboard of the bearing. An inboard mounted probe would therefore have greater amplitude. If the proximity probes are measured outboard of the bearing the amplitudes will be lower as the shaft has passed through a point of zero deflection at the node point within the bearing. Figure 8 shows the deflection of a rotating assembly going through the first mode of bending. The deflection inboard of the bearing is greater in amplitude than the deflection outboard of the bearing.

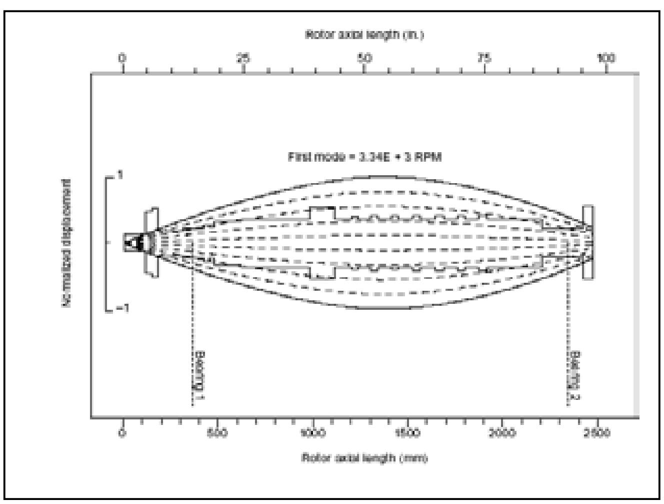



Fig 8 - Deflection of Rotating Assembly through First Mode.

Due to the various requirements for a particular design or enclosure it is not always feasible to mount proximity probes inboard of the bearing. It is however, important to recognize that motor designs do vary regarding the location of the proximity probes and that it is important to fully understand the vibration data being presented for that particular design.

## IV. WHAT EFFECT DOES HIGH AMBIENT TEMPERATURES AND HIGH ALTITUDES HAVE ON THE SIZE OF A MOTOR AND THE OPERATION OF A MOTOR?

NEMA defines the Ambient Temperature as “the temperature surrounding the cooling medium, such as gas or liquid, which comes into contact with the heated parts of the apparatus”. NEMA also defines the rated maximum ambient of a motor to be 40°C, unless otherwise specified.

Motors name plated with a 40°C maximum ambient temperature should successfully operate from -25°C to 40°C. The majority of industrial motors have a rated maximum ambient of 40°C, however motors with ambient temperatures greater than 40°C are less common.

When the ambient temperature exceeds 40°C, a motor rated for the peak ambient temperature should be purchased. High ambient temperatures impact the operation of a motor in several ways.

Elevated ambient temperatures directly impact the total stator and rotor temperatures. The total temperature of the stator and rotor will increase at a rate greater than the increase in ambient. Since the resistance of stator windings and rotor bars increase with temperature. The increased resistance causes higher stator and rotor $I^2R$ losses, which increase the motor temperature rise.

For example, an 800 hp, TEFC motor with a rated speed of 1789 rpm has a full load efficiency of 96.1%. The motor temperature rise by resistance is 78°C with 5990 watts of stator $I^2R$ losses and 3620 watts of rotor $I^2R$ losses. Increasing the ambient temperature from 40°C to 50°C increases the stator $I^2R$ losses by 180 watts and the rotor $I^2R$ losses by 110 watts. The temperature rise of the motor increases 2°C. The total stator temperature increases from 118°C to 130°C for an ambient change from 40°C to 50°C. The efficiency of the motor drops 0.1% to 96.0% at rated load.

Note that the temperature of the motor increased an additional 12°C and that the efficiency of the motor decreased. The total temperature of the motor in the above example would be acceptable with the 50°C ambient, if the above motor has a NEMA class F (155°C) insulation system. However, higher ambient temperature affects more than the motor insulation system. The bearing temperature must also be considered when dealing with a high ambient temperature.

Using the same 800 hp, TEFC motor in the previous example, the impact on bearing temperature can be considered. If the motor is a sleeve bearing motor, the bearing rise is 40°C at rated load. With a 40°C ambient the total bearing temperature is 80°C. If the ambient is increased to 50°C, the bearing total temperature will be 90°C. The temperature rise is now at the recommended alarm temperature for the bearing and within 5°C of the recommended trip temperature.

The impact on ambient temperature will vary by motor design, enclosure and load. If ambient temperatures above 40°C are required the motor manufacture should be consulted. Low ambient temperatures must also be considered. Ambient temperatures below -25°C may require special frame material such as ductile iron and bearing consideration such as sump heaters or special grease. Table 1 lists the usual recommended winding and bearing temperatures along with alarm and trip temperatures.

**TABLE 1**

| **Stator Winding** | **Class F Insulation** | **Class H Insulation** |
|---|---|---|
| Maximum running total temperature | 155°C | 175°C |
| Alarm total temperature | 160°C | 180°C |
| Trip total temperature | 170°C | 190°C |

| **Sleeve Bearing – Oil Lubricated** | **Bearing** | **Oil** |
|---|---|---|
| Typical running total temperature | 80°C | 60°C |
| Alarm total temperature | 90°C | 70°C |
| Trip total temperature | 95°C | 75°C |

| **Anti-Friction Bearing – Grease Lubricated** | **Bearing** | **Grease** |
|---|---|---|
| Typical running total temperature | 90°C | 80°C |
| Alarm total temperature | 110°C | 100°C |
| Trip total temperature | 120°C | 110°C |

Altitude will also affect the performance of a motor. The temperature rises defined on motor nameplates are for operation in altitudes of 1000 meters (3300 feet) or less unless otherwise noted. When motors are installed at altitudes greater than 1000 meters (3300 feet) special consideration must be given to insure the maximum winding and bearing temperatures are not exceeded.

The thinner air at higher altitudes reduces the heat transfer of motor losses to the surrounding ambient air. Therefore, motors operating in higher altitudes have increased temperature stator and bearing temperature rises. The impact of altitude on stator temperature rise can be calculated as follows:

For altitude in meters:

$$\mathbf{T_{RSL} = T_{RA}\ [1 - (Alt\text{-}1000)/10000]}$$
$$\mathbf{T_{RA} = T_{RSL}\ /\ [1 - (Alt\text{-}1000)/10000]}$$

For altitude in feet:

$$\mathbf{T_{RSL} = T_{RA}\ [1 - (Alt\text{-}3300)/33000]}$$
$$\mathbf{T_{RA} = T_{RSL}\ /\ [1 - (Alt\text{-}3300)/33000]}$$

Where:

$T_{RSL}$ temperature rise of motor in degree C at sea level.
$T_{RA}$ desired temperature rise in degree C at altitude.
Alt altitude above sea level in meter (feet) at which the machine is to be operated.

An example of the calculation for altitude is as follows: If the same 800 hp, TEFC motor used in the previous example was applied to an application at 8000 feet, the motor calculated winding temperature rise would be as follows:

Motor rise at sea level ($T_{RSL}$) = 78°C
$T_{RA} = T_{RSL}$ / [1 – (Alt-3300)/33000]
$T_{RA}$ = 78°C / [1 – (8000-3300)/33000
$T_{RA}$ = 91°C

To calculate the required temperature at sea level to meet an 80°C rise at 8000 feet, the calculation is as follows.

$T_{RSL} = T_{RA}$ [1 – (Alt-3300)/33000]
$T_{RSL}$ = 80°C [1 – (8000-3300)/33000]
$T_{RSL}$ = 68°C

The motor temperature rise at sea level should be 68°C or less to maintain an 80°C rise at 8000 feet. A larger motor would be required to meet the 80°C rise requirement at 8000 feet. Often higher altitudes have lower ambient temperatures. If the ambient temperature at 8000 feet can be lowered to 25°C from 40°C, the standard 800-hp motor could be applied at 8000 feet.

The total temperature of an 80°C motor with a 40°C ambient is 120°C. If the motor at 8000 feet with the 91°C rise is applied with a 25°C maximum ambient the total temperature will not exceed 116°C. The motor temperature remains within the 120°C total temperature for a class B rise.

Bearing temperature rises should also be addressed when considering high altitude applications. The same rise calculation applies to the bearing temperatures. While bearing temperature rises are not stated on the motor nameplate, it is important to be sure the issue is addressed. High altitudes will increase bearing temperature rises and may cause safe temperatures to be exceeded. Motor manufactures should always be advised of special altitude conditions when purchasing a motor.

## V. HOW OFTEN CAN A MOTOR BE RESTARTED AND WHAT CONSIDERATIONS SHOULD BE TAKEN INTO ACCOUNT?

NEMA MG-1 Part 12.54.and Part 20.12.1 defines the number of starts for an AC induction motor as:

a. Two starts in succession, coasting to rest between starts, with the motor initially at ambient temperature.

b. One start with the motor initially at a temperature not exceeding its rated load operating temperature.

In both starting conditions the load inertia must be equal to or less than the value listed by NEMA MG-1 for the rated horsepower and speed of the motor. During the accelerating period, the connected load torque must be equal to or less than a torque that varies as the square of the speed and is equal to 100 percent rated-load torque at rated speed. For special purpose or large motors, the same number of starts applies providing the inertia of the load, the load torque during acceleration, the applied voltage, and the method of starting are those for which the motor was designed.

The number of times a motor is started should be kept to a minimum. The number of starts affects the life of a motor. Excessive starts or high numbers of successive starts may lead to premature motor failure.

Some applications require starting that exceeds the values listed in NEMA MG-1. In these cases the motor manufacture should be consulted before ordering the motor. Often design changes can be made to accommodate special starting conditions or cycles.
NEMA MG-1 does not define the amount of time before the starting cycle can be repeated. The amount of time between cycles depends on the motor design, starting and load conditions. However, the following is a good guideline for most motor applications:

a. 45 minutes with the motor shutdown (sitting idle) before re-start.

b. 15 minutes with the motor running without load before shut down and re-start.

c. 30 minutes with the motor running at rated load before shut down and re-start.

Induction motors are occasionally exposed to operating conditions that temporarily disconnect the motor from its electrical power supply. The motor is usually reconnected at a later time ranging from a few cycles to several minutes. The amount of time before reconnection to the power supply is of concern since large transient torques can occur during the reclosure of the power.

Induction motors transient torque peaks, which occur under normal starting conditions, can be up to 5 times rated torque. Motors are designed to handle normal starting transient toques. However, the fast reconnection of a motor onto a power supply after a short power interruption can generate transient torque peaks up to 20 times rated torque.

Two types of reclosures are defined by NEMA MG-1 Part 20. They are as follows:

a. Slow transfer – Delay the reconnection until one and half-open circuit time constants or more has elapsed.

b. Fast transfer – The reconnection occurs in less than one and a half-open circuit time constants.

It is recommended that the slow transfer method be used to limit the possible damage that may occur to motor and/or driven equipment. When a fast transfer is desired a study of the electromechanical interactions of the motor, the driven equipment and the power system should be completed.

The open circuit time constant of a motor is the time required for the stator voltage to decay to 37% of the original voltage after power is removed from stator terminals. The open circuit time constant can be provided from the motor manufacture. When motor equivalent circuit data is available the open circuit time constant ($T'_{do}$) can be calculated as follows:

$$T'_{do} = (X_m + X_2)/2\pi f\, r_2$$

Where:

- $X_m$ Magnetizing reactance per phase.
- $X_2$ Rotor leakage reactance per phase at rated current.
- $f$ Rated frequency, hertz
- $r_2$ Rotor resistance per phase at rated speed and operating temperature.

## CONCLUSION

The specification of a highly engineered motor requires the expertise of rotating equipment engineers in evaluating issues relating to the long-term successful performance of motors. The topics discussed in this paper attempts to provide the user with the basic knowledge of items that should be carefully considered when the motor specification process is initiated.

## BIBLIOGRAPHY


[1] 3rd Edition of API-541, 1995.

[2] 1st Edition of API-684, 1996.

[3] 2nd Edition of API-670, 1986.

[4] Machinery Diagnostics Course Reference Book, Bently Nevada, 1992.

[5] Victor Wowk, Machinery Vibration Measurements and Analysis, McGraw Hill, New York, 1991.

[6} Robert O. Parmley, P.E., Illustrated Sourcebook of Mechanical Components, McGraw Hill, New York, 2000.

[7] Gary Donner, William Subler and Steve Evon "Motor Primer Part 1" in IEEE Transactions and Conference Records.

[8] Robert L. Mott, P.E., Machine Elements in Mechanical Design, Bell & Howell Company, Columbus, Ohio, 1985.

[9] Motion System Design Handbook, Penton Publishing, Cleveland, Ohio, 2001.


` ` ` ` ` ` ` ` `