A MOTOR PRIMER
Part 3

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Abstract

In recent years much has been written about motors on Variable Speed Drives, high speed rigid shaft motors, impact of API Standard 541, motor diagnostics, etc. 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 few colleges teach motor theory today, and that application experience at motor user locations has been reduced in recent years.

Introduction

This paper is the third in a series of papers where the authors provide answers to 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.

1. How do you size a motor?

Considerations for selecting a motor: Motor selection is a process containing numerous tradeoffs. The objective of motor selection is to arrive at the best possible installation, taking into account the following criteria: life cycle cost, horsepower and frame size for the specified life expectancy, load torque, load inertia, and duty cycle of the specified application.

The following discussion assumes that the motor to be selected will be a single speed induction motor, operating from normal power and not connected to a variable frequency drive. If a motor-drive combination is required, it is recommended that the motor and drive be supplied by the same manufacturer to insure that a compatible system is obtained.

For other types of motors, application assistance from the motor manufacturer is suggested. Before the size of a motor can be determined, several factors must first be evaluated. They are:

The Torque Versus Speed Requirements of the Load: The initial motor size must reflect the fact that in many cases the load torque increases as the equipment ages or operation conditions change. Failure to accurately establish the load torque requirements is the most frequent cause of incorrectly sized motors.

The Area Classification Where the Motor Will Be Located: If the motor is started in an atmosphere that may be combustible, the internal temperature of the motor should be limited to no more than 80% of the lowest auto ignition temperature of the gas that will be present during starting. Special precautions may be necessary to ensure safety of the installation. Some examples are:

* Limiting rotor bar and rotor hot spot temperature total rise to 200^0 C.
* Providing special seals to allow purging of the motor before starting.
* Ensuring that all covers and attachments are bonded so no sparking occurs.
* Using a totally enclosed machine to minimize winding contamination thus reducing air gap and end turn sparking.

The Number of Motor Starts: The number of starts the motor can expected to make during its life has a definite limit. For larger motors the number of starts is limited to 5000. If the motor will be expected to make more starts, the design will usually require the use of stronger shaft material, a larger diameter shaft or both.

Voltage: It is essential to have an accurate model of the power system to limit voltage drop in order to prevent some motors from shutting down when another motor is started. The voltage at the motor terminals during both starting and running must be known in order to insure that the new motor will start without affecting other equipment on the power system. If starting other motors will depress the voltage of the new motor below its critical recovery voltage, the new motor will stall. Also, low voltage can result in the motor being overloaded.

Uniform application of single-phase loads can help assure proper voltage balance in the electrical distribution system that supplies polyphase motors. Unbalanced voltage affects the motor's current, speed, torque, temperature rise and efficiency. A relatively small unbalanced voltage will significantly increase motor losses and decrease motor efficiency. The gains achieved by purchasing a premium priced, premium efficiency motor that reduces losses by 20% will be negated by a voltage unbalance of only 3.5%, because that small voltage unbalance will decrease the efficiency of the motor by 20%.

Load Profile: An accurate evaluation of the load profile is essential. Reciprocating or cyclic loads will have an impact on rotor, shaft, bearings, winding and housing design.

One of the most common sources of motor losses is a motor that is not properly matched to its load. In general, for standard NEMA frame motors, motor efficiency reaches its maximum at a point below its full load rating. Efficiency peaking below full load is a result of the interaction of the fixed and variable motor losses. Power factor is also load variable and increases as the motor is loaded. If the motor is operated above full load in order to take advantage of its service factor, the power factor begins to decrease because the motor's resistance to reactive ratio begins to decrease and the power factor declines. Motor loading and motor power factor have to be weighed against each other in order to obtain optimum motor efficiency.

In some applications where motors run for an extended period of time at no load, shutting down the motor and restarting it at the next load period could save energy.

Motor Sizing: Here are two methods that can be used to select the actual motor rating once a complete analysis of the motor service has been performed.

The most accurate method, regardless of motor horsepower, is to obtain an accurate speed-torque curve from the driven equipment manufacturer. This curve will have an end of curve horsepower rating. Create a graph that shows both the driven equipment speed-torque curve and the motor speed-torque curve at 80% voltage. See Figure 1.

Locate the pinch point, which is the point where the difference between the speed-torques of the 80% voltage curve of the motor and the load curve of the driven equipment is the least. The motor torque must be at least 10% greater than the load torque at the pinch point. The final check is to insure that the motor full load torque is always above the driven equipment full load torque, as it is in Figure 1.

The second method is for NEMA size machines, usually 500 Hp and less. There are four standard speed-torque characteristics available. They are NEMA A, B, C, and D. Each classification of motors has its own distinctive speed–torque relationship. See Figure 2.

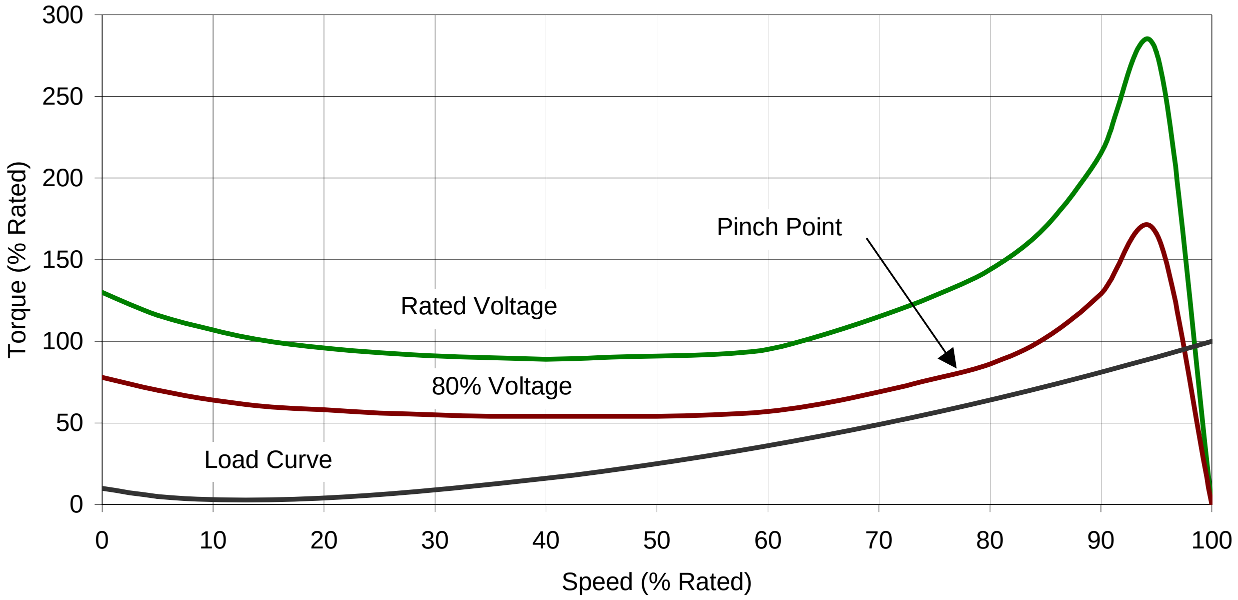


Figure 1 - Motor Speed-Torque Curves at 100% Voltage and 80% Voltage vs. Driven Load Speed-Torque Requirements

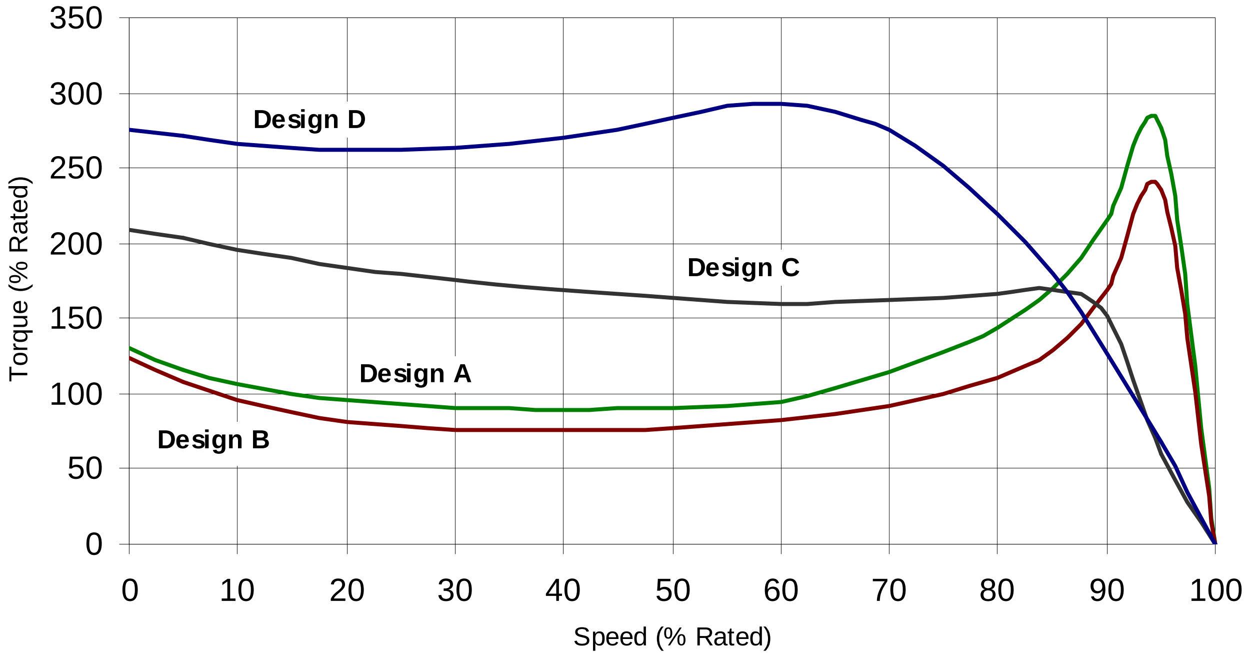


Figure 2 - Motor Speed and Torque for NEMA Design Motors

Comparison of NEMA Standard Design Motor Parameters					
NEMA DESIGN	LOCKED ROTOR TORQUE- % OF FULL LOAD TORQUE	BREAKDOWN TORQUE- % OF FULL LOAD TORQUE	LOCKED ROTOR CURRENT- % OF FULL LOAD CURRENT	% SLIP	RELATIVE EFFICIENCY
A	70-275	175-200	600-1000	0.5-5	MED-HIGH
B	70-275	175-300	600-700	0.5-5	MED-HIGH
C	200-250	190-225	600-700	1-5	MEDIUM
D	275	275	600-700	5-25	MEDIUM

Table 1- Comparison of NEMA motor characteristics

NEMA Design A and B Motors: The NEMA Design A motor is a variation of the B design, having a higher locked rotor current than the B design. These two designs are intended for general applications such as fans, blowers, centrifugal pumps, compressors, motor-generator sets, etc.

NEMA Design C Motors: This design is intended for applications where the motor will be starting under load, such as: conveyors, crushers, stirring machines, agitators, reciprocating pumps and compressors.

NEMA Design D Motors: This design is intended for high peak loads, with or without a flywheel, such as punch presses shears, elevators, extractors, winches, oil well pumpers, and wire drawing machines. Design D motors deliver high starting torque and are designed with high slip (more than 5%) so that motor speed can drop when fluctuating loads are encountered. Although Design D motor efficiency can be less than other NEMA designs, it is not possible to replace a Design D motor with a more efficient Design B motor, because the Design B motor would not meet the performance demands of the load.

See Table 1 for a comparison of NEMA motor characteristics. Figure 3 explains where each value of torque occurs on the motor speed-torque curve.

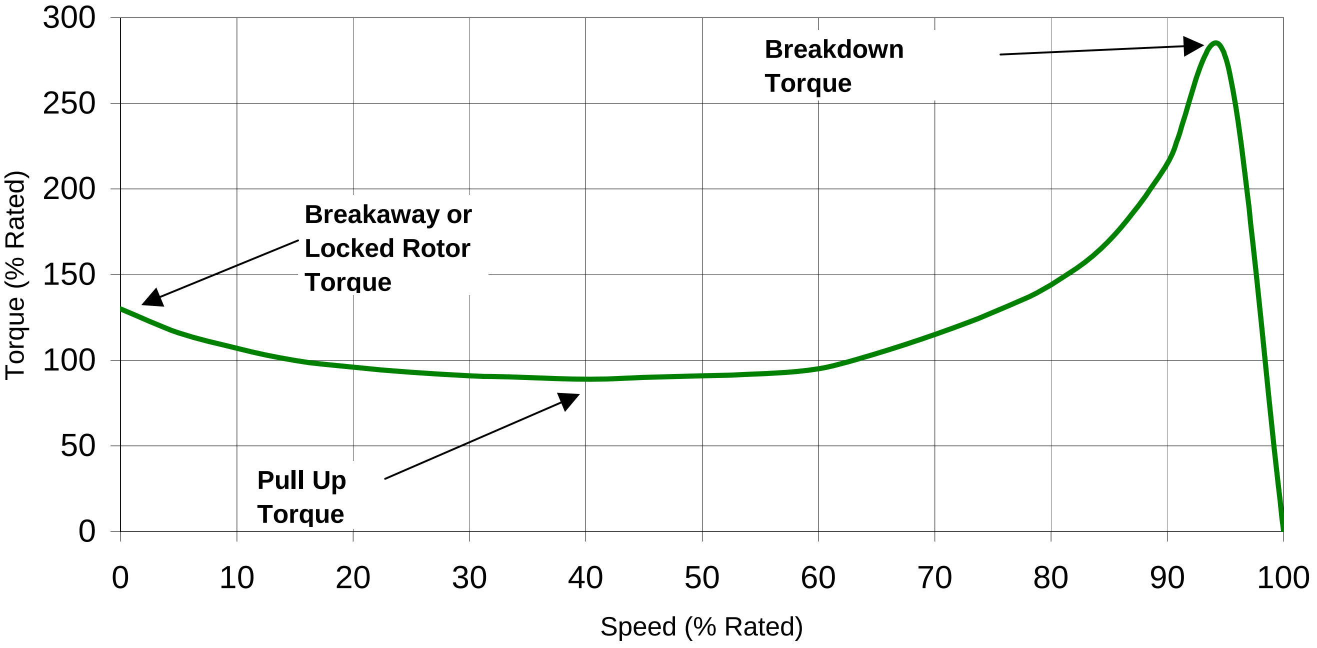


Figure 3 - Explanation of Motor Torques

Other Considerations for Motor Sizing When Using the Second Method:

Motor efficiency is not the most important consideration when selecting a motor because the motor with the highest operating efficiency does not always provide the lowest energy cost. If the motor is in cyclic service, a higher slip motor may actually save energy. Selecting the most efficient motor of a given size and type does not insure that energy savings are being optimized. Every motor is connected to some form of driven equipment: a crane, a machine tool, a pump, etc., and motors are often connected to their loads through gears, belts or slip couplings. By examining the total system efficiency, the component, which offers the greatest potential improvements, can be identified and purchased.

Energy efficient motors may be the most cost effective answer for certain applications. Here are simple guidelines to keep in mind when making this determination:

- Choose applications where motor running time exceeds idle time.
- Review applications involving large horsepower motors, where energy usage is greatest and the potential for cost savings can be significant.
- Select applications where loads are fairly constant and where load operation is at or near the full load point of the motor for the majority of the time.
- Consider energy efficient motors in areas where power costs are high. In some areas power rates can run as much as 18¢ per kilowatt-hour. In these cases the use of an energy efficient motor might be justified in spite of long idle times or reduced load operations.
- Utility rebate programs can also have a strong influence on the decision to purchase a high efficiency motor. In some areas of the U.S. and Canada the net cost of an energy efficient motor after rebate is less than that of a standard efficiency motor.

The simple guidelines outlined above will lead to the selection of correctly sized motors.

2. What causes shaft and bearing currents?

Currents that flow through the shaft and bearings of a motor are the results of a voltage potential between the motor shaft and frame. All motors have some voltage potential between the shaft and frame. It is only when this voltage potential is large enough to cause damaging currents through the motor bearings or driven equipment bearings that failures can occur.

Bearing currents are not new and existed since motors were first built. C.T. Pearce (1) in a 1927 article in the Electric Journal on the origins of bearing current remarked “If it were possible to design a perfectly balanced and symmetrical machine, both practice and theory indicate that no bearing currents, could exist." It may be possible to manufacture perfectly balanced and symmetrical machine but not cost effective.

Today motors are powered by two sources of power, sine wave power and variable frequency power. The power source of the motors is often used to distinguish between two categories of shaft currents. Since many papers have been published in recent years about shaft currents due to variable frequency drives (VFD's), most of this section will discuss shaft currents in sine wave applications.

Shaft currents in sine wave applications can be divided into two categorizes, homopolar flux and alternating flux linking the shaft. While both types of shaft currents can occur, the second type is much more common.

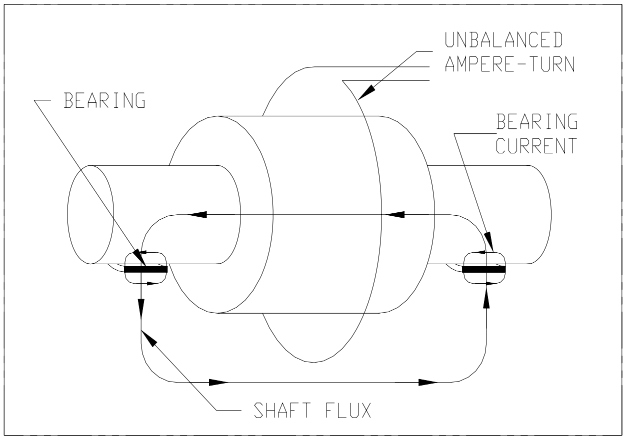


Figure 4

Homopolar flux. The homopolar flux is direct or alternating flux flowing in the shaft, through the motor bearing into frame and through the same bearing back into the shaft. It can also be referred to as shaft magnetization flux or "through flux". Figures 4 & 5 show the axial shaft flux and bearing current path.

Homopolar flux is usually constrained to high speed sleeve bearing machines, as it requires the bearing surface to span the flux path. It is caused by an unbalanced ampere turn encircling the shaft. Possible causes of the unbalance ampere turn are:

- Asymmetrical winding connection.
- Broken rotor bar.
- Sectionalized end ring.
- Shaft residual magnetization.
- To a lesser extent, uneven air gaps (rotor eccentricity).

The bearing current induced is usually minor. There is no established method to measure this type of bearing current flow since it is normally contained in the bearing. An insulated bearing may reduce the bearing current, but does not eliminate homopolar flux. Installing nonmagnetic bearings, bearing housings and/or shaft can isolate the source.

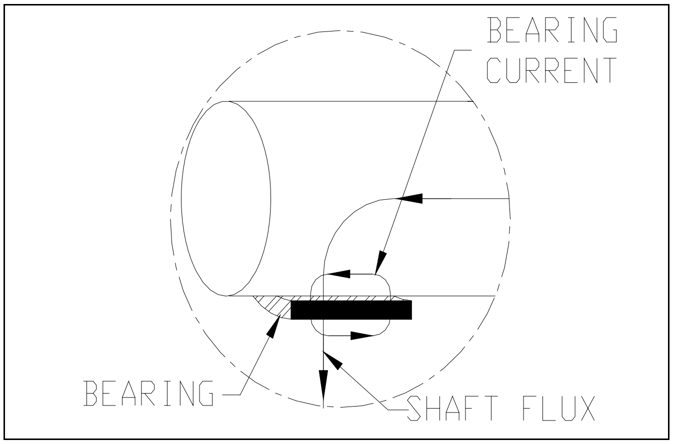


Figure 5

Circulating shaft currents. Circulating shaft/bearing currents due to time varying flux linking the shaft are more common than homopolar flux. The axial flux is generated through transformer action. Since electrical steel is not totally homogenous, flux paths in the motor are not entirely symmetrical. An asymmetric rotor and/or stator core construction may cause a net flux to encircle the shaft. The shaft, bearing and frame may be seen as a one-turn secondary winding of a transformer and an e.m.f. along the shaft will be induced. The resultant current will flow through the shaft, down the bearing, through the frame and back through the second bearing. The flux path and current path can be seen is Figure 6.

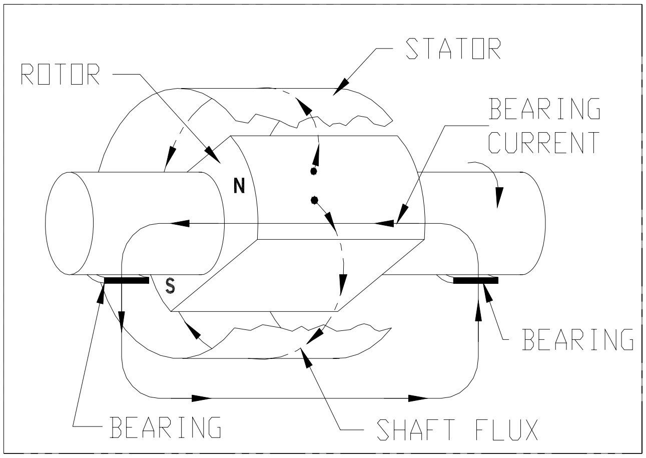


Figure 6

Many of the causes of shaft current can be controlled. Laminations are designed as symmetrical as possible. One common cause of high shaft currents is the number of straps or welds holding a stator together. Even symmetrical combinations such as four and six welds or straps may induce shaft current for certain number of poles. For example, six stator straps or welds will induce shaft currents in four and eight pole motors. Dissymmetries in the rotor construction may also cause shaft currents. An example of a weld location dissymmetry can be found in Figure 7.

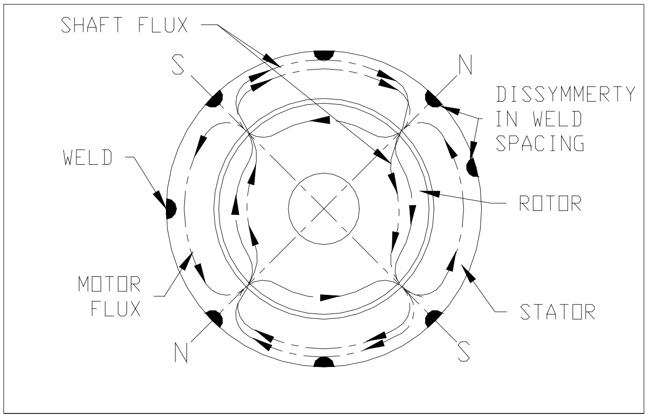


Figure 7

When both bearings are uninsulated and conducting, shaft voltages as small as 500 millivolts can cause shaft currents greater than 20 amps. Properly designed motors will not have shaft current levels that are high enough to cause bearing damage. It has been the experience of the authors that current levels of 20 amps or greater require at least one insulated bearing. NEMA MG1-1998 recommends insulated bearings if shaft axial voltage exceeds 300-millivolt peak under sine wave operation.

Unlike homopolar shaft currents, circulating shaft currents can be measured by shorting both ends of the shaft with a low impedance cable. (See IEEE Standard 112-1996). The circulating shaft current can only be present when both motor bearings are conducting or when the opposite drive end bearing of the motor is uninsulated.

Protecting bearings from circulating shaft currents can be accomplished by insulating the opposite drive end bearing or insulating both bearings. Insulating only the drive end bearing may still result in circulating shaft currents. Since the shaft voltage is generated in the rotor, the driven equipment bearing may complete the conductive path through the shaft to opposite drive end bearing. Therefore, circulating shaft currents continue to flow through the bearings, perhaps damaging the bearings in the driven equipment.

Shaft current from VFD's. Bearing currents may also be generated when motors are powered by variable frequency drives (VFD's). The previously discussed internally sourced bearing currents may still exist on VFD's. The externally source currents are a result of the wave shape from the VFD. Bearing currents from VFD's can occur, although the majority of motors on VFD's do not have bearing currents levels that are high enough to damage to motor bearings.

Many papers have been written over the last several years about VFD sourced bearing currents. Proper high frequency grounding is critical to limit the currents caused by VFD's. When bearing currents from VFD's are present, techniques such as insulated bearings, shaft-grounding brushes, good high frequency grounds and/or electrostatic shielded windings may provide protection for the bearings. Since so many papers are available on this subject and it would consume the entire paper to properly address the subject it will not be addressed in detail. The best method of avoiding bearing currents on VFD's may be to purchase the motor and drive from one manufacturer.

3. What causes rotor bars to break?

There are many causes of rotor bar failures, but most can be divided into three categories: thermal stress, magnetic stress or mechanical fatigue. Both cast aluminum and copper bar rotors can be subjected to these types of stresses. However, the construction and design of both will have an impact on the thermal and mechanical stress levels they can withstand before failure.

Rotor failures may appear to be sudden, but with a few exceptions they usually occur over a long period of time. Early detection of rotor bar failures is difficult. The primary reason for most rotor failures is operation of the motor beyond its design limits.

Thermal Stress: Thermal stress failures occur when the motor is pushed beyond its designed capability. Thermal stress failures can occur quickly or over time. The most common causes of thermal stress are excessive number of consecutive starts, long periods of rotor acceleration, rotor stalling, excessive overloads, and rotor rubbing the stator.

Perhaps the most common cause of motor thermal stress failure is from excessive consecutive starts. Induction motor currents during starting are usually 5 to 7 times full load current and produce high I^2R losses in the rotor. Since the heat is generated quicker than it can be dissipated to the laminations, the rotor bars and end rings, the temperature of the rotor rapidly increases. It is not uncommon for rotor bar and end ring temperatures to rise more that 20°C per second during starting. The end rings become a heat sink for the rotor bars causing the end ring to expand radially, subjecting the rotor cage to high thermal stresses and dynamic loads.
While the temperature increase in the end rings is rapid, the temperature increase in the laminations is gradual. The combination of the coefficient of expansion for copper being greater than steel and the large difference in temperatures between the rotor laminations and rotor end rings causes a radial bending force to be exerted on the rotor bars. This results in a high stress point where the bars exit the rotor core. Figure 8 shows an exaggerated example of the force for a copper bar rotor. Figure 9 shows the force on cast rotor.

The bars usually extend from 1 to 3 inches beyond the end of the rotor laminations on bar rotors. This bar extension allows for a certain amount of radial growth. However, when the designed limit of radial growth is exceeded on a regular basis, as in the case of excessive consecutive starts, the bar material will fatigue and crack, which will over time lead to a rotor failure.

Cast aluminum rotors, see Figure 9, do not have bar extensions. Cast aluminum rotors do have other features that the bar rotors lack. A cast aluminum rotor assembly is one integral unit, the aluminum cage is cast directly into the lamination core allowing the material to maintain intimate contact with the laminations. The intimate contact improves the heat transfer from the aluminum to the cast rotor laminations, reducing the amount of heat transferred to the end rings. The rotor fans are integrally cast into the end rings and also provide excellent heat sinks for the end rings. The lower end ring temperature will limit radial end ring expansion. However, excessive consecutive starts will also fatigue the rotor bar and end ring joint similar to a bar rotor and cause rotor bar cracking and failures.

NEMA MG-1 12.54.1 and 20.43.1 states that motor should be capable of:

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.

If more consecutive starts are required, the motor manufacturer should be consulted at the time of purchase to ensure that premature rotor failures do not occur.

Consistent excessive overloads effect the rotor similarly to excessive consecutive starts. The excessive overloads cause thermal cycling in the rotor bar and end ring joints. If the overloads were not considered in the design of the motor, the thermal cycles may causes excessive radial stresses in the bar and end ring joints. Leading to rotor bar cracks and rotor failure. Consistent excessive overloads should be avoided or addressed at time of motor purchase.

Rotor stalling may also cause rotor bar failures. Rotor stalling occurs when the load starting torque requirements exceed the torque produced by the motor. Single phasing of the motor, low voltage or improper motor sizing are some causes of stalling. The motor will either not start or will run at a speed below breakdown. In either case the motor current is normally 5 to 7 times rated current. If the motor is not taken off line and is allowed to run under these conditions for a short period of time, excessive rotor and stator heating will occur. Rotor temperature increases of 20°C per second or greater often occurs. Rotor failures can then occur as the rotor bar temperature reaches the melting point of the aluminum or braze material.

Rotors rubbing the stator may also cause rotor failures. Rotor rubs can occur during bearing failures, from improper air gap alignment and from unbalance magnetic pull. The heat from the friction of the rotor and stator can lead to rotor bar failures.

Magnetic Stress: Magnetic stresses can be grouped into two categories: unbalanced magnetic pull and electromagnetic stress. Magnetic stresses are usually due to a defect in the motor. While application conditions may accelerate the failure, the cause is usually manufacturing or design related.

All motors have some amount of unbalanced magnetic pull. In the manufacture of a motor some dimension tolerances are allowed. Therefore, the motor air gap is usually not uniform. The nonuniform air gap results in a greater magnetic pull toward the small side of the air gap. If the motor air gap eccentricity is too great or the shaft too flexible then rotor pull over may occur. Rotor pull-over results in the rotor striking or rubbing the stator. The friction of the strike will eventually result in a rotor failure due either to excessive temperature or the forces from the strike. Motor manufactures normally design motors with low air gap eccentricities and shaft stiffness with adequate safety margin to prevent magnetic pullovers.

Electromagnet stresses occur when a vibration occurs as a result of electromagnetic forces. One type is the movement of the bar radially between the top and bottom of the slot. The vibration occurs at twice rotor current frequency and is a result of the slot linkage flux generating electrodynamic forces on the bars. The deflection normally occurs during starting and results in a fatigue failure of the bars. Failures will not occur in properly designed rotors with controlled clearances. Another type of electromagnet stress occurs from eccentric air gaps. The eccentric air gap causes electromagnetic vibrations or noises. These vibrations may lead to fatigue failures or may simply cause increased noise levels at rated load. The vibration or noise will increase with an increase in air gap eccentricity.

Mechanical Stress: Rotor failures related to mechanical failures of the rotor bars, end rings and lamination may also occur. These failures are normally related to centrifugal forces. Centrifugal force is proportional to the product of the mass times the square of the speed.

Centrifugal Force Equation:

$F = mrw^2$, where
m = mass
r = radius
w = speed in rpm.

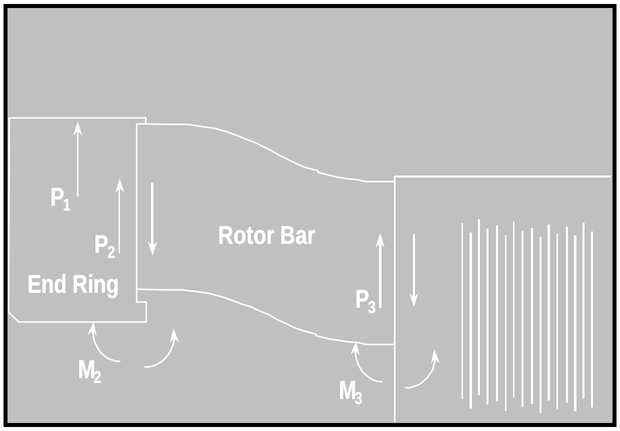


Figure 8 – Copper Bar Rotor

Large motors are where centrifugal forces become large enough to be of concern. Variable speed motors running above safe operating speed also may have rotor failures due to centrifugal forces on the rotor. Excessive centrifugal forces place the rotor end ring in hoop stress and the bar extension in shear with a bending moment. Figures 8 and 9 show examples of these stresses on bar and cast rotors.

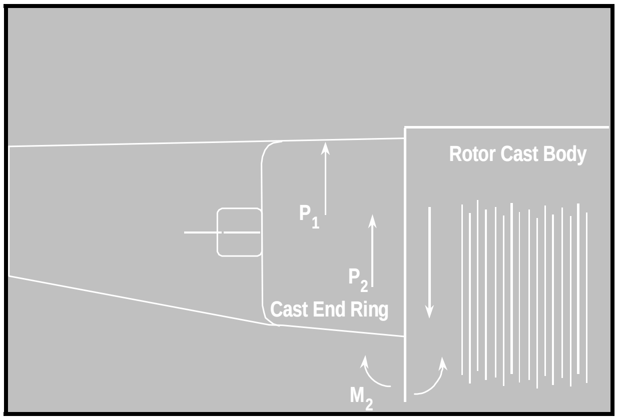


Figure 9 – Cast Aluminum Rotor

Motor designers can reduce the stress due to centrifugal forces by modifying the rotor design. Modifying the bar extension lengths reduces the compression and tensile stresses in the lower and upper edges of the bar. Modifying the slot shape and height, see Figure 10, can redistribute the stresses and extend the rotor life. Modifying the short circuit end ring shape or incorporating shrink rings can minimize the radial expansion of the end ring. Alloy changes are also possible to obtain higher endurance limits.

Figure 10

The rotor laminations are subjected to high stresses induced by centrifugal forces on the rotor cage (Fig. 11). The centrifugal forces act directly on the rotor bars forcing them against the bridge. This force places the bridge in bending and in shear at the outside edges of the slot.

High stresses can also cause yielding or the fracturing of the laminations between the slots, bridge bending, etc., (Fig 12a and 12b).

Minimizing the stress levels prevents possible vibration problems by a loss of the interference fit, which can create uneven thermal expansion, imbalance, and lower critical speeds. When lamination stresses exceed stress limits, mechanical engineers will work with electrical engineers to minimize the stresses. Minimizing stresses may involve a new slot design, changing the number of bars, modifying the bridge thickness, etc.

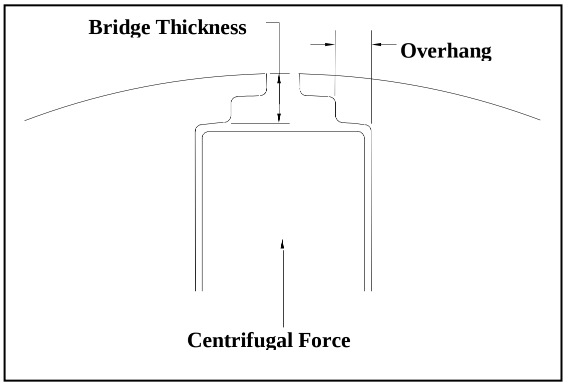
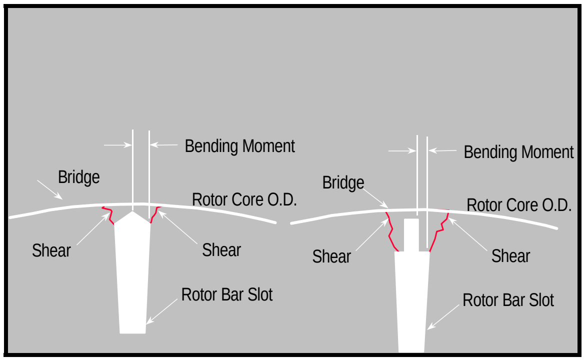
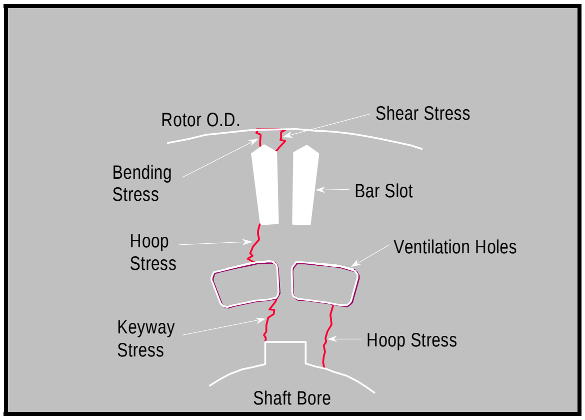


Figure 11 - Centrifugal Force on the Lamination Slot

Rotor failures due to centrifugal force can usually be attributed to poor motor design, manufacturing issues or operating the motor above its maximum safe operating speed. Proper modeling tools allow the motor designer to predict force and stress levels and design motors that will not have failures due to centrifugal force.





Figures 12.a and 12.b - Lamination Stresses

4. What does the bearing L_{10} life really represent?

Bearing life and reliability. Bearing performance revolves around evaluating the rating and average life of the bearing based on the loading of the bearing. As in all rotating machinery exposed to stresses rolling bearings have a definite life span and will eventually fail due to fatigue.

The basic dynamic load rating for a bearing defines the bearing load that will provide a basic rating life of 1,000,000 revolutions of the inner ring. The rating life (L_{10}) of an identical group of bearings is the life where 90% of a given bearing population will not fail due to material fatigue. The average life (L_{50}) of an identical group of bearings is the life that will result in a 50% survival rate.

Material fatigue is the result of high cyclical stresses occurring in the load carrying zones of the bearing. When the endurance life of the material is exceeded a crack initiates beneath the area of high stress and propagates to the surface resulting in a spalling away of the surface material. The spalling or flaking is actually bearing material separating from the raceways and rolling elements.

Bearing life can be calculated in various degrees with the Basic Rating Life Equation being the simplest method:

$L_{10h} = (C / P)^p / (1000000 / 60n)$ where;

L_{10h} = Basic rating life, operating hours
C = Basic dynamic load rating
P = Equivalent dynamic bearing load
n = Operating speed
p = Life equation exponent
- Ball Bearings use 3
- Roller bearings use 10/3

Many factors affect the overall bearing life, with the major ones being:

- Proper Installation
- Lubrication
- Operating Temperature
- Operating Alignment
- Shaft and Housing Fits
- Material Quality

To address some of the factors affecting bearing life the Adjusted Rating Life Equation was developed to provide a more detailed evaluation of the bearing life by taking into consideration factors for reliability, material and operating conditions.

The Adjusted Rating Life Equation:

$L_{10ah} = a_1 a_2 a_3 L_{10h}$ where;

L_{10ah} = Adjusted rating life, operating hours
a_1 = Reliability life adjustment factor
- 90% reliability use 1.00
- 95% reliability use 0.62
- 99% reliability use 0.21

a_2 = Material and operating condition life adjustment factor:
- Bearing steel improvements allow 1 for this factor.

a_3 = Operating conditions life adjustment factor:
- Factor based on the amount of lubrication.
- A viscosity ratio (k) is the ratio of the actual viscosity to the viscosity required for adequate lubrication. Depending on the viscosity ratio k, the a_3 factor can range from approximately 0.7 to 2.6.

In recent years the New Life Theory was developed that expands upon the adjusted rating life equation to evaluate the bearing life concerning lubrication type and contamination values.

$L_{10aah} = a_1 a_{skf} L_{10h}$ where;

L_{10aah} = Adjusted rating life according to new life theory, operating hours

a_1 = Reliability life adjustment factor
- 90% reliability use 1.00
- 95% reliability use 0.62
- 99% reliability use 0.21

a_{skf} = Life adjustment factor based on new life theory
- This factor takes into consideration the viscosity ratio (k) along with the level of contamination (n_c) and the fatigue load limit (P_u) that represents the load below which fatigue will not occur in the bearing. Values of a_{skf} are given as a function of n_c (P_u/P) for different values of the viscosity ratio k. The level of contamination (n_c) can range from 1.0 for a very clean environment to 0.00 for a heavily contaminated environment.

Application of all Three Methods. Example: Assuming a purely radial load of 2000 lbs. what is the Basic, Adjusted and New Life Theory L_{10} life of a 6222 bearing operating at 1500 rpm.

Bearing Size:	6222			Dynamic Load Rating (C):	32200	lbf
Bearing Bore (d):	110	mm		Fatigue Load Limit (Pu):	899	lbf
Bearing OD (D):	200	mm		Equivalent Dynamic Load (P):	2000	lbf
Actual Viscosity (v):	15.1	mm/s^2		Operating Speed (n):	1500	rpm
Required Viscosity (v_1):	9.3	mm/s^2		Viscosity Ratio (k):	1.62	

		Factors						
		a_1	a_2	a_3	n_c	n_c (P_u/P)	a_{skf}	Hours
Basic Rating Life:	L_{10h}	x	x	x	x	x	x	46370
Adjusted Rating Life A:	L_{10ah}	1	1	1.36	x	x	x	63063
Adjusted Rating Life B:	L_{10ah}	1	1	0.85	x	x	x	53603
New Life Theory A:	L_{10aah}	1	x	x	1	0.45	50	2318489
New Life Theory B:	L_{10aah}	1	x	x	0.2	0.09	3.5	220720

Table 2 – Bearing Life Example

From the tabulated results above the Adjusted Rating Life A has sufficient viscosity whereas in B the bearing is operating with an insufficient viscosity resulting in a lower life. The New Life Theory examples show New Life A operating in a clean environment with no contamination. The New Life B example is operating in a contaminated area resulting in a significant life reduction as compared to the New Life A example. The tabular results indicate for the Adjusted and New Life Theory that the operating conditions must be well defined as estimations of factors can lead to errors and magnify life calculations.

References:

1. C.T. Pearce, "Bearing Currents – Their Origin and Preventions," The Electric Journal, Vol. XXIV, no. 8, pp. 372-376, August 1927.

2. P.L. Alger and W. Samson, "Shaft Currents in Electric Mahines," AIEE Trans., vol. 43, pp. 235-245, 1924.

3. J. Boyd and H. N. Kaufman, "The Causes and Control of Electrical Currents in Bearings," Lubrication Engineering, Vol. 15, pp. 28-35, January 1959.

4. S. Evon and B. Oakes, "Variable Frequency Drive Principles and Practices (Above NEMA) AC Motors for Variable Frequency Applications", 1999 TAPPI Trans.

5. A. Bonnett and G. Soukup, "Rotor Failures in Squirrel Cage Induction Motors", IEEE Trans, Vot. IA-22, No. 6, November/December 1986.

6. M. Bradford, "Unbalanced magnetic pull in a 6-pole Induction Motor", Proc. Iee, Vol.. 115, No. 11, November 1968.

7. SKF General Catalogue, Third Edition, 1999.

8. R. McFarland, "Use Vibration Analysis To Increase Bearing Life", Maintenance Technology, December 1997.