ELECTRIC MOTORS FOR CENTRIFUGAL COMPRESSOR DRIVES

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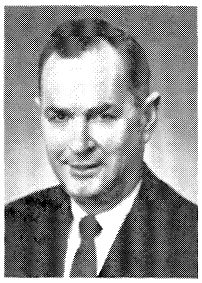

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INTRODUCTION

In the past, the decision to use electric motors for centrifugal compressor drives has been made only after all other possibilities to use either a steam or gas turbine have been exhausted—or so it has seemed to motor suppliers. This picture may however be changing as a result of the increase in fuel costs, particularly petroleum. In addition to the considerations of heat balance and investment cost, a third factor—the availability of feed stock—must be considered. Why burn a part of the needed hydrocarbons when electric power generated from coal, hydro, or nuclear energy can be purchased with a resultant saving of feed stock to make more product? Obviously, with these alternatives this should not occur—particularly when electric motors of high efficiency and reliability are available to drive almost any configuration of centrifugal compressor. The purpose of this paper is to highlight the important elements of selecting a motor for centrifugal compressor drives.

CORRECT MOTOR SIZE SELECTION

Nothing is more useless than a motor installed on its foundation that is too small to start the machine to which it is coupled. There is no excuse for this problem because the widespread use of computers by machine designers makes it practical to predict accurately the brake horsepower requirements of the compressor. It is easy to avoid this problem because motors are available in standard ratings in the larger horsepower sizes in average increments of about 15 percent. Therefore, a practice of selecting the standard motor rating that matches or exceeds the brake horsepower of the driven machine will result in 0 to 15 percent margin.

The practice of adding 10 percent to the calculated brake horsepower and then using the next larger standard horsepower motor will result in margins of 10 to 25 percent.

With this wide selection of standard ratings, it is easy to find a good match between the driven machine size and the motor rating without the use of continuous overload capabilities.

Past practice has been to use motors with an overload capability of 115 percent. This can reduce investment cost if the machine is to be operated at 115 percent load. The motor life, however, will be shortened because the temperature rise at 115 percent load is 90C. Conversely, the life of the same motor will be increased if this motor is operated at 100 percent load or approximately 70C rise.

CORRECT MOTOR TORQUE SELECTION

Torque selection would be easier if a simple match between full-load brake horsepower and the motor rating were the whole story. To get the benefits of the motor's simplicity, the plant designer must understand its peculiarities. When a motor is starting, it draws four to seven times its full-load current. This depresses the voltage, particularly on large motors, and results in a reduction of motor starting torque by the square of the voltage drop. If the voltage drops to 80 percent, the torque drops to 64 percent $[(80\%)^2=64\%]$. This will be no problem if the designer has planned for this condition and matched the motor speed torque curve at reduced voltage to that required by the compressor.

It is good practice to apply a motor with torque capability at the voltage which the system drops to when the motor is being started that is at least 10 percent greater than the torque required by the driven machine at all points between zero and full speed. See Figure 1.

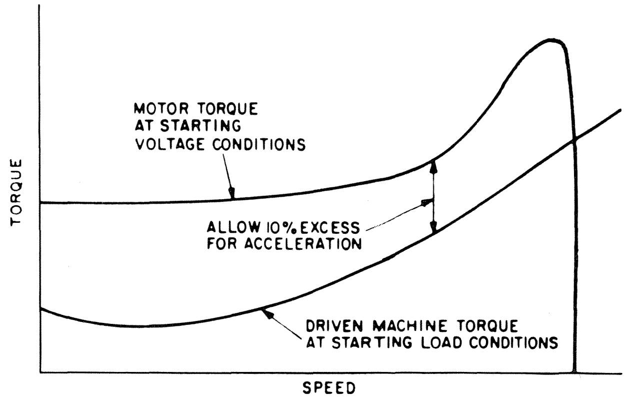


Figure 1. Typical motor vs driven machine speed-torque curve.

A motor with special torque characterictics, a stiffer power system, or a method of unloading the driven machine may be required. On some very large motors, all three of these may be required to ensure a drive that will operate.

MOTOR VOLTAGE AND STARTING METHOD

Selection of the proper motor voltage is usually determined by the available electrical system and its size. The most frequently used voltage for 500-4000-hp motors is 4160 volts. On large electrical systems, motors 4000 hp and larger may have to be rated 6600 or 13,200 volts because of available switchgear.

When motors are started, they draw high inrush current that depresses the line voltage. The amount of voltage drop depends upon the size of the electrical system. Standards usually indicate this voltage dip should be less than 10 percent; however, there are many good systems where the motor driving a compressor is the largest in the plant and where the voltage dip is 20 to 30 percent or more.

As pointed out previously, motor torque varies as the square of the applied voltage. This reduced available torque usually requires that motor-driven compressors be started with the inlet valves or guide vanes closed. Most compressors will operate under these conditions for a short time (usually 60 to 120 seconds) without damage from heating or surge until the motor reaches full speed.

In many cases, the compressor breakaway torque is 15 to 20 percent and as the compressor comes up to speed, it evacuates itself until the torque at full speed is 15 to 20 percent. These curves typically are as shown in Figure 2.

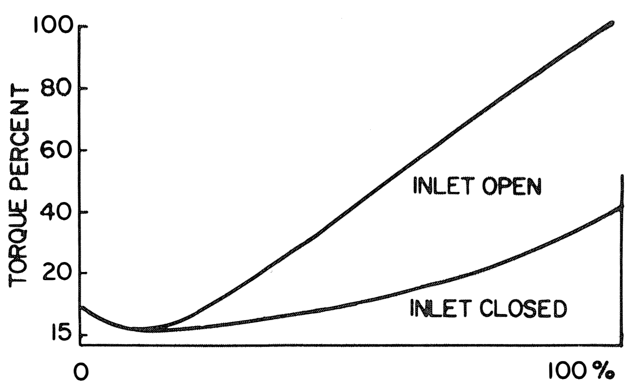


Figure 2. Typical compressor speed-torque curve.

With low starting torque requirements, it is possible on large ratings to design special low-inrush low-torque motors that will cause less voltage drop when starting.

The three commonly used methods of reduced-voltage starting in use at present are: reactor start, auto-transformer start, and unit transformer start. The most popular of the three is the unit (or captive) transformer start because it does not require extra switching, can be designed to meet individual system requirements simply by selecting the transformer impedance, and can permit the use of medium or small motors on 13,800-volt systems without penalizing motor characteristics. Reduced voltage starting is never a drive requirement, but is a requirement to prevent or limit the electrical system disturbance.

WK^2 OF THE COMPRESSOR

The energy required to accelerate the inertia of the driven machine shows up as heat in the motor rotor as the machine comes up to speed in exactly the same way that the kinetic energy of a moving automobile is converted to heat in the brake linings as the automobile is stopped. Frequent stops or too heavily loaded stops of an automobile will cause the brake linings to overheat and fail. In a similar fashion, starting the motor too frequently or starting a driven machine with a load inertia larger than the motor was designed for, will cause the motor rotor winding to overheat and fail. Here the analogy stops for, if the motor is properly matched to the load inertia, the rotor winding will last indefinitely. Industry standards have been established for the load WK^2 capability of the standard motor. These values are high enough for most centrifugal compressor drives. Many high-speed compressors require step-up gears. The WK^2 at the motor shaft is proportional to the square of the gear ratio. For this reason, special motor characteristics may be required.

MOTOR SPEED AND TYPE

Motor Speeds

Centrifugal compressors of higher speeds are being used more frequently than ever before. The top motor speed available is 3600 rpm on a 60-hertz system. To meet requirement speeds higher than 3600 rpm, it is necessary to use step-up gears.

The introduction of a step-up gear permits a normal induction motor speed of 1800 rpm. For 500 to 20,000-hp drives, 1800 rpm is the lowest first-cost induction motor. Similarly, the lowest first cost synchronous motor speed is 1200 rpm in sizes from 5000 hp and up. There is really no size limit when building a motor. Four 45,000-hp motors, Figure 3, coupled in tandem to drive a 180,000-hp axial-flow compressor at Ames Aeronautical Laboratory, Moffet Field, California, have been operating for many years.

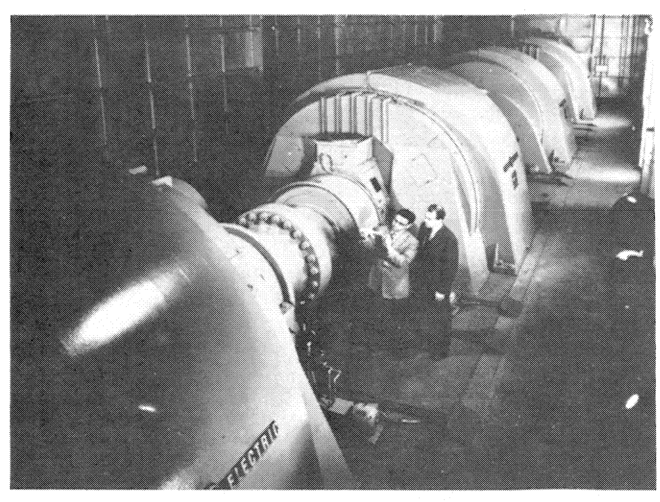


Figure 3. Four 45,000-hp wound rotor induction motors connected in tandem to drive a wind tunnel 180,000-hp axial-flow compressor.

Motor Types

There are three types of motors available for compressor drives:

1. Induction
2. Synchronous
3. Wound Rotor Induction

The induction motor is the first choice for drives from 500 to 5000 hp because it has one insulated stator winding and one uninsulated, shorted rotor winding. For 1200-, 1800-, and 3600-rpm compressors, no speed increaser is required. The 1800-rpm motor is the least expensive of the three speeds; therefore, this speed is usually selected for higher speeds using step-up gears.

The synchronous motor is usually the primary choice for very large ratings because of price when power factor and efficiency are prime considerations. These motors have three windings: one insulated winding on the stator and two windings on the rotor—one insulated and one uninsulated amortisseur winding that is used for starting only.

The insulated synchronous motor rotor winding requires a d-c power supply that is usually supplied from a brushless exciter mounted on the motor shaft.

Synchronous motors are available at the same speeds as induction motors; however, in practice only the 1200-rpm speed is used. Step-up gears are always used to match the compressor speed requirements. The control and operation of synchronous motors require more devices, but these are well understood and are very reliable. The system is such that the operator controls the machine in the same way he controls an induction motor.

The synchronous drive is more efficient, can operate at 1.0 pf or 0.8 pf, and is lower in first cost in the larger horsepower sizes. Recent changes in motor prices have altered the specific relationships of synchronous versus induction motor prices, but a general relationship continues to exist as shown below:

For Geared Drives	Lowest First Cost	
	Induction 1800 rpm	Synchronous 1200 rpm
1. 500-15,000 hp	X	
2. 6,000-50,000 hp		X

The overlap between 6,000 hp and 15,000 hp is intentional because specific drive characteristics and requirements will change the choice.

For very large drives above 30,000 hp, the use of non-self-starting synchronous motors at 3600 rpm is possible. These are synchronous generators that must be brought up to speed with a starting turbine or motor and synchronized with the system and operated as a motor. Obviously, it is a requirement on these drives that the compressor be unloaded to 15 to 20 percent of full load to keep the starting equipment size reasonable.

Wound-rotor induction motors have not been used for compressor drives because of their cost and high losses when resistors are used for speed control. One exception is the very large drives used in wind tunnels where there

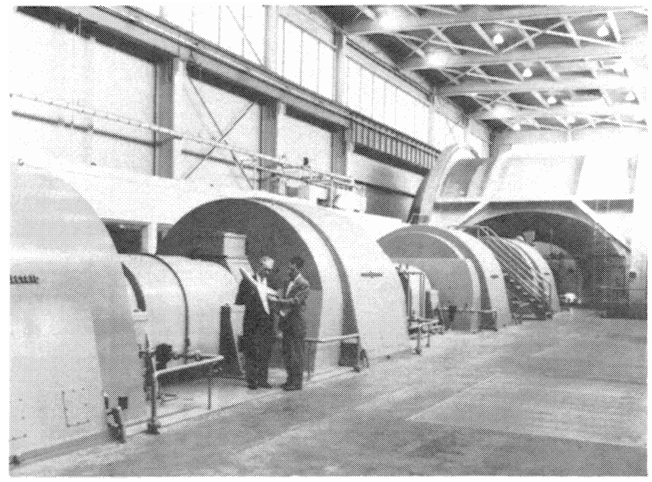


Figure 4. Four 37,500-hp motors connected in tandem to drive a wind tunnel 150,000-hp axial flow compressor.

is no other way to control the compressor output except by speed control. The units shown in Figure 4 are installed at NASA Lewis Flight Propulsion Laboratories, Cleveland, Ohio, and consist of four 37,500-hp motors in tandem driving a wind tunnel 150,000-hp axial flow compressor.

The increased cost of energy and the availability of thyristors in large power sizes is causing a new interest in the wound-rotor motor because instead of discarding the slip power as heat at lower than synchronous speed, this power is "recycled" by solid-state rectifier inverter techniques and put back into the power system. (See Figure 5.) On applications where speed control is required (and particularly where the load cycle requires operation at reduced speed for long periods), this drive results in relatively high efficiency over the entire speed range. Figure 6 shows a 16,800-hp a-c adjustable speed drive motor installed at the Schenectady Plant of the General Electric Company.

MOTOR ENCLOSURES

Motor winding life can be substantially increased by the proper choice of enclosure.

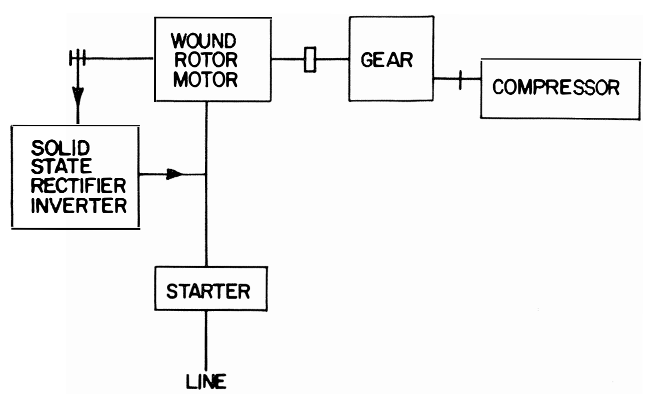


Figure 5. Slip frequency is recycled for high efficiency at reduced speed.

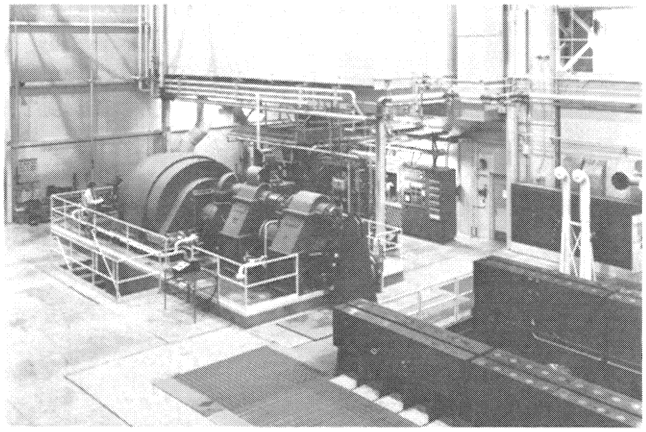


Figure 6. AC adjustable-speed test stand driven by a 16,800-hp motor.

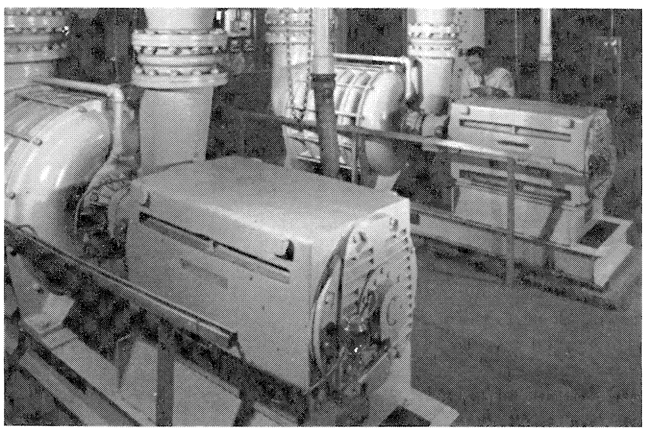


Figure 7. Dripproof enclosed induction motor, rated 450 hp, 3600 rpm.

1. Open Dripproof—For use in indoor, relatively clean locations. See Figure 7.

2. Weather Protected II (WP II)—For use in outdoor locations. This enclosure can easily be modified to include filters when used in extremely dirty locations. See Figure 8.

3. Totally Enclosed, Water-air-cooled (TEWAC)—For use in outdoor or indoor, clean or dirty locations. Since the heat from the motor losses is taken away by the water, motors having this enclosure are well suited for installation in small rooms. See Figure 9.

4. Totally Enclosed, Fan-cooled (TEFC)—These machines are arranged so that the heat is transferred from the motor to the ambient by utilizing an air-to-air heat exchanger either mounted on the frame of the motor or built as a part of the motor with fins cast into the endshields or frames. See Figure 10.

5. Hazardous Locations—This requirement must be met by pressurizing the motor enclosure with clean air or inert gas. Explosion-proof motors are not generally available above 500 hp.

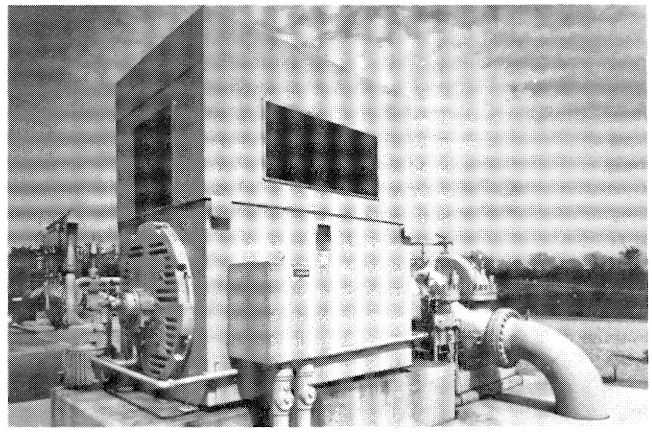


Figure 8. NEMA Weather-protected Type II (WPII) induction motor, rated 6000 hp, 1800 rpm.

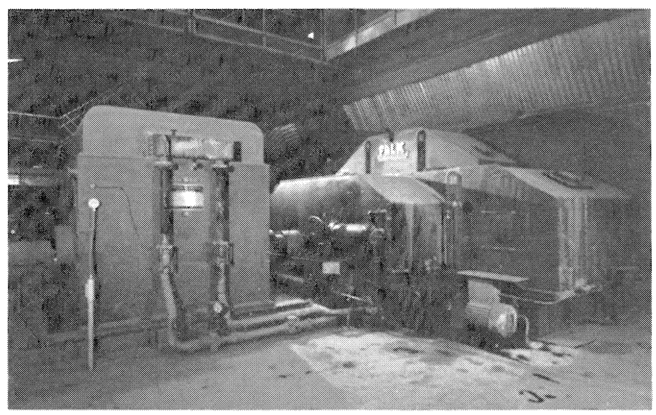


Figure 9. Totally enclosed synchronous motor with top-mounted air-to-water heat exchanger.

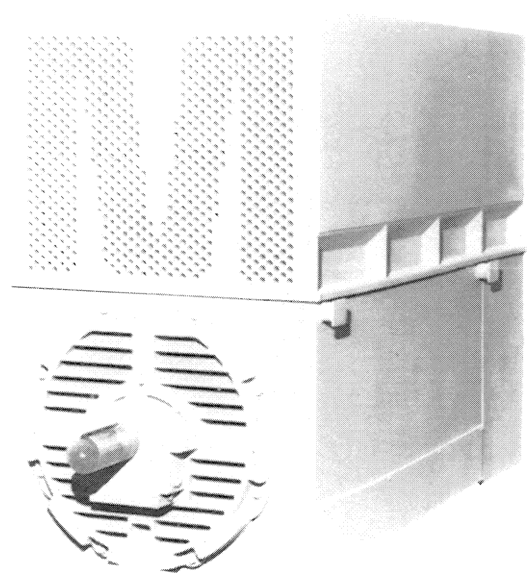


Figure 10. Totally enclosed fan-cooled induction motor with top-mounted air-to-air heat exchanger.

INSULATION SYSTEMS

The insulation system used on most motors in the 500- to 20,000-hp size is rated Class B and is based on mica and epoxy. Mica has long been referred to as the standard for dielectric capability, particularly in the area of corona resistance. Mica is applied to the coils, and then by use of various vacuum pressure methods (See Figure 11), the coils are impregnated with epoxy which is highly resistant to moisture. Epoxy is inert to most chemicals, has excellent mechanical characteristics, and because of the impregnation process, serves to enhance the mechanical strength of the winding by completely filling it and bonding it together. The mechanical strength of the epoxy mica system is great. To it have been added bracing systems like the General Electric COIL LOCK system, Figure 12, utilizing polyester glass materials to mold a support system designed to hold the motor end turns practically immobile during the high-current starting conditions. These systems virtually eliminate coil movement and chafing between turns which have been a primary cause of winding failure in the past.

Class B insulation, normally furnished with NEMA standard 100 percent rated motors, operating at 80C rise by resistance in a 40C ambient, has a design life of approximately 15 to 20 years, which matches normal plant life. Because the rate of an oxidation reaction is doubled for each 8-12C rise in temperature, it has been said that the life of a motor insulating system is cut in half for each 8-12C the total temperature of the winding is increased. Therefore, another way of applying motors more conservatively is to specify a lower than standard temperature rise.

Because of torque limitations discussed previously, in practice the lower temperature rise can better be secured by specifying an oversize motor. This procedure does not penalize the motor operating characteristics, power factor and efficiencies, if no lower than ¾ load operation is contemplated. Most motors will operate at approximately 55C rise at 75 percent load. For temperature rise at loads in between 75 percent and full load, the curve, Figure 13, is typical.

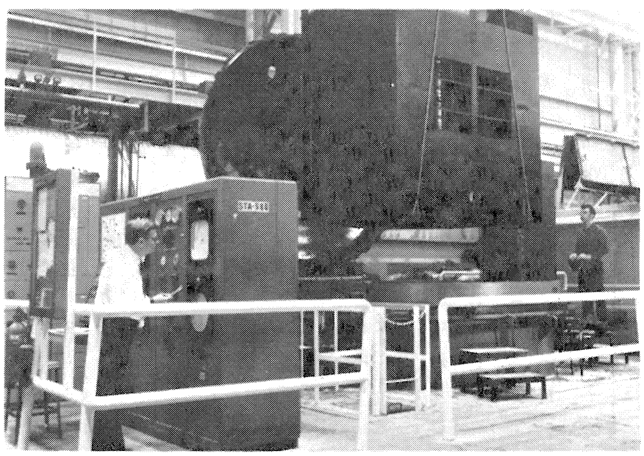


Figure 11. Complete stator being placed in vacuum pressure impregnation (VPI) tank.

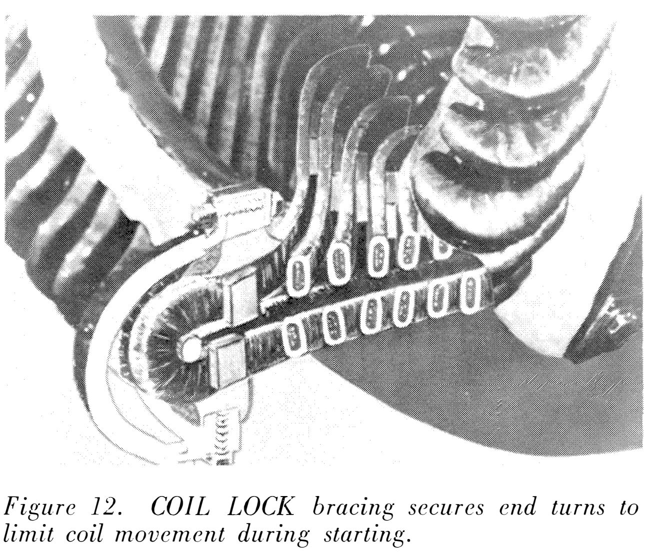


Figure 12. COIL LOCK bracing secures end turns to limit coil movement during starting.

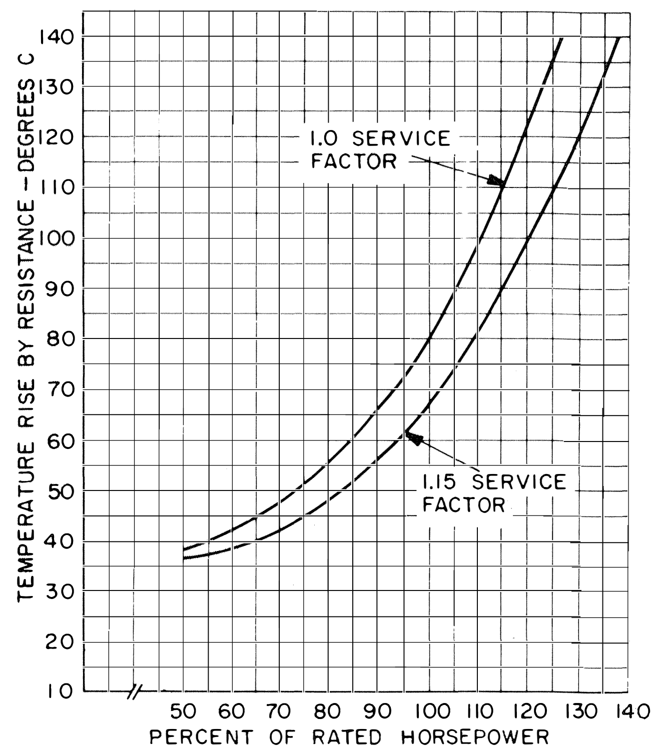


Figure 13. Typical temperature rise vs percent of rated horsepower curve.

CONTROL AND PROTECTIVE EQUIPMENT

It is generally beyond the scope of this paper to go into the details of the control except to list the functions needed and to stress the importance of care in selecting and coordinating this equipment with the motor design.

The function provided by the control should include those listed in Table I.

TABLE I

Function	Ind. Motor	Syn. Motor
1. Stall protection	X	X
2. Running overload protection	X	X
3. Protection against single-phase operation	X	X
4. Protection against failure to synchronize		X
5. Protection against loss of excitation		X
6. Protection against operation out of synchronism		X

In addition, the following will give the motor greater protection and should be used on large and critical drives:

Temperature Detectors (Stator Winding)

Temperture detectors in the stator windings are available on all motors being considered. With relays installed in the control, the signal from these detectors can be made to sound an alarm when there is an abnormal change in motor winding temperature. This signal can be used as an indication of motor distress so that the operator can check for overload, clogged filters, or other obstructions in the ventilating passages to the motor. A second relay can be arranged to shut the machine down when the operating temperature exceeds the design limit by a nominal amount (approximately 10C).

Bearing Temperature Detectors

Bearing temperature detectors or relays are available that will sense the bearing temperature on sleeve bearings and should be used on large drives to minimize damage. Bearing temperatures usually do not change unless a loss of oil film (caused by foreign material in the oil, loss of oil supply, change in alignment, etc.) has caused bearing damage. These devices are available for all motors and should be used on all drives where downtime must be held to a minimum. High-temperature alarm and high-temperature cutoff are available and should be set at a nominal amount above normal operating temperature and at the maximum temperature permitted by the design, respectively. These devices may not give sufficient warning to saving the bearings, but journal and rotor damage is prevented.

Lightning and Surge Protection

Lightning arrestors and surge protection should be installed on all motors rated 1500 hp and larger or 4000 volts and higher. It may be desirable to protect even smaller machines on critical drives. The lightning arrestors can be located in the control or at the main plant substation if there is no exposed line between this point and the motor. The surge protection must be located at the motor terminals. These surge capacitors slope off the voltage surges caused by switching other equipment in the same fashion that surge tanks or flow restricting devices soften hydraulic surges caused by valving in a hydraulic system. Analogous to the hydraulic system, it does little good to locate the surge tank very far from where the surge can cause damage. Over-size terminal boxes with lightning arrestors and surge protection capacitors, Figure 14, are optional accessory items for large motors.

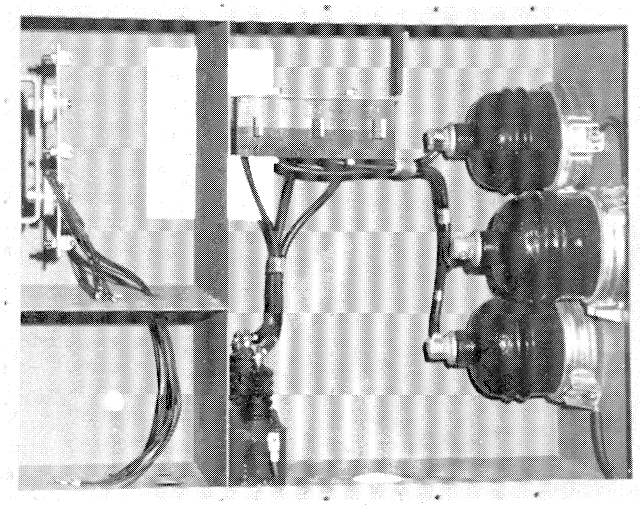


Figure 14. Motor terminal box with lightning arresters and surge protection capacitors.

Space Heaters

Space heaters should be installed on motors located out-of-doors or indoors where changes in ambient temperature, when the motor is shut down, can cause accumulation of moisture by the dew point phenomena. These heaters simply maintain the motor temperature slightly above ambient so that dew will not form during temperature changes. Corrosion of the mechanical parts and moisture damage to the insulation is thereby minimized or eliminated.

Differential Protection

Damage caused by arcing when a motor fails can be minimized by the use of differential protection. This is accomplished by comparing the current in each phase of the motor winding to sense any current unbalance caused by winding turn insulation failure. An alternate and less expensive system, called a ground sensor relay, is frequently used on critical drives. This device senses a line current unbalance caused by a fault to ground. Both systems are very sensitive and will limit motor punching damage from arcing to ground.

MECHANICAL CONSIDERATIONS OF MOTOR-GEAR-COMPRESSOR ARRANGEMENT

Most of the considerations discussed thus far are well understood by electrical engineers. However, in many cases, mechanical coupling of the motor, gear, and compressor is often overlooked. The three most important factors to be considered in a discussion of mechanical coupling are torsional analysis, vibration, and coupling selection. Not withstanding their importance, it is possible to deal with them in only a cursory fashion in the following sections.

TORSIONAL ANALYSIS

Mechanical consideration of the motor-gear-compressor arrangement must be carefully designed as stated by P. B. Thames and T. C. Heard in 1959 at an AIEE Middle Eastern District Meeting.

"The necessary precautions to assure satisfactory torsional vibration performance have long been the object of routine analytical procedure for equipments using synchronous machines directly connecting reciprocating compressor and engines. The primary responsibility for such analyses, and the provisions to be made to assure satisfactory torsional performance, has traditionally rested with the compressor or engine manufacturer. Less well appreciated, though, is the importance of making a similar analysis of the torsional mass elastic system for other types of drives. Examples of other types in which such analytical precautions have become routine are certain types of wood chippers used in the paper industry and geared turbine-compressor sets. With the increasing usage of large motor-gear-compressor drives in the petroleum, gas transmission, and rare gas industries, the same analytical precautions have been extended to this category of equipment. Historically speaking, the chances are not great that difficulty in this respect will be encountered on any given equipment. On the other hand, the risk incurred with the increasing size of the drives and the number of installations using a multiplicity of identical drives, warrants a thorough investigation of torsional performance before the equipment is manufactured.

Any system consisting of inertias connected by shafting has one or more natural torsional frequencies. The values of these frequencies should be determined, and it should be ascertained that these natural frequencies are not in near resonance with possible continuous torsional exciting forces. Prolonged operation at or near resonance will result in shortening life of the gear and couplings and possible damage to these components or the shafting. It should further be determined that the equipment will not be damaged by conditions of unavoidable resonance which may exist transiently."[1]

The above paper develops methods of calculating torsional performance of a motor, gear compressor system by the use of analog computers. However, many motor-compressor drives are put together without this type of analysis—with surprising success. It has been this author's experience that when a torsional analysis is made, either the shaft sizes of the gear or motor, the coupling flexibility, and occasionally the size of the gear is changed to improve torsional performance.

CONTROL OF VIBRATION AND BALANCE

General

Excessive vibration is undesirable since it imposes unnecessary forces and stresses on the equipment, its foundation, and nearby equipment. In addition, vibration is often the source of undesirable air-borne noise.

Excessive vibration may be produced by:

1. Excessive vibratory forces, resulting in the generation of forced vibrations of unacceptable levels.

2. Resonant amplification, resulting in the generation of unacceptable vibration levels by normal vibratory forces.

3. A combination of Items 1 and 2.

The vibration level experienced by an installed machine is a function of many variables. Some of these variables

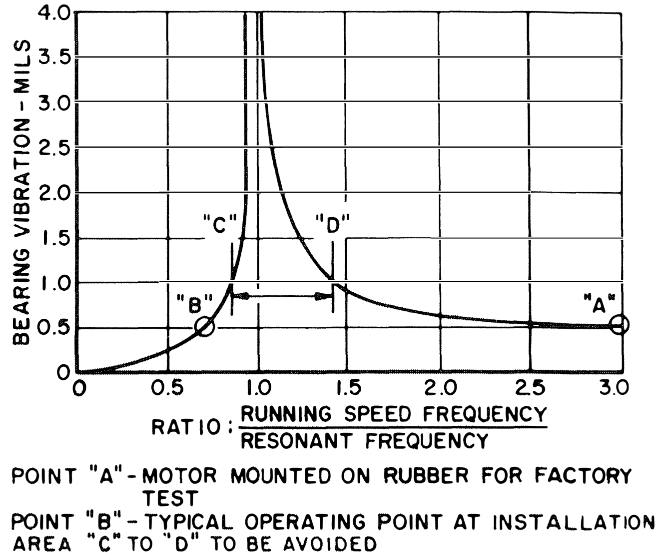


Figure 15. Typical resonant response curve.

are controlled by the motor manufacturer; some are functions of the foundation, driven equipment, alignment, etc., and are obviously beyond the control and responsibility of the motor manufacturer. The responsibility for the satisfactory mechanical operation of the motor-driven compressor system generally lies with the compressor builder or the user who is putting the system together.

System Vibration at Resonance

A motor and the machines it drives form a system which has many resonant frequencies. The amount of vibration that will result from impressing periodic or vibratory forces on the system will be influenced by the relationship between impressed forcing frequencies and the resonant frequencies. Figure 15 is a typical vibration response curve which shows how the vibration amplitude varies, due to rotating unbalance, at different ratios of running speed frequency to system resonant frequency. The desirability of preventing the coincidence of a forcing frequency with a resonant frequency is obvious. It is usually desirable to have the resonant frequency 25 percent above or below the forcing frequency.

Motors are designed and built so that resonant frequencies of the motor components by themselves, or as an assembly, do not approach any known forcing frequency. It is possible, however, that the combination of the mass and stiffness of a motor with the mass and stiffness of a base or foundation will result in a combined resonant frequency which lies too close to a primary forcing frequency, resulting in excessive vibration. The problem of resonant amplification should be one of the concerns of those assembling and/or installing the motor-driven compressor system.

SOURCES OF MOTOR VIBRATION

Mechanical Unbalance of Rotor

All rotating bodies have a certain amount of unbalance and also have a point defined as the residual unbalance.

This point represents the practical lower limit to which balance can be refined. As standard procedure, the rotors of new motors are balanced to generally accepted limits for most applications. The degree of balance a given motor has is generally defined by the amount of vibration that will be produced under certain standard test conditions. The vibration produced under the conditions is not only a function of the actual amount of unbalance in the rotor, but is also a function of the mass of the motor, the stiffness of the supports on which the motor is mounted. etc. This system is used because most people are not concerned with the number of ounce-inches unbalance the rotor of a motor may have, but rather with the vibration this unbalance will produce at some measurable point on the motor.

Since, in most cases, it is not practical for a motor manufacturer to duplicate actual motor mounting conditions, NEMA (National Electrical Manufacturers Association) has set up standard mounting and test procedures. For these tests, the motor is mounted on soft rubber mounts, which results in a system resonant frequency well below the running speed frequency of the motor. The purpose of this mounting is to decouple the motor from foundation effects so that the vibration produced by rotor unbalance, magnetic forces, etc., can be measured without inconsistent outside influences. NEMA standards describe the following test procedure for factory vibration test:

"*Method of Measuring the Motor Vibration (Dynamic Balance).*

A. Place the motor on an elastic mounting so proportioned that the up and down natural frequency shall be at least as low as 25 percent of the test speed of the motor. To accomplish this it is required that the elastic mounting be deflected downwards at least by the amounts shown in the following table due to the weight of the motor. When a flexible pad is used, the compression should in no case be more than 50 percent the original thickness of the flexible pad; otherwise, the supports may be too stiff.

RPM	*Compression Inches*
900	*1*
1300	*1/4*
3600	*1/16*
7200	*1/64*

B. The amplitude of vibration shall be measured on the bearing housing, in any direction, with the axis of the shaft in the normal position. With the motor operating at no load, it shall be balanced with one half a standard key in the keyway; that is, a key of full length flush with the top of the keyway. An alternating-current motor shall be operated with rated voltage and frequency applied and a direct-current motor shall be operated from a ripple-free power supply."[2]

The factory limits for the vibration produced by a motor when tested in the previously described manner, are given in Table II which follows:

TABLE II
FACTORY-TEST VIBRATION LIMITS
(In Mils Peak-to-Peak Vibration)

MOTOR SPEED (RPM)	NEMA STANDARD	SPECIAL
3000 and up	0.001	0.0005
1500 to 2999	0.002	0.001
1499 to 1000	0.0025	0.001

NOTE: Large 3600 rpm motors usually require special balance.

It should be recognized that the vibration amplitude limits given in Table I apply to a given set of conditions of mounting, temperature, etc., for factory tests and are not necessarily the limits which can be expected under actual operating conditions. The vibration amplitudes that are produced under actual operating conditions may be more or less, depending upon a variety of factors.

Magnetic Forces

Inherent in the design of electric motors are the magnetic forces acting across the air gap between the rotor and stator. These forces produce the useful work performed by the motor; however, they also produce side effects that can result in undesirable vibration and noise. The magnetic flux across the air gap produces a steady force which tries to pull the rotor and stator together; the flux also produces a varying force which tries to vibrate the rotor and stator at a frequency of twice the frequency exciting the motor. These forces are fairly well neutralized by centering the rotor in the stator to produce a uniform air gap and are minimized by the design and connection of the stator windings. However, the forces cannot be eliminated; those remaining are responsible for normal motor vibration and noise at twice-line frequency and the resultant harmonics.

It is important to note that these forces can be transmitted from the motor to the driven equipment, foundations, and adjacent structures and that resonant amplification in these parts can result in excessive vibration and noise. In addition, when a motor is installed, it can be subjected to conditions that may increase twice-line frequency vibrations by increasing the amount of vibratory force and/or changing the resonant frequencies, either of which may result in resonant amplification. Vibratory forces may be increased by running the motor at higher than rated voltage. Mechanical misalignment can cause shaft deflections that produce air gap dissymmetries with the resultant increase in unbalanced magnetic forces.

The resonant frequencies of major motor components can be changed by the characteristics of the foundation on which the motor is installed, by the type of coupling used to connect it to the driven equipment, or by differential thermal expansion.

When designing a motor-driven compressor installation, the coincidence of system resonant frequencies with the various forcing frequencies present in the system should be avoided. The primary forcing frequencies originating in the motor are:

1. Running-speed frequency
2. Line (power supply) frequency
3. Twice line frequency

SOURCES OF SYSTEM VIBRATION

The vibration a motor-driven system will experience is not only a result of the vibratory forces produced by the motor and driven equipment, but also the dynamic response of the system. This system includes bases, foundations, piping and all associated parts and structures. The sytem, then, must include all components participating in the vibrations.

The primary vibratory forces associated with motor operation have been discussed and it is beyond the scope of this paper to discuss the vibratory forces generated by the wide variety of compressors. Each type of compressor may produce a particular set of vibratory forces and they act on the system and create vibratory responses.

The manner and precision with which a motor is coupled to the driven equipment can greatly influence the amount of vibration produced by the motor and system. The proper selection of couplings for the particular application is important, and precise alignment between shafts is necessary to maintain vibration at acceptable levels.

Improper coupling and alignment practices can result in excessive vibration, not only by the creation of unnecessarily large vibratory forces, but also by altering the resonant characteristics of the system to cause resonant amplification.

The amount of vibration various types of machinery can safely be subjected to without incurring undue hazard of premature failure, safety, etc., varies greatly. This is determined not only by the type of machine, size, speed, etc., but also by the critical nature of equipment application and what other consequences may occur due to a forced shutdown or premature failure. Generally, it is up to the user to determine the standards of performance. The tolerance curves shown in Figure 16 are typical, but are intended for guidance purposes only.

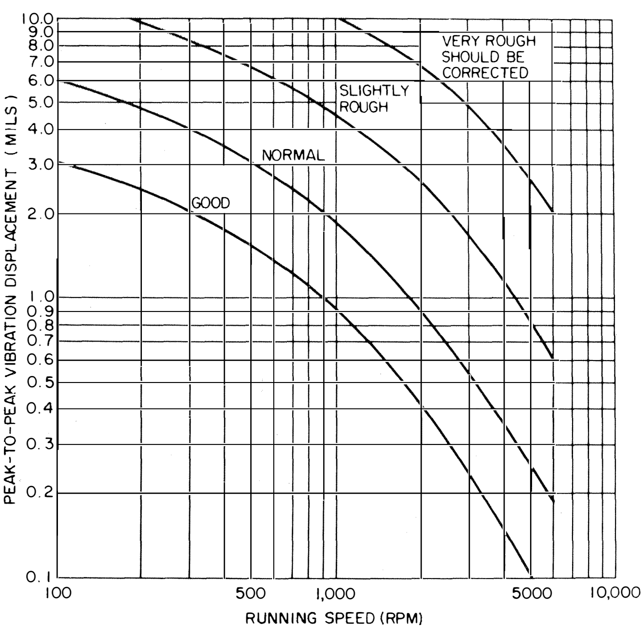


Figure 16. Machinery vibration tolerance curves.

There are few published standards of either vibration (or balance) limits or methods of measuring motor vibration. However, the NEMA standard procedure for measuring motor vibration and the typical factory vibration limits shown in Table II are generally acceptable for most applications.

Customers who specify vibration limits for motors generally do so in terms of mils vibration produced at the bearing housings with the motor operating at rated line voltage and frequency on standard NEMA mounts. This appears to be the most practical approach to the problem because it recognizes that the manufacturer cannot duplicate actual installation characteristics for test purposes; it also recognizes that there are other possible sources of motor vibration in addition to rotor unbalance.

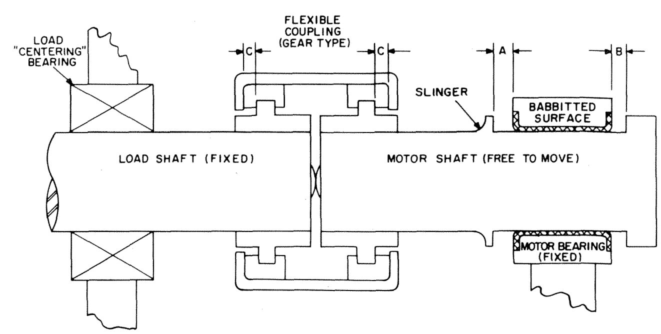
COUPLINGS—CHOICE IS IMPORTANT

There are many types of couplings in use, but most can be classified as either rigid or flexible.

The flexible coupling is the type most often used with induction and synchronous motors. For compressor drives, the flexible coupling comes in many forms: gear type, flexible-disc type, pin and bushing type, spring-grid type, and rubber-biscuit type.

Experience has shown there is considerable misuse of the flexible coupling. Many users have the mistaken idea that flexible couplings eliminate the need for precision line-up of equipment. This is not true! The function of the flexible couplings is mainly to compensate for some minor misalignments which might occur during operation due, for example, to change in relative foot heights as motor and driven equipment heat to operating temperature.

A typical coupling arrangement is illustrated in Figure 17. It is good practice to have a "centering" or thrust-carrying bearing as an integral part of the equipment, which determines, within fairly close limits (0.015 inch), the position of the rotating element of the driven equipment (fixed shaft in Figure 17). If the coupling shown in this arrangement were of the rigid type, the position of the motor shaft would be set relative to its own bearings



MOTOR SHAFT POSITION	MOTOR END FLOAT (INCH)	A	B	C
SHAFT ENDS TOGETHER	1/4	11/64	5/64	3/64
	1/2	11/32	5/32	3/32
SHAFT ENDS APART	1/4	5/64	11/64	0
	1/2	5/32	11/32	0

Figure 17. Recommended clearances with limited end-float flexible coupling.

except for the axial growth of the shafts due to temperature changes or due to a change in position of the "centering" bearing. Reasonable motor bearing clearances ("A" and "B") would then protect the motor bearing from having the slingers come in contact with the babbitted end of the bearing.

Flexible couplings of the disc type and/or rubber-biscuit type also limit axial movement of the shafts sufficiently to prevent motor bearing damage with normal bearing clearance.

Many designs of flexible couplings (gear type, pin and bushing type, spring-grid type) permit relative motion between the shafts. Unless this movement ("C" + "C") is limited to less than the motor bearing clearances ("A" + "B"), it is possible for the shaft slingers to come in contact with the motor bearing. Axial thrust may then be transmitted through the coupling to the bearing shoulder. Experience has shown that in many applications, sufficient axial thrust to cause bearing damage may be transmitted through the flexible coupling. The limited end-float coupling is designed with typical clearances as shown in the table, Figure 17, to prevent the shaft slingers from contacting the motor bearing positioning surfaces.

Axial thrust may be due to:

1. gravity forces, because the rotational axis is not level,
2. forces produced within the coupling due to misalignment wear, and
3. axial expansion of the shaft system and displacement of the "centering" bearing.

Regardless of how well-designed the coupling is, when it is operating under load, the load torque will produce sufficient friction between the coupling elements so that a fairly large axial force may be required to change the axial spacing of the coupling halves. This friction depends upon a number of factors such as misalignment, amount of wear on the elements, and the cleanliness of the assembly. Thus, a loaded coupling tends to be an axially "locked" device and will transmit thrusts produced by items (1), (2), or (3) mentioned previously.

CONCLUSION

Electric motors are easy to apply as centrifugal compressor drivers. Their efficiency is usually over 95 percent so they fit today's need to save energy. The application principles outlined in this paper are well understood and these simple motors with only one moving part are among the most reliable machines produced in industry. Indeed, the trend will be toward increased usage of motors for compressor drives as the balance of energy costs shift.

REFERENCES

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