Electric Motors and Their Rating for Sucker - Rod Pumping

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Three-phase, squirrel cage induction motors are used almost exclusively for oil well pumping on electrified leases. The squirrel cage motor is available with various designs, enclosures and other electrical and mechanical variations. This paper will discuss squirrel cage motors most ideally suited and generally used as a drive for individual walking beam sucker rod pumps.

NATURE OF THE LOAD

The torque required to start or break away and accelerate a walking beam pump load is rather high and hard to predict. Factors such as the position from which the pump is started, condition of the beam and gear unit, formation of paraffin in the well tubing and the temperature and icing conditions may result in a tremendous friction load for breakaway, acceleration and running.

The load of a walking beam sucker rod pump is cyclic, having two peak (maximum) and two minimum power demands for each stroke. On ideally balanced pumps, the peak power in pumping on the up stroke equals the peak power in lifting the counterbalanced weights on the down stroke. Also, the minimum demands should be of the same magnitude, and power should not reverse.

In actual practice, a pump is considered properly balanced if the peak power demands are approximately equal. The term "approximately equal" is used because the polished rod horsepower may vary depending on fluid column, fluid friction, and spasmodic gas agitation of the fluid column, making it impossible on some wells to continuously maintain an ideally balanced pump.

On leases with relatively few motors and with a power system and source just adequate for the average load, cyclic peak loads may become particularly objectionable from a system voltage and/or frequency standpoint. A number of wells may get in step causing pyramiding of the cyclic peak loads, which may result in severe voltage and frequency regulation and stalling of motors or dropping out of contactors.

The minimization of peak load current values is especially important when power is generated on the lease with limited generating capacity. The differential of peak power current for Design C and D motors increases with increased pumping speeds and on systems with high voltage regulation.

Peak mechanical load at the polished rod and in the rod string are essentially the same with normal slip or high slip motors, if the same strokes per minute are maintained. Peak torques in the speed reducer are lower with the high slip motor for the same average speed.

MOTOR CHARACTERISTICS

Standard Designs

Squirrel cage motors that may be considered for oil well pumping are as follows:

Design B - Normal starting inrush current
Normal starting torque
Normal slip - 5% or less.

Design C - Normal starting inrush current
Medium high starting torque
Normal slip - 5% or less

Design D - Normal starting inrush current
High starting torque
High slip - two designs - 5-8% and 8-13%

Note - For specific Kva inrush, locked rotor torques and breakdown torques, refer to ASA or NEMA Standards.

Design B

The Design B motor is the least expensive of the three designs considered. However, it does not have the starting and accelerating torque characteristics or full load slip desired for a walking beam sucker rod pump drive. Because of the low starting and accelerating torque of the Design B motor and the unpredictable starting torque requirements of the pump, Design B motors generally should not be used on automatically controlled pumps. The probability of a locked rotor in cold icy weather on small motors is greater than on large motors because the friction load is a higher percentage of the total load.

Design C

Design C motors are satisfactory in most climates and are commonly used on automatic leases where starting is accomplished by a program time switch. This is particularly true where the required motor rating is 10 horsepower or greater.

Design D

The Design D motor torque characteristics are adequate for walking beam sucker rod pump drives in any climate and for automatically controlled leases. The high starting and accelerating torque of the Design B motor assures positive breakaway and acceleration for all normal conditions encountered.

The motor torque varies as the square of the terminal voltage (Torque $\propto V^2$). Thus, the system voltage regulation for starting should receive as much consideration as the motor design selected. This may dictate that Design D be specified to develop the required starting and accelerating torques.

Effects Of Cyclic Loads

The magnitude of the peak and minimum power requirements (two of each per stroke of the sucker rod) is influenced by the full load slip of the motor and pumping speed as well as the polished rod load. Normal slip Design B and C motors will follow the instantaneous torque requirements of the pump closer than the Design D motor. Therefore, the normal slip Designs B and C motors draw higher cyclic peak power from the system for a given average horsepower than does the high slip Design D.

Design D motors are assisted in riding through the peak torques by energy stored in the pumping unit or flywheel effect, since the motor slows down as the high torques are imposed. For the same strokes per minute, the minimum power demand of the high slip motor is higher than for the normal slip motor, because the motor is accelerating the load for this portion of the cycle.

As pumping speeds are increased, the average horsepower and the peak power demands increase at a much higher rate than the speed is increased. Also, with higher pumping speeds, the peak power of a normal slip (Designs B and C) motor increases at a much higher rate than that of a high slip (Design D) motor. For these reasons, there is a practical maximum pumping speed above which the overall pumping efficiency becomes exceedingly poor. Also, it should be noted that the Design D is better suited for high speed pumping.

The effect of cyclic load is to raise the required nameplate rating of the motor above the average motor horsepower load determined by the formula:

$$\text{Average Motor Hp.} = \frac{\text{Hyd. hp} + \text{Subsurface losses}}{\text{Surface efficiency}}$$

The rating of an electric motor is based on a permissible temperature rise. This temperature rise is a function of the motor losses, which are largely I^2R losses. Therefore, the motor heating is a function of the thermal current or RMS (root-mean squared) current. The RMS current is the square root of the mean of the squares of the currents of definite time intervals.

The average motor current is proportional to the pumping load. If the pumping load is constant and noncyclic, the average current drawn by the motor is equal to the RMS or thermal current. On cyclic loads, such as beam type pumps, where high peak currents are drawn, the RMS or thermal amperes exceed the average current. The ratio of the RMS current to the average current equals the "Cyclic Load Factor".

$$\text{CLF} = \frac{\text{RMS Current}}{\text{Average current}}$$

The thermal loading of a motor is a function of the cyclic load RMS current rather than the average horsepower delivered by the motor. For the same pumping strokes per minute, a normal slip motor, larger than a high slip motor, is required. Cyclic load factors are used to determine the proper size motor from a thermal standpoint for the average horsepower required to drive a pump. These cyclic load factors (CLF) have been determined by test for various polished rod average speeds, type of pump and design of motor, and are given in Table 1.

Other Considerations

High slip motors have a lower rated efficiency than normal slip motors for a constant load application; however, on cyclic loads, the lower peak load results in a lower RMS or thermal load. The lower RMS current results in lower power input (both motor and system), which more than compensates for the lower published efficiency. Also, the resultant P.F. of the Design D is approximately 20 percentage points higher than Designs B and C for normally loaded oil well pumping motors.

The characteristics of the Design D high slip motors are well suited to the requirements of walking beam sucker rod pumping. The 5-8% design usually affords adequate reduction in peak current values for most applications. Further reduction in current peaks using 8-13% slip motors does not justify the additional price of the motor.

The Design C motor is probably more commonly used than the Design D. However, more consideration

Table 1

CYCLIC LOAD FACTORS FOR NORMAL SLIP AND HIGH SLIP MOTORS, WITH ROTARY OR BEAM TYPE COUNTERWEIGHTS

	Multiplier to obtain Nameplate Horsepower from Average Horsepower			
Polished Rod Average Speed - Inches Per Min. (Stroke x SPM x 2)	Rotary Counterweights		Beam Counterweights	
	NS Motor	HS Motor	NS Motor	HS Motor
Up to 1500	1.10	1.05	1.10	1.05
1500 to 2000	1.20	1.10	1.20	1.10
2000 to 2500	1.30	1.15	1.40	1.25
2500 to 3000	1.40	1.20	1.55	1.35

is being given to the use of Design D for the reasons mentioned above. In selecting a motor, the horsepower/dollar cost is in favor the Design D (5-8% slip) even though, on an equivalent horsepower rating, the Design D cost 5% more.

TRIPLE AND DOUBLE RATED MOTORS

Triple rated constant speed oil well pumping motors are worthy of consideration where, water encroachment is contemplated, flow is initially aided by gas or water pressure, and where production is on a prorated basis. The smallest rating of the motor is used initially. As additional fluid is pumped or higher hydraulic horsepower requirements are imposed to make the allowable production, the motor is reconnected for the higher horsepower ratings.

For oil well pumping service, the triple rated motor is available in ratings 3-2.3-1.2 through 30-22.5-12 horsepower in Designs C and D. These motors, when connected for the horsepower required, afford operation at the most favorable efficiency and power factor.

Double rated motors are also available which are the same as the triple rated except the intermediate rating is deleted.

SINGLE PHASE MOTORS

A small percentage of electrified leases are served by single phase power using single phase motors for pump drives. Such applications are usually limited to small horsepower ratings (5 HP and below); however, larger single phase motors are sometimes used where the distribution circuit serving the lease will permit the starting inrush and resultant voltage regulation.

The capacitor-start-induction-run motor up to 3 horsepower and the capacitor-start-capacitor-run motor 5 - 10 horsepower (both rated 230 volts) single phase are most commonly used. These motors have high starting torque and low starting current.

Due to the presence of the switch or relay for transferring from the start to the run position with single phase motors, particular caution should be observed regarding application in gaseous atmospheres.

For application where requirements exceed 10 h.p. and the current inrush must be limited to the inrush of a 10 horsepower motor, a special 20 h.p. single phase capacitor-start-capacitor-run motor with starting inrush currents comparable to a 10 hp motor may be used. This low inrush is accomplished by connecting the starting winding and starting capacitors in series with the running winding.

After running speed has been reached, the control switches the windings to the conventional connection. The starting torque on this special motor is somewhat better than a 10 h.p. motor. Before applying this motor on automatic pumping units, the starting torque requirements of the pumps should be carefully considered with the motor manufacturer.

HORSEPOWER RATING BY CALCULATIONS

The factors entering into the determination of motor horsepower rating for oil well pumping are:

1. The hydraulic horsepower of the pump.
2. The subsurface losses.
3. The surface losses which determine surface efficiency.
4. Allowance for the effect of the cyclic nature of the pumping load on the heating of the motor, and consequently on its required nameplate rating, as compared to the calculated average horsepower.

The required motor rating may, therefore, be expressed as follows:

$$\text{Motor h.p. rating} = \frac{\text{Hyd. h.p.} + \text{subsurface losses}}{\text{surface efficiency}} \text{ X CLF.}$$

Any formula for motor horsepower rating must recognize all of these factors. However, because subsurface losses and the CLF vary widely with well characteristics and conditions of operation, it is not possible to establish one formula with the value of these factors fixed, to give an accurate motor horsepower for all wells. The local conditions must always be considered, and formula factors evaluated accordingly.

Other methods or formulas may be used to estimate or calculate the motor horsepower, but calculated values should always be checked by test at the polished rod or at the motor, as soon as possible.

Hydraulic Horsepower

Hydraulic horsepower is based upon the rate of production, the net fluid head on the pump and the specific gravity of the fluid in the tubing. It may be expressed by the formula:

$$\text{Hyd. h.p.} = \frac{Q \times H \times \text{Sp. Gr.}}{135{,}600}$$

H = Effective fluid head or distance in feet from ground surface to fluid surface in casing.

Sp.Gr. = Specific Gravity of fluid column in the tubing.

Q = The rate of plunger displacement per 24 hour period or the continuous capacity of the pump for a 24 hour period. For example, if 50 barrels of fluid are pumped in 12 hours, the production rate would be 100 barrels; making Q = 100. However, if there is gas present in the fluid and Q is determined by the fluid volume/gas content, then Q would be the 24 hour production rate times (100% + percentage of gas by volume).

Q may be calculated for any equipment from the formula:

$$Q = \frac{(D^2SN) \text{ x pump displacement eff.}}{8.583} = \text{bbls. per 24 hrs.}$$

D = Diameter of pump barrel, in inches.

S = Plunger stroke length, in inches (discussed below).

N = Number of strokes per minute.

Pump Disp. Eff. - The displacement efficiency of a modern pump in good condition is probably not less than 95%. Using this value of pump displacement efficiency, the formula becomes:

$$Q = \frac{D^2SN}{9} \text{ barrels of fluid that can be pumped in 24 hrs.}$$

Effective Plunger Stroke

It is commonly recognized that the actual plunger stroke is greater or less than the polished rod stroke. This is because the string of sucker rods is quite elastic. With a load alternately applied and removed at the bottom, the travel of the plunger may be of a different magnitude than that of the polished rod at the top. The action is similar to oscillating a small weight on the end of a rubber band, the weight being removed on the down stroke. Obviously, if Q is to be calculated with the greatest degree of accuracy, the actual or effective plunger stroke of the pump should be used for S in the formula rather than the polished rod stroke on the surface.

The effective plunger stroke may be calculated from a formula which is based on theoretical consideration and test results. A simplified form of the formula, to permit use in the field, is given below.

$$EPS = \frac{S}{\text{Cosine } (0.0004LN)^{\circ}} - \frac{W_{OD}L}{2\,(10)^{6}}\left(\frac{1}{A_R} + \frac{1}{A_T}\right)$$

EPS = Effective plunger stroke, in inches.
S = Polished rod stroke, in inches.
L = Length of rod string in feet.
N = Number of strokes per minute (SPM)
W_{OD} = Differential fluid load (weight of net fluid head on full area of plunger, in pounds)
A_R = Cross sectional area of sucker rod, in square inches.
A_T = Cross sectional area of tubing, in square inches.

In the absence of a table of trigonometric functions, the cosine of the angle expressed as $(0.0004\ LN)^{\circ}$, may be calculated from the equation:

$$\text{Cosine } (0.0004\ LN)^{\circ} = 1 - \frac{X^2}{2} + \frac{X^4}{24}$$

$$X = \frac{0.0004\ LN}{57.3}$$

For unanchored tubing, eliminate the term $1/A_T$.

This simplified formula, which neglects the damping factor, may be used for any well condition which does not approach a condition of resonant pumping speed. (Resonant pumping speed exists when, 4 x string length in feet x strokes per second = 17,000). For such critical conditions, the complete formula must be used.

The effective fluid head may ultimately be the total distance from the ground surface to the pump when the well is pumped off and the tubing is full of oil. Consideration of expected well conditions over the life of the well should determine the proper value for H.

The Specific Gravity value used should represent that of the composite fluid column in the tubing - oil, water, and gas. Ultimately this value will probably approach 1.0, when the well is "dead" and water content is high.

Subsurface Losses

Subsurface losses vary to an unpredictable degree, due to the effect of such factors as crookedness of hole, angle of deflection of modern directionally drilled wells, sand in fluid, paraffin accumulation in tubing, specific gravity, and viscosity of fluid pumped. It has been common practice to allow for subsurface losses, along with surface losses and all other factors, by simply multiplying the hydraulic horsepower by a single factor to obtain the required motor horsepower. The factor most commonly used is 2.

Such a practice obviously assumes that the subsurface losses are some multiple of the hydraulic horsepower. This assumption does not seem logical, due to the nature of the factors which affect the magnitude of the subsurface losses. For any given hole, the principal string losses are due to sliding friction between rod and tubing, friction of the fluid moving upward in the tubing and the friction of the rods and couplings moving through the fluid. All these components increase proportionately with length of string, but not necessarily with size of plunger - which does directly affect the hydraulic horsepower.

For a modern straight hole, the losses may not vary materially with change in plunger size, but only with speed of pumping. It, therefore, seems logical to add to the hydraulic horsepower an increment for string losses, rather than simply consider the string losses as some multiple of the hydraulic horsepower.

Results of tests on a 3500 foot dead well, with a maximum crookedness of 4°, pumping practically all water, are shown by Table 2. The values vary little with plunger size, but considerably with speed of pumping. The curve of Fig. 1 is plotted from average values for the three plunger sizes.

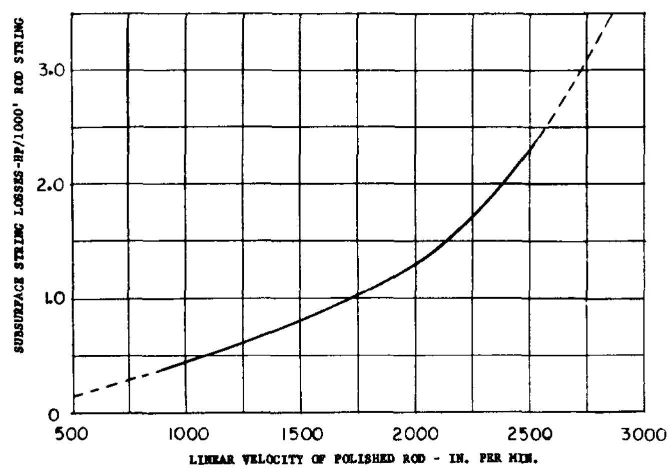


Fig. 1 CURVE OF SUBSURFACE LOSSES BASED ON AVERAGE VALUES FROM TABLE II

Values higher than those shown by the curve would, no doubt, be proper for crooked or deflected wells, for wells with fluid of high viscosity, or for wells with paraffin accumulation. Proper values should be determined, as soon as possible, by test or by calculations based on test.

The difference between horsepower values, shown by a polished rod dynamometer and the calculated hydraulic horsepower at the pump, should give reasonably accurate values of string loss. Values so determined on a few wells in a field should be applicable to other wells in the same field or other fields with similar conditions, where actual tests are not yet possible or desirable.

Surface Losses Or Efficiency

Surface losses or efficiency may be expressed as the ratio of the input horsepower to the horsepower delivered at the polished rod. These losses occur in the bearings, gears and V-belts. Published pumping unit efficiencies vary up to 85 per cent. Depending on the per cent loading of the pumping unit reduction gear and the motor, the overall surface efficiency varies from 40-75%. Since neither the pumping unit nor motor are generally fully loaded, a more practical maximum surface efficiency is 60%.

Sample Calculation

Assume a well depth and fluid head of 4000 feet, rod size 3/4 inch, tubing size 2-1/2 inches unanchored, pump barrel diameter of 1.25 inches, speed of pumping 18 SPM, polished rod stroke 44 inches (calculated effective plunger stroke 38 inches), pump displacement efficiency 95 per cent, Specific Gravity of tubing fluid column 1.0, subsurface losses from Fig. 1, surface efficiency 65 per cent, CLF from Table 1 for HS (high slip) motor and rotary counterweights.

$$Q = \frac{D^2 SN}{9}$$

$$= \frac{1.25^2 \times 38 \times 18}{9} = 120$$

$$\text{Hyd. hp} = \frac{Q \times H \times \text{Sp.Gr.}}{135{,}600}$$

$$= \frac{120 \times 4000 \times 1.0}{135{,}600} \quad 3.5 \text{ Hyd. hp}$$

Req. Mot. Rating:

$$= \frac{\text{Hyd. HP} + \text{Subsurface losses}}{\text{Surface efficiency}} \times \text{CLF}$$

$$= \frac{3.5 + 3.8}{.65} \times 1.1$$

$$= 12.5$$

Use 15-hp, high slip motor.

Quick Approximation Formula:

The following formulas may be used for quickly estimating the approximate motor horsepower, at relatively low pumping speeds (below 2000 inches per minute at polished rod) and when well conditions are favorable.

For normal slip motors $\frac{D^2 SNH}{450{,}000}$ = required motor hp

For high slip motors $\frac{D^2 SNH}{540{,}000}$ = required motor hp

These formulas assume a fixed ratio of average motor horsepower to hydraulic horsepower of approximately 2, and a cyclic load factor of 1.33 for the normal slip motor and 1.1 for the high slip motor.

RATING BY TEST

The empirical methods, outlined above, for applying electric motors to sucker rod pumping will give an approximation of the horsepower required; however, the subsurface losses and surface losses may cause the actual horsepower to vary considerably. For this reason, it is necessary to make electrical tests to verify the calculated horsepower requirements when the motor is placed in service. Also, periodic checks of power requirements are desirable to determine both the surface and subsurface losses, as well as the productive load of the pump.

Very simple electrical tests, using only the following meters, will give the required information to make a complete electrical analysis:

1. Clip on ammeter.
2. Watthour meter.
3. Thermal ammeter (magnetic circuit opening variety commonly used).
4. Voltmeter (indicating or recording - 1000 Ohms/-Volt or greater).
5. Pump off recording meter (used when wells are not pumped continuously).

Modern day oil well pumping controls incorporate provisions within the control cabinet for using these meters. A metering loop, which is a part of the power circuit to the motor, is stored in the bottom of the control cabinet and may be used with either clip-on or thermal ammeters when they are equipped with an opening magnetic circuit. A watthour meter socket is provided for the pump off recorder and other plug in base meters and instruments.

Table 2 - TEST VALUES OF SUBSURFACE LOSSES ON 3500-FOOT WELL STRING OF 3/4-INCH RODS WITH 42-INCH POLISHED ROD STROKE

STROKES PER MINUTE	TEST WITHOUT PLUNGER LOAD	TEST WITH 1.25 INCH PLUNGER	TEST WITH 1.75 INCH PLUNGER	AVER. LOSS VALUE HP/1000 FEET OF ROD STRING	POLISHED ROD LINEAR VELOCITY FEET/MINUTE
10	1.25	1.22	1.24	.35	844
16	2.40	2.27	2.95	.73	1350
24	3.94	4.50	5.58	1.33	2025
30	8.63	7.34	9.10	2.39	2532

The electrical tests performed will determine the following:

1. Voltage
2. Instantaneous amperes.
3. Kilowatts (average).
4. Thermal amperes (45-minute reading).
5. Pump off of the well.

When a well is first placed in service, a clip-on or A-C ammeter is utilized to properly balance a counter-weighted pumping unit. Correct balance results in the lowest current demand and the minimum deviation during a complete cycle of the pump.

The average kilowatt input to the motor is determined by the seconds of time it takes the disc of the kilowatt-hour meter to make a given number of revolutions, or

$$KW_{avg} = \frac{\text{Meter "}K_h\text{" constant x Disc RPM x 60 minutes}}{1000}$$

From the Average KW, the horsepower is determined.

By

$$HP_{avg} = \frac{KW_{avg}}{.746} \text{ x Motor efficiency.}$$

Typical motor full load efficiencies are approximately = 0.85

Thermal amperage is the root mean square of the continuous cyclic load current. This factor determines the motor thermal capacity to carry load imposed on it. The nameplate current rating of the motor should not be less then the thermal amperage measured by the thermal ammeter.

Using the thermal current, average kilowatts, and average line voltage, the average power factor may be determined by using the following formula:

P. F. (Average) =

$$\frac{KW_{avg} \text{ x } 1000}{1.73 \text{ x Amps. (Thermal) x Volts (aver. line-to-line)}}$$

If it is desired to determine the cyclic load factor (CLF), the average reactive power drawn by the motor should be obtained using a reactive meter or a watthour meter connected to read reactive power, then use the formula below.

$$KVA_{avg.} = \sqrt{(KW_{avg.})^2 + (KVAR_{avg.})^2}$$

$$I_{avg.} = \frac{KVA_{avg.}}{\sqrt{3} \text{ x } V_{LL}}$$

$$CLF = \frac{\text{RMS Amps.}}{\text{Average current}} = \frac{I_{RMS}}{I_{avg.}}$$

The pump off recorder is commonly used to determine the time required to pump the fluid level down to the pump plunger. The instrument commonly used is a recording thermal ammeter. A typical chart from one such instrument is shown in Fig. 2. Automatic well pumping programs may be derived from such charts.

Test On Sucker Rod Using Dynamometer

The dynamometer draws a closed diagram which records the polished rod load throughout the stroke. A timing wave is also provided. From these records, the following may be obtained.

1. The maximum and minimum polished rod loads.
2. The average horsepower at the polished rod.
3. The velocity and acceleration of the polished rod during any portion of the stroke.
4. The strokes per minute.

A sucker rod pumping unit is a mechanical system which is quite elastic. It consists of a string of small diameter rods, reaching possibly more than a mile below the surface of the earth to a plunger pump, actuated by surface equipment consisting of a counter-balanced walking beam, a connecting rod, a crank arm speed reducer, belt drive and a prime mover.

Due to its elasticity, it provides many possibilities for variation in peak and average load with changes in pumping speed or stroke length, or both. To these possibilities must be added factors affecting well conditions, such as casing fluid level, gas and water content of fluid, and viscosity of fluid. It is practically impossible to arrive at reliable conclusions from tests taken without all surface and subsurface conditions being known.

In order to learn how loads and losses vary in the several members and portions of the system, a set of very comprehensive tests were made by Westinghouse and a leading producing company on one 3500 foot "dead" well. Magnetic strain gauges and other test devices were placed on various members of the system from the pump to the prime mover.

Figs. 3 and 4 show test values of peak polished rod loads and crankshaft torques plotted against speed of pumping. Fig. 5 shows a comparison of actual plunger stroke and surface polished rod stroke, at various strokes per minute. During these tests, only one factor was varied at a time. The factors varied were speed of pumping, polished rod stroke length, type and amount of counterweighting and type of prime mover.

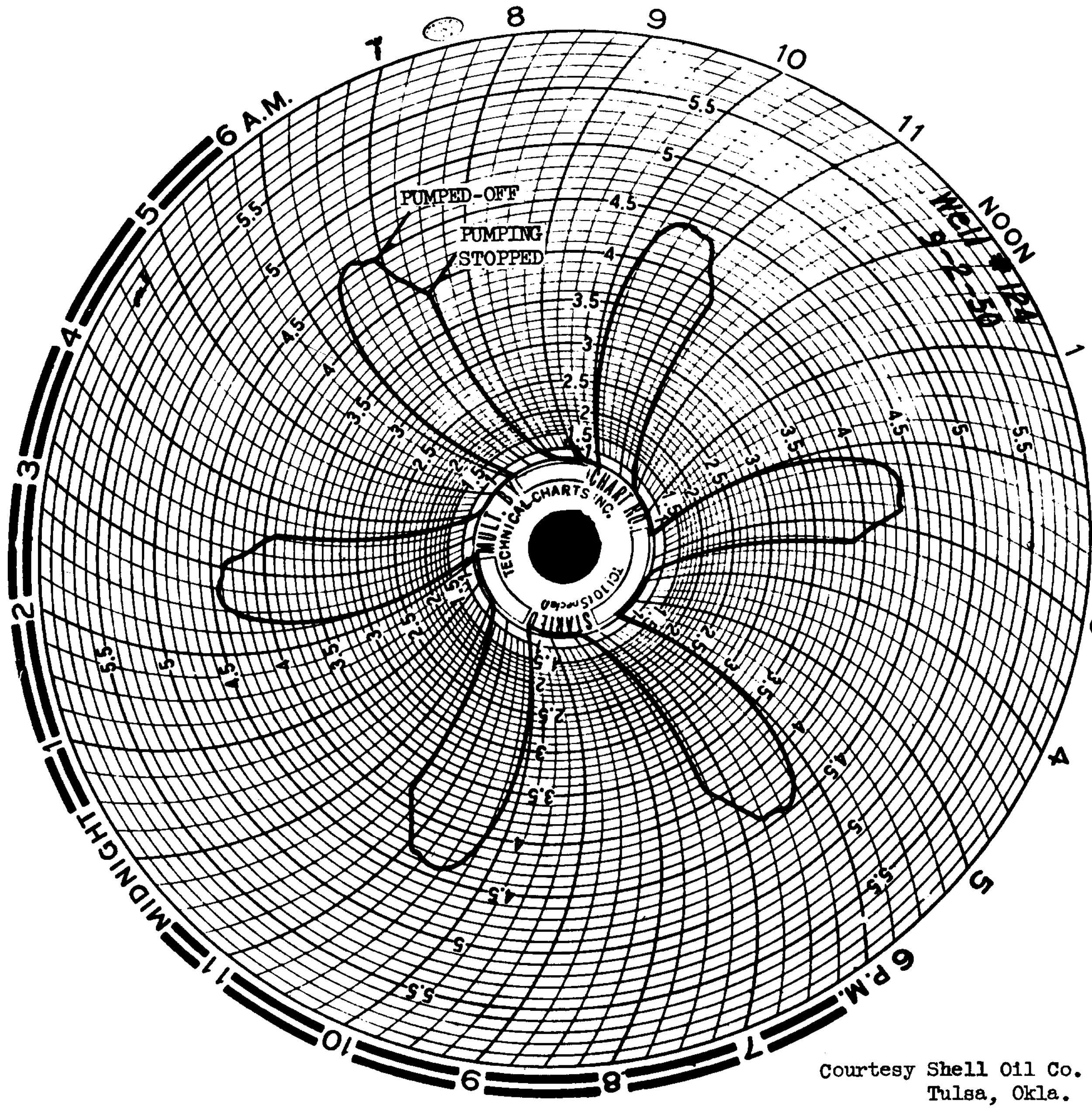
The polished rod horsepower may be determined from the dynamometer card and may be used with motor input horsepower to determine the surface losses or efficiency from the following formula:

$$\text{Surface Efficiency} = \frac{\text{Polished rod horsepower}}{\text{Motor input h.p. x motor eff.}}$$

MOTOR ENCLOSURES

Classifications

Motor enclosures, as defined by NEMA, are classified as either "OPEN MACHINES" or "TOTALLY ENCLOSED MACHINES". These two enclosures are further divided into several classifications, of which the following are more commonly used for oil well pumping.



Courtesy Shell Oil Co.
Tulsa, Okla.

Fig. 2 TYPICAL PUMP-OFF RECORDING CHART

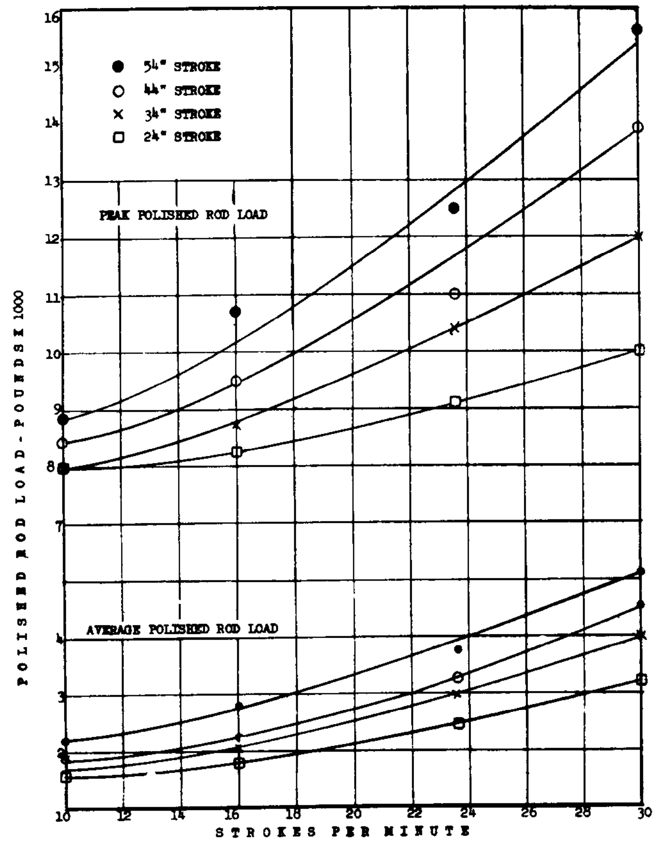


Fig. 3 PEAK AND AVERAGE POLISHED-ROD LOADS DETERMINED WITH DYNAMOMETER

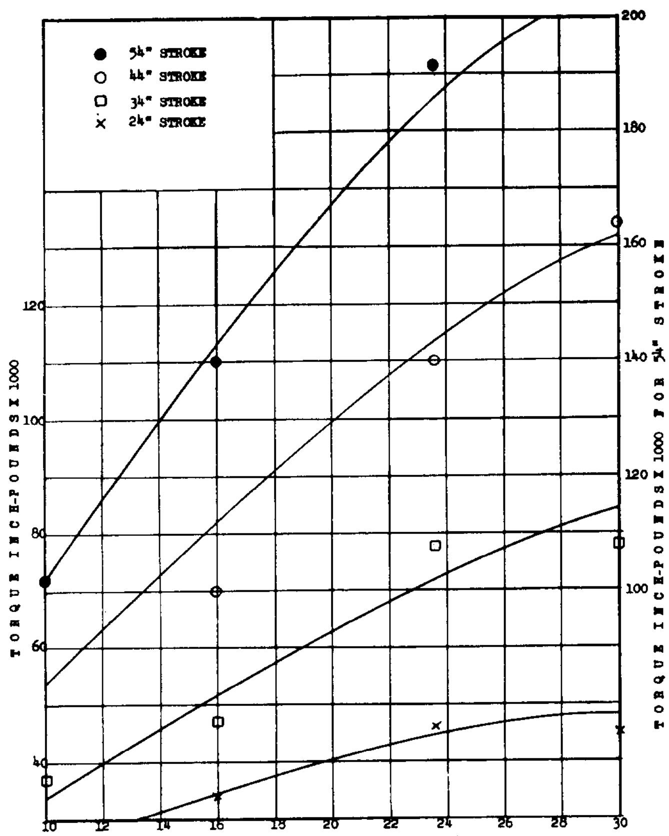


Fig. 4 PEAK CRANKSHAFT TORQUES, NORMAL-SLIP HIGH -TORQUE MOTOR, 3,500 FOOT WELL-ROTARY COUNTERWEIGHTS

A. Drip proof, guarded.
B. Splash proof, guarded
C. TEFC - Totally enclosed fan cooled.

An open motor has an enclosure which permits the passage of cooling air over and around the stator winding and rotor. Fans, which are an integral part of the rotor, draw air in one end and discharge it at the other or draw air in at both ends and discharge it at the center through openings provided in the frame.

In the totally enclosed machine, there is no free exchange of air between the inside and the outside of the enclosure. The enclosure, however, is not termed air tight.

Enclosures Used

Motors used in the oil fields are suitable for outdoor applications. These are drip proof, splash proof and TEFC. The open enclosures are usually equipped with a screen over the opening to prevent the entrance of rodents and large particles.

The drip proof, guarded enclosure is almost universally accepted. This enclosure is so constructed that liquids or solid particles falling at any angle not greater than 15 degrees from the vertical cannot enter the motor either directly or by striking and running along a horizontal or inwardly inclined surface of the motor.

Splash proof, guarded enclosures are still offered in the larger frame sizes (Frames 364-505S) and are

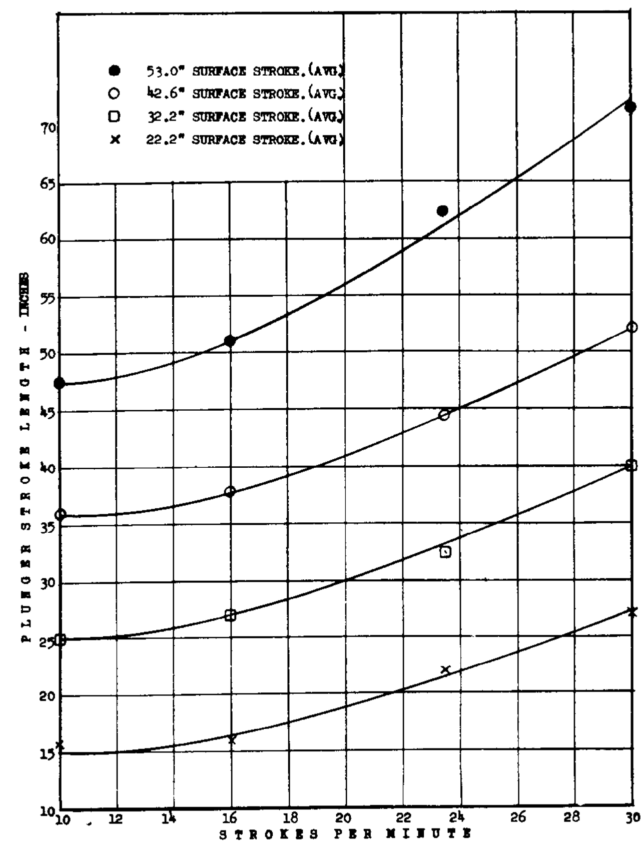


Fig. 5 VARIATION OF ACTUAL PLUNGER STROKES WITH STROKES -PER - MINUTE (SPM) 3,500-FOOT ROD STRING- 3/4 - INCH ROD, 1 - 1/4 INCH DIAMETER PLUNGER

suitable for oil field applications. The splash proof motors are constructed so that drops of liquid or solid particles falling on them at any angle not greater than 100 degrees from the vertical cannot enter the motors either directly or by striking and running along a surface of the motor.

The TEFC motor is recommended where the weather is very severe such as coastal areas or in corrosive atmosphers. In many cases, the use of TEFC motors is justified for very dusty areas. If the atmosphere is considered hazardous, an explosion proof TEFC motor is recommended. The atmosphere around an oil well is not normally hazardous because the well is capped when the motor is operating.

MOTOR BEARINGS

General

Oil well pumping motors are available with either prelubricated or regreasable ball bearings. The ball bearing is universally accepted on small fram size motors. The reasons for this popularity are high thrust capacity, high radial load capacity, compactness, relative freedom of wear and symmetrical design, permitting load to be carried in any direction.

Another important feature of the ball bearing is the use of grease lubrication. This eliminates the problem connected with oil lubrication and minimizes maintenance expense through the elimination of periodic greasing requirements.

The object of the lubricant in the ball bearing is to reduce the friction between the balls and their retainer and to provide protection against corrosion of the highly polished surfaces of both balls and raceways. The small amount of lubricant required by the ball bearing can readily be supplied by grease, which is oil held by a suitable absorbent, usually a sodium or lithium soap.

If the minute quantity of grease required to lubricate the ball bearing is kept clean and protected against the elements of the atmosphere, it is adequate for many years' service. Following the development of adequate seals, the prelubricated ball bearing has been applied on a general basis since 1941.

PreLubricated Ball Bearings .

Fundamentally, the prelubricated ball bearing does not differ from the conventional open ball bearing. It is made from the same material and to the same specifications with regard to tolerances and fits. It is rates to carry the same load and have the same life as the regreasable ball bearing. Prelubricated bearings supplied on Westinghouse motors are all single row bearings of double row width. This design provides ample space for the double labyrinth seal and a liberal quantity of grease for the ball and retainer assembly.

Fig. 6 illustrates the construction of the double labyrinth seal which most effectively retains the grease within the bearing and prevents the entrance of foreign material.

The organic grease used in standard open and splash proof motors is suitable for operating temperatures from -40°C to + 90°C. However, to obtain an extended life, the bearing operation is limited to -25°C to + 65°C. The upper limit of 65°C is a very conservative value, especially so if the high temperature is due to temporary (day-to-day) high ambient on a seasonal basis.

At high operating temperatures, there is a gradual reduction in the life of the grease, progressing more rapidly as the temperature is increased. All prelubricated bearings, filled with grease for -25°C to + 65°C operating temperatures, are identified by bright metal shields in the seals.

For bearing operating temperatures up to 100°C, a specially compounded silicone grease is used in prelubricated bearings. These bearings are identified by having a blue annealed or black steel shield.

For bearing operating temperatures up to 125°C, which is a practical limit for any bearing, the bearings are heat stabilized in manufacture. Such bearings are stamped or etched with the letters "STB" in addition to having black shields.

Bearings for extremely low temperatures (-25 to -70°C) require a special grease. Several types of this kind of bearing are available. These bearings are identified by bright shields and the letters "LT".

The risk of losing the grease, and eventually the bearing, by the grease leaking out through the seals, is a question frequently raised. However, this has not been a problem with the seals and grease approved for the prelubricated bearing, despite applications where exceptionally high temperature tends to make the grease more fluid.

A look at the melting point or drop point value for any good grade of commercial grease shows that this temperature lies between 180° to 250°C. These values are so much above any temperature reached in applications at which the bearing is expected to operate that there is no danger of leakage even where the temperature is higher than normal. If a bearing shows some leakage around the seal, it must be concluded that excess grease is running out or that the seal is defective.

If such leakage occurs it is most likely due to an excess grease charge, and the bearing in running has thrown off or purged itself of the excess. The amount thrown off is very small and cannot do any damage to the motor. If wiped off, it will probably never reoccur. Small leakage from a new bearing is a guarantee that the bearing is fully charged, and that the small cracks around the seal are filled with grease thus sealing the bearing more effectively against entrance of foreign matter.

Bearing Sizes

Bearing sizes are selected for a given frame size motor by the most severe application that can be imposed on the motor. On this basis, the average life expected is 88,000 hours or 10 years continuous duty and a minimum life of 17,500 hours or 2 years continuous duty. It is expected that not more than 10% of the bearings on a certain application shall fail within the minimum life interval for the calculated load.

Even in tough applications, like belted walking beam sucker rod pumps using NEMA minimum pulleys, the calculated bearing load rarely approaches the limits permitted. For coupled motors, the bearing load is normally so far below the rated values that its life may be expected to be several times the above mentioned values.

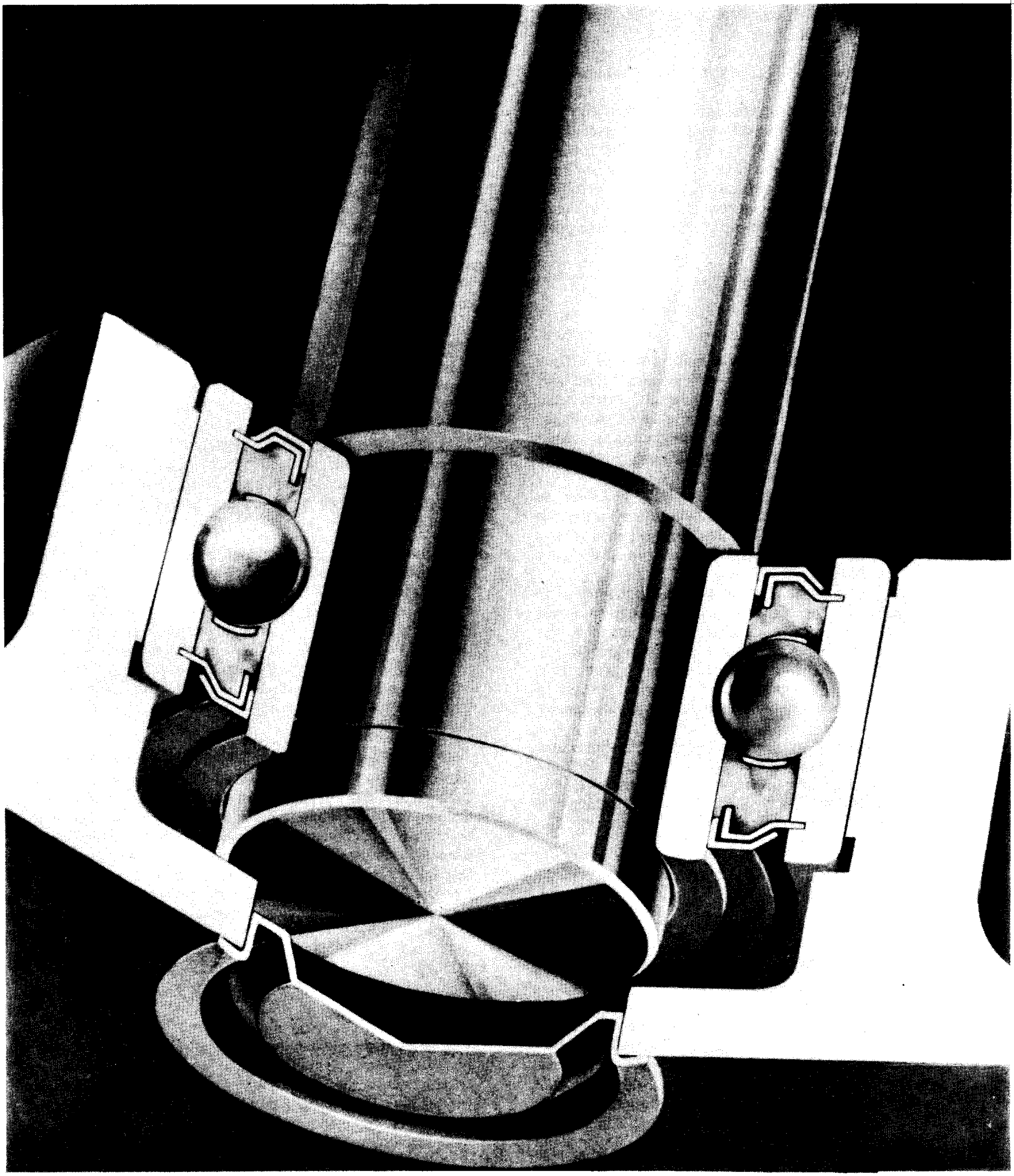


Fig. 6 CONSTRUCTION OF DOUBLE LABYRINTH SEAL BALL BEARING

INSULATION AND TEMPERATURE

Atmospheric conditions, ambient temperature and users' preference dictate the type of insulation in the application of a motor. ASA and NEMA have established the classes of insulation and maximum temperatures applicable to these classes.

For normal service life, the temperature of motor windings should not exceed the maximum allowable temperature for a particular winding utilizing Class A, B, or H insulations. The maximum internal temperatures for various classes of insulation are:

Class A Insulation . . . . 105°C
Class B Insulation . . . . 130°C
Class F Insulation . . . . 155°C
Class H Insulation . . . . 180°C

If rotating equipment is operated at lower temperatures than stated, longer operating life can naturally be expected from the windings. On the other hand, if such temperatures are exceeded consistently, an appreciably shorter insulation life will result.

To express insulation life in another manner, it has been determined from study and analysis of various applications and failures that for every 10 degrees C rise above the temperature values stated, the insulation life is cut approximately in half.

Insulation Composition

At this point, perhaps it would be advantageous to discuss briefly the definition of the various classes of insulation referred to - namely, Class A, B, and Full Class H, as standardized by ASA.

Class A

Class A Insulation consists of: 1. cotton, silk, paper, and similar organic materials when either impregnated or immersed in a liquid dielectric; 2. molded and laminated materials with cellulose filler, phenolic resins, and other resins of similar properties; 3. films and sheets of cellulose acetate and other cellulose derivatives of similar properties; and 4. varnishes (enamel) as applied to conductors.

Class B

Class B insulation consists of mica, asbestos, fiberglass, and similar inorganic materials in builtup form with organic binding substances. A small proportion of Class A materials may be used for structural purposes only. Fiberglass or asbestos magnet wire insulations are included in this temperature class. These may include supplementary organic materials, such as polyvinylacetal or polyamide films.

Class H

Class H insulation consists of: 1. Mica, asbestos, fiberglass, and similar inorganic materials in builtup form with binding substances composed of silicone compounds, or materials with equivalent properties; and 2. silicone compounds in rubbery or resinous forms or materials with equivalent properties. A minute proportion of Class A materials may be used only where essential for structural purposes during manufacture.

Temperatures

The internal winding or hotspot temperature is the critical temperature and is of the most concern. Actually, the internal temperature of windings is the sum of four factors. They are:

1. The "ambient temperature" which is the temperature of the air immediately surrounding the equipment.
2. The "temperature rise" which is the number of degrees that the hottest part of the winding, accessible for thermometer measurement, rises above the ambient temperature. This is the temperature rating that is usually indicated on the nameplate attached to the equipment.
3. "Hot-spot allowance" which is the difference that can be expected between the hottest accessible part of the windings and the hottest inaccessible spot.
4. The "service factor" allowance which is the degrees rise allowable over the rated temperature rise occurring when the 15 per cent overload capacity is utilized.

Table 3 is a chart showing the allowable temperature rise by thermometer for open drip proof and TEFC Class A insulated motors which are standard classes of motors utilized for pumping service.

It is usual practice to make an analysis between the ambient temperature and the temperature rise. The manufacturer's nameplate on the equipment indicates a rating of temperature rise, by thermometer, that has been based on an ambient temperature of 40°C.

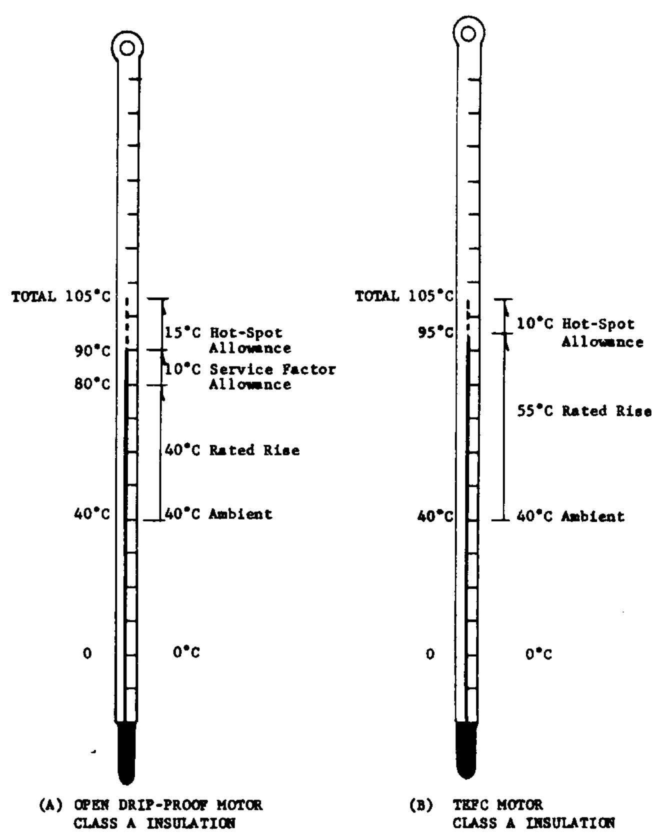


Table 3 ALLOWABLE TEMPERATURE RISE BY THERMOMETER METHOD

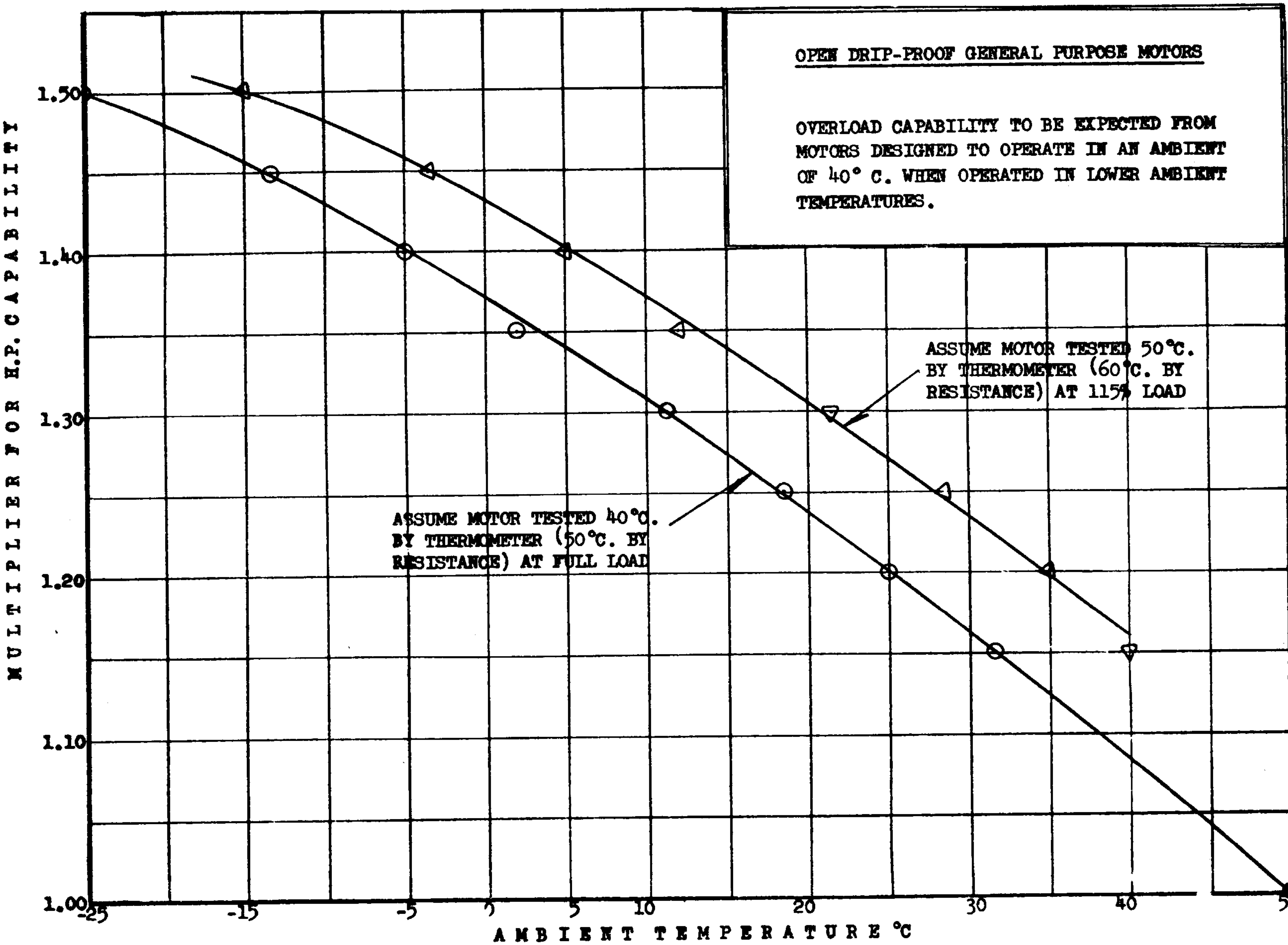
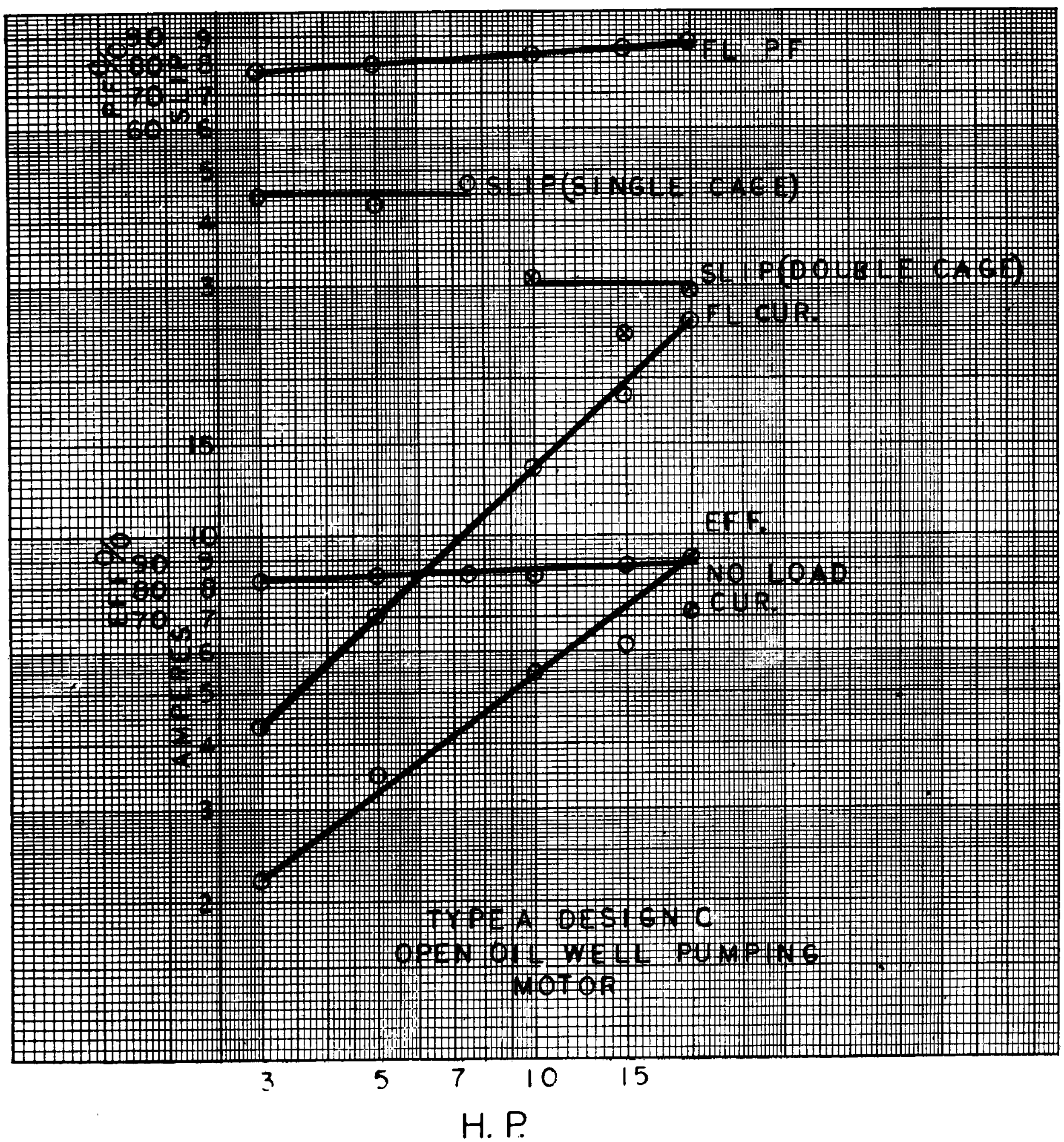


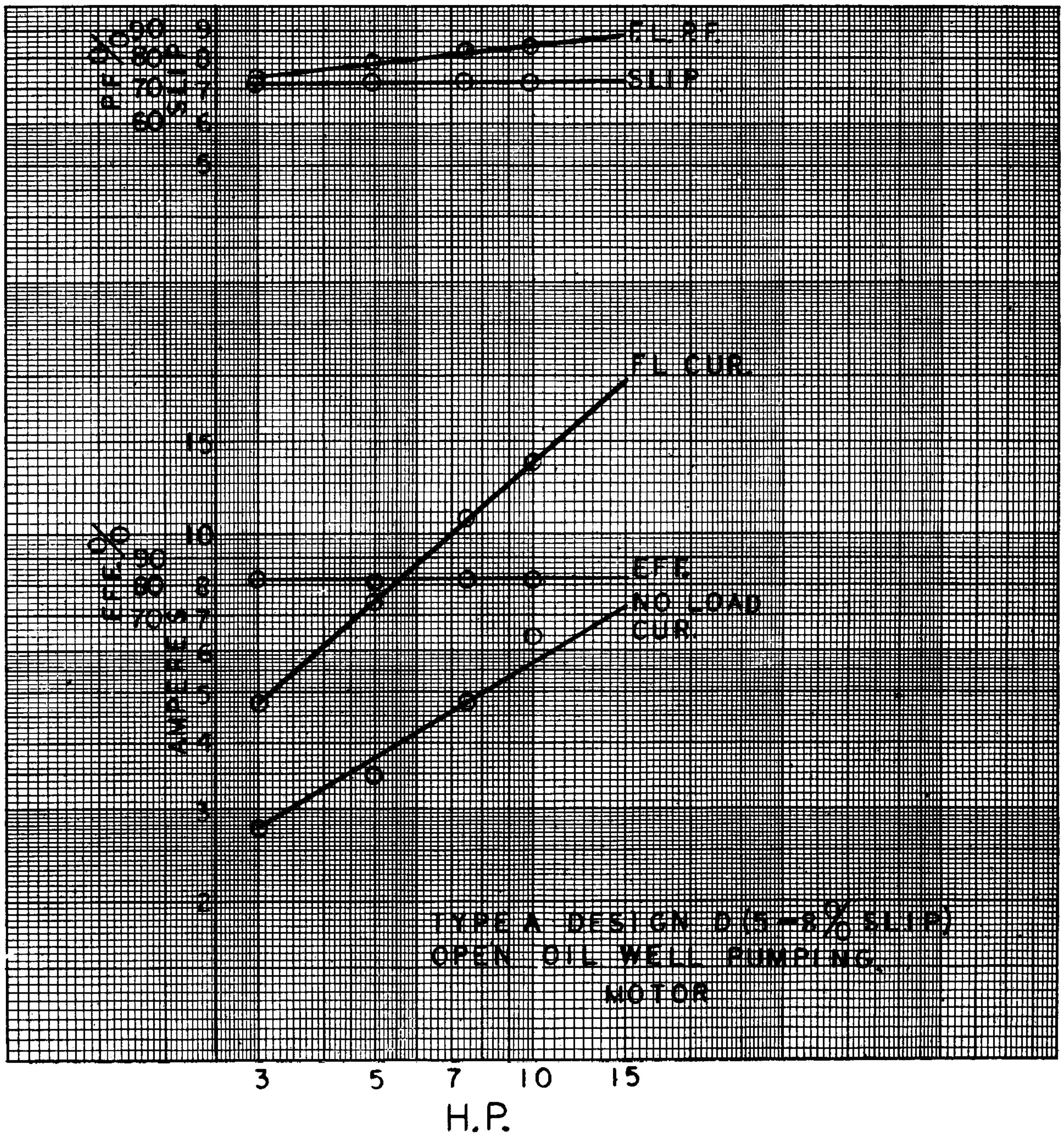
Fig. 7

MOTOR CAPABILITY VERSUS AMBIENT TEMPERATURE FOR OPEN DRIP-PROOF GENERAL PURPOSE MOTORS



P.F. %
90
80
70
60
SLIP %
9
8
7
6
5
4
3
EFF. %
90
80
70
AMPERES
15
10
9
8
7
6
5
4
3
2
FL PF
SLIP(SINGLE CAGE)
SLIP(DOUBLE CAGE)
FL CUR.
EFF.
NO LOAD
CUR.
TYPE A DESIGN C
OPEN OIL WELL PUMPING
MOTOR
3
5
7
10
15
H. P.

Fig. 8



PF %
SLIP
FL PF
SLIP
FL CUR.
EFF. %
AMPERES
EFF.
NO LOAD CUR.
TYPE A DESIGN D (5-8% SLIP)
OPEN OIL WELL PUMPING MOTOR
3
5
7
10
15
H.P.

Fig. 9

For example, if the ambient temperature is above 40°C (104°F) and the temperature rise is higher than the temperature value indicated on the nameplate, the insulation is being worked above its continuous rating and its life will be shortened. Conversely, if the motor is operating in an ambient temperature of less than 40°C, the allowable temperature rise may be increased, thereby increasing the motor continuous capability without exceeding the allowable temperature rise.

Fig. 7, Curves of "Motor Capability Versus Ambient Temperature For Open Drip Proof General Purpose Motors" is given. Note that if all factors (voltage, frequency, etc.) are at rated value, the service factor may be utilized; therefore, two curves are given.

OPERATION AT HIGHER ELEVATIONS

Motors are guaranteed to operate without exceeding the nameplate temperature rise at any elevation not exceeding 3300 feet. Most oil fields are at lower elevations. The temperature rise may be expected to increase one per cent of the nameplate rise for each 330 feet above 3300 feet. Therefore, the motors must be derated for higher elevations. The following table lists derating factors:

TABLE 4

DERATING OF MOTORS
OPERATING AT HIGH ELEVATIONS

Altitude Feet	Current or HP Derating %
3300	0
3630	.5%
3960	1.0%
4290	1.52%
4520	2.02%
6600	5.13%
8250	7.20%

REFERENCES

1. "Oil Well Tests Suggest New Rating Standards" By H. E. Dralle and E. H. Lamberger. Presented at the Annual Meeting of the American Petroleum Institute, November, 1951, in Chicago, Illinois.
2. "Calculation of Load and Stroke in Oil Well Pumping Rods" By B. F. Langer and E. H. Lamberger. Presented at the National Meeting of the Applied Mechanics Division, American Society of Mechanical Engineers, at Cambridge, Massachusetts, June, 1942.
3. "Oil Well Pumping Tests" By H. E. Dralle and E. H. Lamberger, Oil and Gas Journal, May 7, 1942.