Selection And Application Of Prime Movers For Oil Well Pumping

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INTRODUCTION:

Often the oil operator fails to recognize the importance of a careful study of the prime mover, yet each well that does not flow involves a problem in the selection and application of a suitable prime mover. Many formulas have been derived to determine the prime mover size. Basically, these formulas give essentially the same results when the same allowances have been made. An over all multiplier is generally applied without much thought as to the exact factors involved. Sometimes, very important factors are overlooked in obtaining an efficient, economical prime mover installation. Two of these factors are:

(1) Efficiency of pumping unit and V belt drive.

(2) Instantaneous peak torque converted to horsepower.

All of us are prone to think in terms of average steady horsepower instead of a varying load with relatively large peaks, which are encountered in oil well beam pumping. This requires a study of above factors to properly apply a prime mover.

In areas where there are similar pumping wells in the same field, dynamometer cards can be used to analyze load requirements and to develop torque curves plotted against crank angles. The torque curves can then be converted to horsepower. The result is instantaneous peak horsepower, useful in determining prime mover size.

It is important to remember that normally a unit and prime mover are purchased for the life of the well. Early pumping conditions due to high fluid levels, gas help, and other factors, may be quite different in the later pumping stages therefore experience is necessary in determining prime mover size. Conditions today may indicate a prime mover size too small for tomorrows condition.

FORMULAS:

In order to discuss factors related to selection of the size of prime mover, it is necessary to start with some of the usual formulae for determining prime mover size. There are:

1. *PRIME MOVER SHAFT BHP = HYDRAULIC HP X 2.5

Where:

Hydraulic HP =

$$\frac{\text{Bbls/da. X Wf X D}}{\text{33,000 X 24 X 60}}$$

Wf = Weight Fluid per Bbl. — Use 350 lbs.

D = Depth to Pump in Feet

2. *PRIME MOVER SHAFT BHP = POLISHED ROD HP X 1.5

Where:

Polished Rod HP =

$$\frac{\text{Bbls./da. X D}}{\text{85,000}}$$

3. *PRIME MOVER SHAFT BHP = (HYD. HP + Fr HP) 1.5

Where:

Hydraulic HP is Shown in Formula 1

And Fr HP =

$$\frac{\text{SPM X L X Wr}}{\text{1,600,000}}$$

SPM = Strokes per Minute
L = Length Strokes in Inches
Wr = Weight Rod String Pounds

The above formulae assume an overall unit and V belt drive efficiency of approximately 75 percent. It should be pointed out that this is not necessarily true and may lead to serious errors in selection of the prime mover

*This term is used to mean the actual horsepower at the prime mover sheave shaft. This is the size of the prime mover required after service factors have been applied.

size. Factors to be considered in the efficiency of the pumping unit are:

(1) Relatively high average structural loads (beam load) tends to reduce unit mechanical efficiency.

(2) Relatively high torque (polished rod HP) tends to increase unit mechanical efficiency.

THE R FACTOR:

Efforts have been made by oil companies and by pumping unit manufacturers to determine unit and drive efficiencies. To date, these efforts have not been entirely satisfactory because of many variables occurring in beam loading, counterbalance, torque, and polished rod loading. Much electronic equipment will be necessary to complete this job.

Sufficient data has been obtained, however, to permit suggesting a method which will adequately serve in the selection of the prime mover size. This method is more accurate than formulas above when applied to units operating below 12-14 SPM on deep wells where the beam load is high, but the fluid lifted is small, resulting in a large counterbalance and a low average polished rod horsepower.

Generally, it is assumed that if the fluid production (Bbls/da) and depth of pump known, the average polished rod horsepower is then:

4. POLISHED ROD HP =

$$\frac{\text{Bbls/da. X Depth}}{\text{85,000}}$$

To more accurately secure the horsepower that will be required of the prime mover, an "R" factor, Fig. 1, must be applied. In order to apply the "R" factor, determine the unit nominal horsepower:

5. UNIT NOMINAL HORSEPOWER at 20 SPM

$$= \frac{\text{Peak Torque (API)}}{\text{4960}}$$

then divide the unit nominal horsepower into the polished rod horsepower thus:

6. PERCENT LOAD =

$$\frac{\text{Polished Rod Horsepower}}{\text{Unit Nominal Horsepower}}$$

Now from Fig. 1, determine the "R" factor as a decimal and establish the net prime mover shaft horsepower:

7. PRIME MOVER SHAFT HP =

$$\frac{\text{Polished Rod HP}}{R}$$

Where the polished rod horsepower is 50 percent or more of the unit nominal horsepower, use .75 as the "R" factor.

In deep well pumping, volumes of fluid are sometimes low, however, a large unit is required to handle the heavy rod string (structural load) and provide the necessary counterbalance at the gear box. The use of the "R" factor will provide a prime mover of sufficient size.

The "R" factor takes into account certain friction horsepower losses from the prime mover sheave to the polished rod. The unit efficiency is generally assumed to be high, however, when you are utilizing a small portion of the horsepower capacity of the unit, in case of deep wells and low polished rod horsepower, percentage wise, the friction loss is large in proportion to the well load, giving an apparent low efficiency. Therefore, the "R" curve is an experience factor based upon ratio of polished rod horsepower to unit horsepower which make allowances for wells of such condition and gives a more accurate prime mover shaft horsepower requirement.

Let's take an example of a well to be pumped from 9500 feet, dead lift, and assume 100 Bbls, fluid production per day. The unit selected is a 640D API size. Then, using formulae 4, we have:

POLISHED ROD HP =

$$\frac{100 \times 9500}{85{,}000} = 11.15 \text{ HP}$$

Determine unit nominal horsepower, using formulae 5:

UNIT NOMINAL HP =

$$\frac{\text{PT.}}{4960} = \frac{640{,}000}{4960} = 129 \text{ HP}$$

Then, divide unit nominal horsepower into polished rod horsepower as in formulae 6, giving

$$\frac{\text{POLISHED ROD HP}}{\text{UNIT NOMINAL HP}} = \frac{11.15}{129}$$

= 8.65 percent

Now from Fig. 1, locate 8.65 percent and read from the curve, the "R" factor of .47. Formulae 7, we then obtain:

PRIME MOVER SHAFT HP=

$$\frac{11.15}{.47} = 23.7 \text{ HP}$$

This is the net horsepower required of the prime mover. Now, let us assume we will use on the above well, a slow speed engine. The size will then be (from formulae 10)

$$\frac{23.7}{.75}$$

= 31.6 HP = Engine HP rating at speed to be operated.

The above formula is, of course, based on the maximum peak torque of the gear reducer, however, it must be remembered that the pumping torque of the gear reducer, however, torque peaks are much larger than the average and in the deep wells with low volumes of fluid will be 300 percent or more of the average torque. The average polished rod horsepower does not indicate these pumping peaks.

This does not mean that the prime mover horsepower must be three times the average horsepower in all cases. In some wells even a larger percentage may be necessary. The size of the prime mover applied depends upon torque, WR2 of the flywheel, type of motor, etc. Certainly, a prime mover with sufficient horsepower to prevent overloading on the peaks must be used.

PRIME MOVER SERVICE FACTORS:

After arriving at required prime mover shaft BHP as given in above formulas, it is necessary to apply service factors to arrive at correct size and rating of prime mover. Many different methods are used to rate prime movers. The cycle variation of a pumping well requires larger service factors than are applied to other types of machinery to prevent overloading and to insure long trouble free service. The following service factors are generally used.

8. ELECTRIC MOTOR:
High Slip: Net Shaft BHP = Name Plate Rating X .75.

Normal Slip: Net Shaft BHP = Name Plating X .56.

9. MULTIPLE CYLINDER FOUR CYCLE ENGINE:
Net Shaft BHP = *Horsepower Rating X .55.

10. SLOW SPEED GAS ENGINES:
*Net Shaft BHP = *Horsepower Rating X .75.

SELECTION OF TYPE PRIME MOVER:

After consideration of required prime mover brake horsepower the type must be dicided. The oil operator has the following types from which to choose:

(1) Electric Motors.
(2) Multiple cylinder four cycle high speed engines.
(3) Slow speed single cylinder four cycle engines.
(4) Slow speed single or twin cylinder two cycle gas engines.

To make a proper selection requires a knowledge of:

(1) Limits of practical application.
(2) Life expectancy of installation.

*Maximum corrected power unit brake horsepower at the speed to be used, (from manufacturer's performance curve).

(3) Source of power medium.
(4) Type of installation.
(5) Possibility of load change.
(6) Company policy, etc.

A discussion of each type of prime mover is then in order so that the most important factors are considered for each type. These will be discussed in order of types given above.

ELECTRIC MOTORS:

Electrical power has many advantages where available for power requirements up to 15 HP. Where larger than 15 HP electric motors are required, power costs and demand factors may prove uneconomical. Then, internal combustion engines should be used. Where the loads are under 15 HP and electric power is not available, it is wise to check into producing your own electric power with engine generator installations. Experience has shown that engine generator installations of 30 KVA to 200 KVA are quite economical in power costs and labor. They may be operated as individual units, or as units in parallel for greater flexibility. Their salvage value is high.

Most standard oil field electric motors are AC, 3 phase, 60 cycle, 1200 RPM, 220/440 volt. Small motors are often single phase for REA lines. These motors are quite expensive and should not be considered above 7 1/2 horsepower. Sometimes, it is practical to use a converter transformer which has "Y" delta connections to give 3 phase current at the motor. Converters have not been too satisfactory where there is much starting and stopping.

Three phase motors are normally classed as:

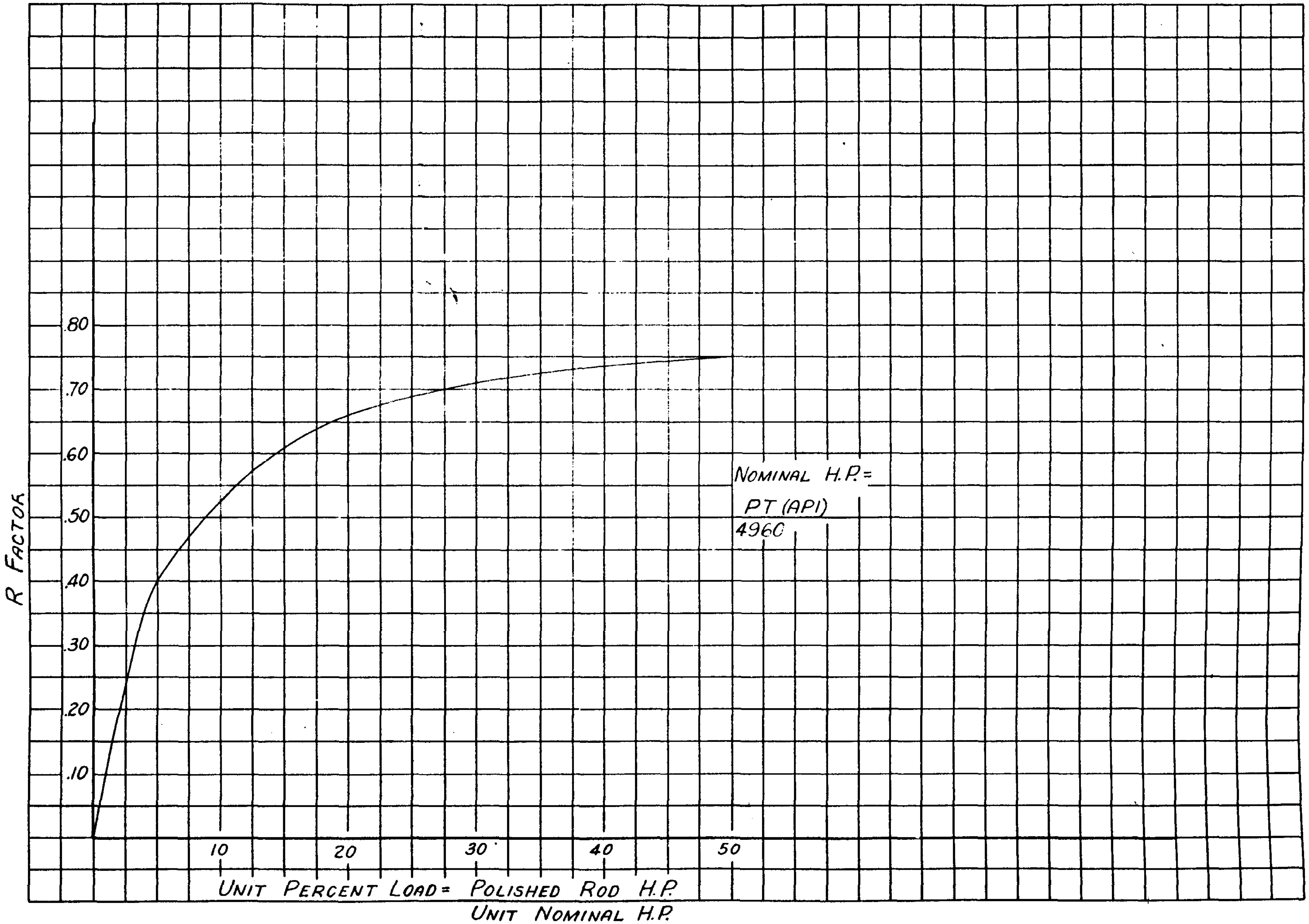
(1) Normal Slip (Type KG or Equal)
(2) High Slip (Type KR or Equal, 5 to 8 percent Slip)
(3) Drip Proof
(4) Splash Proof
(5) Totally Enclosed Fan Cooled (TEFC)

Normal oil field motors are splash or drip proof and type 1. Type 2 is used where cushioning effect is desired and with small isolated engine generator plants. On installations near salt water, it is normal practice to use totally enclosed (TEFC) motors. This type of motor should also be considered for dusty areas where there is a possibility of magnetic particles in the dust.

Electric motors, contrary to common belief, require attention. Proper care must be provided for cleanliness, proper lubrication of bearing and maintenance of control equipment.

The selection of controls are just as important as the selection of the motor. Standard outdoor oil field controls normally include:

(1) Fused line switch, or circuit breaker.
(2) Motor starting magnetic switch.
(3) Low voltage = protecting relay—temperature overload relay. (Be sure heater coils are of proper size to protect motor in case of rod breakage, etc.)
(4) Automatic time clock.
(5) Selector switch (manual or au-



.80
.70
.60
.50
.40
.30
.20
.10
R FACTOR
NOMINAL H.P. =
PT (API)
4960
10
20
30
40
50
UNIT PERCENT LOAD = POLISHED ROD H.P.
UNIT NOMINAL H.P.

FIGURE 1

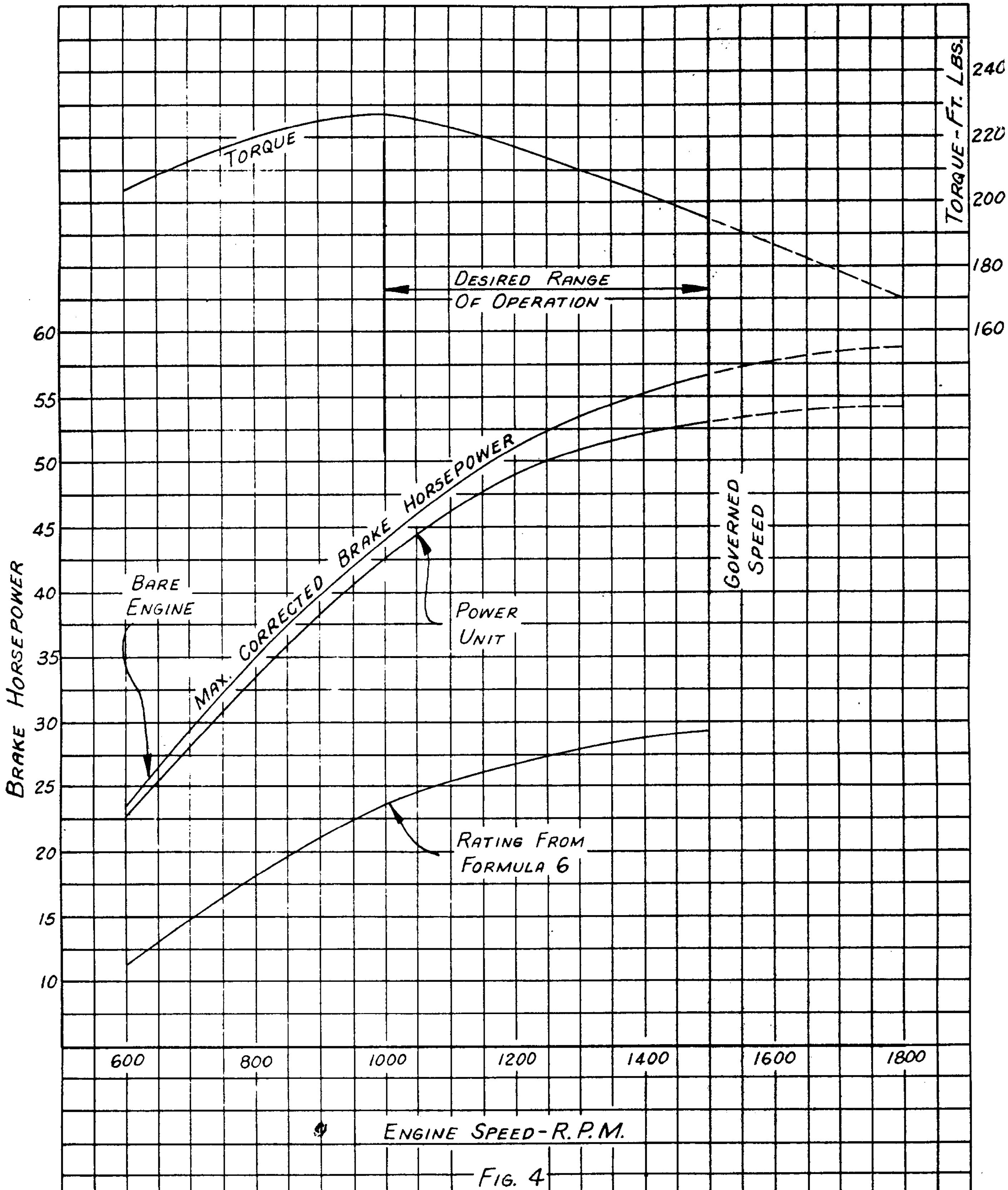


FIG. 4

PERFORMANCE CURVE
TYPICAL 4 CYCLE
4 CYLINDER ENGINE

tomatic starting & stopping)

(6) Lightning protection.

A knowledge of transformer arrangement must be known to obtain proper control and safety. For example, an overload protective device must be provided in each of the 3 phases of motor control circuit rather than on 2 phases as is common. This will prevent so called "single-phasings."

With electric motors, capacitors should be installed at the motor. Long lines and partial loads result in very low power factors. Capacitors of sufficient size should be installed to maintain a power factor above 80 percent where possible.

Electric motors are usually installed on adjustable slide rails for V belt adjustment. Wiring connections must be of approved type with flexible BX cable to the motor to allow movement. Manual control starting and stopping should be quite near the unit brake for safety and convenience of operation.

Internal Combustion Engines

RATING:

Horsepower ratings for all classes of internal combustion prime movers are now standardized somewhat by API. The engine builder must provide performance curves which give the maximum corrected brake horsepower (using power unit, not bare engine ratings) which the engine is capable of producing. No set percentage of this is taken as an API rating as was previously done. The manufacturer may show a rating based upon his experience as to what horsepower the engine will develop continuously and give dependable service. The user may use his own rating or the rating resulting from the application of the service factors shown above. These service factors are time proven and result in economical dependable installations.

Multicylinder four cycle engines must be derated from the maximum corrected horsepower more than slow speed engines because they have much higher speeds resulting in higher piston speeds and inertia forces. Slow speed two cycle engines are built to stand heavy duty continuous service and can be operated nearer the maximum brake horsepower.

Piston speeds in excess of 1000 feet per minute impose high inertia loads. Most multicylinder engines have piston speeds in excess of 1200 feet per minute and high bearing pressures. Slow speed engines have piston speeds of less than 900 feet per minute and large bearing areas with resulting lower bearing pressures, thus longer life.

ENGINE SPEED AND TORQUE:

Each manufacturer of engines designs an engine to operate within a certain speed range and makes recommendations as to normal, minimum and maximum operating speeds. This is shown on the performance curve which the engine manufacturer provides. Fig. 4 is a typical performance curve for multiple cylinder four cycle engines and Fig. 5 is a typical curve for slow speed gas engines. A part of the performance curve is the torque curve which rises, reaches a peak, and then drop off with increase in engine speed. Some slow speed engines develop maximum torque at mimimum speed shown by the manufacturer, therefore, do not show a rising curve. It is desirable to operate any engine at a speed which is on the FALLING side of the torque curve. This is shown as DESIRED RANGE OF OPERATION on Fig. 4 and 5. When so operated, as the load is increased and the speed drops, there is an increase of torque to pull the pumping unit through the pumping peaks. If the engine is operated on the RISING side of the torque curve, when the pumping peaks occur, the engine drops in both speed and torque resulting in shock and inabiliity to carry through peaks with resulting poor performance. It is therefore importont to select sheave sizes which will give required SPM of the pumping unit at the prime mover speed best suited for the particular prive mover.

FLYWHEEL ENGINES:

Why are large open flywheel engines used for oil well pumping?

The purpose of a flywheel is to act as a rotating energy reservoir, absorbing excess energy delivered to the crankshaft by the piston at the time of power stroke and releasing energy when there is no power given out by the piston in order to maintain fluctuation of the engine speed within the desired limits.

To give some idea of the reason why open flywheel gas engines have large flywheels as compared with small flywheels of multiple cylinder engines, let us look at the following table of values required of various types of engines to maintain the same smoothness of rotation. This is known as coefficient of excess energy required.

1 Cylinder 4 Cycle	= 3.00
2 Cylinder 4 Cycle	= 1.80
1 Cylinder 2 Cycle	= 1.25
2 Cylinder 2 Cycle	= .30
4 Cylinder 4 Cycle	= .23

These values are minimum and additional flywheel effect must be added for such pulsating or varying load conditions as those encountered in oilwell pumping to carry through the peaks of the pumping cycle. Same measure of this ability can be obtained by the factor WR2 of the flywheel, where:

W = Weight of Rotating Mass.

R = Means Radius of Gyration of this Mass.

The WR2 factor alone is not a true measure of the flywheel effect as the coefficient of excess energy must be applied to the particular type of engine under consideration. Thus, for a single cylinder four cycle engine, the flywheel must have ten times the flywheel effect of a two cylinder two cycle engine to obtain the same smoothness of operation at the crankshaft.

FUEL SYYSTEM:

Regardless of type of internal combustion engine used, the fuel supply must be considered. Residue gases are best and in the long run, more economical, provided they can be secured. Properly maintained separator gas can be used provided it is dry and has sufficient BTU content. Wet gas causes many engine troubles. Gas direct from casing should not be used until it has gone through a suitable separator.

No engine should be expected to give its best results when operated on sour gas. The two cycle engine will do a much better job on gases which contain sulphur than the four cycle type because the crankcase is sealed from the gases, thus contamination of the lubricating oil is prevented. Also, the cylinders are lubricated by a force feed lubricator which controls the amount of oil applied to the cylinder throughout the life of the engine.

In any case, the gas line should be provided with a scrubber to knock out foreign matter, oil, water, etc., before the gas enters the engine. The scrubber should be of ample size and fitted with drains which are regularly attended. Combination scrubber—volume tanks are good but not always necessary. Some types of fuel systems require a rather large volume of gas at a very low pressure, and require a volume tank at least five times the cubic inch displacement of the engine.

On the engines which use Ensign or equal fuel systems, the volume tank may be omitted provided a regulator of proper pipe and orifice size is placed close to the fuel inlet system. The scrubber is still a necessity, placed so that the gas leaves the scrubber before entering the regulator at the engine.

Where lease gas is available, but not sufficient, a dual fuel installation can be made at the engine. This system is simple and somewhat inexpensive for small amounts of make up gas. The system is so designed that as long as natural gas is available, the engine will operate entirely on it, however, if the pressure drops, the Butane-Propane regulator will then feed enough of this gas to operate the engine. It may be only a small percentage or entirely on butane until the lease gas builds up presssure at which time the butane automatically cuts off.

A similar arrangement is made for four cycle engines where it is desirable to use gasoline instead of butane. This system cannot be applied to two cycle gas engines.

MULTICYLINDER FOUR CYCLE ENGINES:

Since 1945, this type of engine has undergone a number of changes. The most important is that speeds have materially increased. As a result of higher engine speeds, manufacturers have increased horsepower ratings. This has resulted in lighter engines for a given horsepower, higher piston speeds, greater peak firing pressures and bearing pressures. There has also been a tendency toward the use of six cylinder engines with resulting higher speeds. These engines are usually provided with gas-gasoline carburetors so that either fuel may be used. Usually electric starters are furnished as well as some form of safety controls.

The multicylinder engine is built in a wide range of piston displacement and resulting horsepower. The torque developed for displacement is low and most performance curves show the engine actually to have a narrow desired operating speed range, provided they are operated at speeds on the FALLING side of the torque curve as they should be.

The torque developed often times is not sufficient to start the pumping unit without "rocking." Also, the low torque and lack of WR2 in the flywheel accounts for part of the necessity of applying larger service factors to allow a margin of safety and provides enough power to carry the unit through the peak torques of the pumping cycle.

Multicylinder four cycle engines require frequent oil changes or large filtering eqquipment, as the products of combustion find their way into the crankcase. Due to trunk piston design and valve troubles, it is impractical to try to run this type of engine on any fuel containing sulphur, usually termed as sour gas. The result is unsatisfactory service and very short life.

Multicylinder four cycle engines based upon dollars per horsepower give a low initial cost, however, the life of this type engine is short and maintenance is high when applied to oil well pumping. When long life service is desired this type of engine should not be used. Multicylinder four cycle engines are best when used on light wells with part time operation, temporary or testing units or on substructure installations where engine vibration may become a factor.

SLOW SPEED SINGLE CYLINDER FOUR CYCLE GAS ENGINES:

This type of engine with speeds less than 600 RPM, has found wide usage in the oil fields. Because it is simple, it usually can be repaired on the well location and is provided with large flywheel to offset the low torque and provide for inertia to carry through pumping cycles. This engine is manufactured only in smaller sized engines up to 30 horsepower.

This type engine can be operated on either gasoline or natural gas. Starting is normally on gasoline. Starting systems are not usually built into the engine. The heavy flywheels make normal electric starting difficult. Where starters are required, it is usually necessary to resort to a small gasoline driven friction wheel starter or similar type.

Difficulty has been experienced in using the slow speed four cycle engine with sour gases as it is subject to the same crankcase contamination and

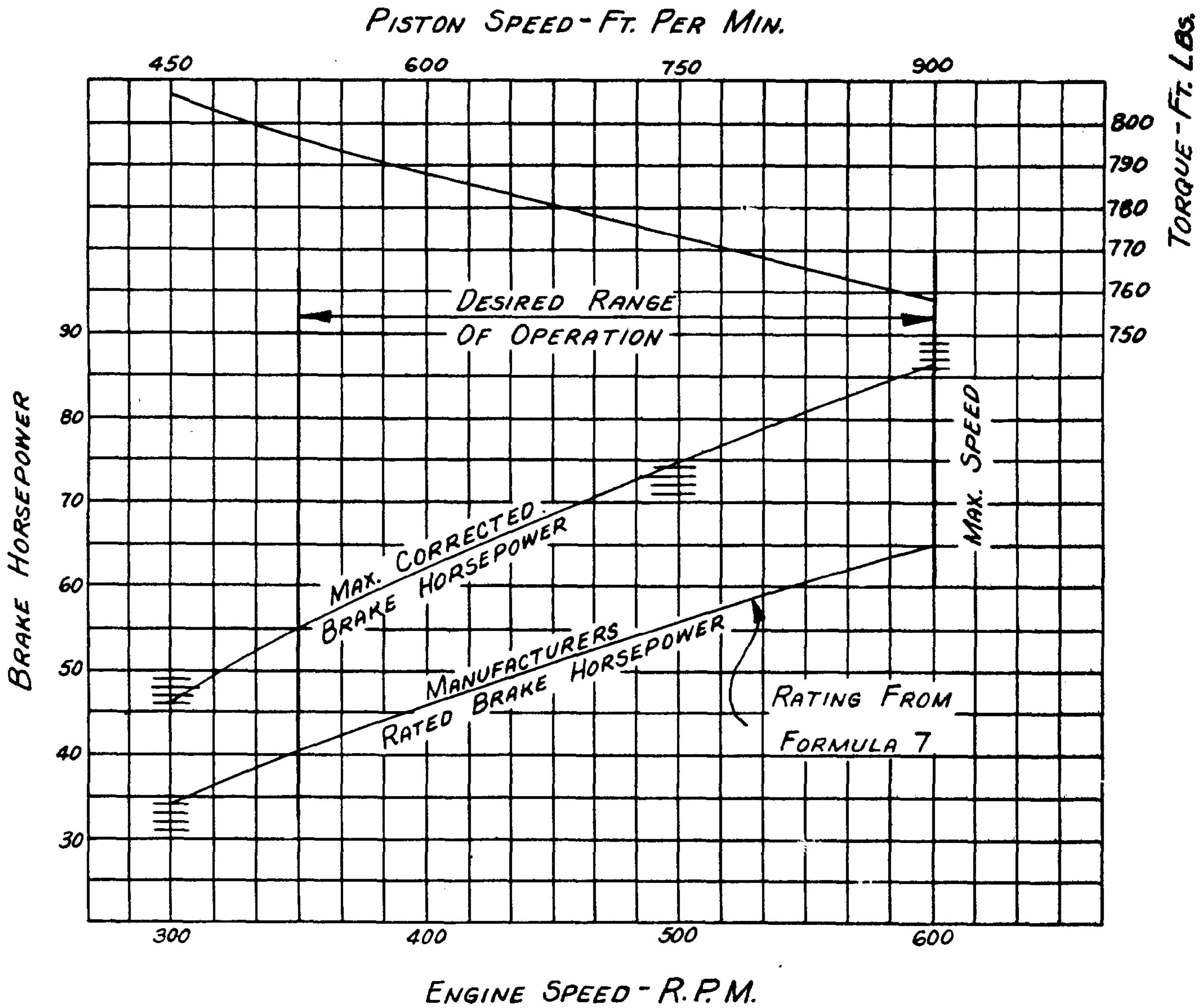


FIG. 5
PERFORMANCE CURVE
TYPICAL SLOW SPEED
TWO CYCLE ENGINE

valve troubles that its cousin, the multicylinder four cycle engine, encounters. This condition on the slow speed engines has been improved by the use of condenser type cooling to maintain higher cylinder temperatures.

Installation of this type engine requires a proper foundation to take the heavy shock loads of the power impulses which occur every other revolution of the crankshaft. The heavy shock loads cause considerable damage to the gear reducer, unit structure and rod equipment. Where the single cylinder four cycle engine is used, in order to reduce this shock, the engine speed should be near the maximum recommended by the manufacturer.

SLOW SPEED TWO CYCLE ENGINES:

This type of engine has become quite popular in the oil fields because of its simplicity, rugged construction, ease of maintenance, medium weight, and long life. The two cycle engine is manufactured as a single cylinder and as a twin cylinder horizontal gas engine, built to heavy duty standards with low piston speeds and large bearing areas resulting in long life.

A crosshead in the base takes the angular thurst of the connecting rod, permitting sealing of the crankcase from the combustion gases, thus preventing many of the difficulties of the four cycle trunk piston engine. The two cycle engine is made for operation only on natural gas or butane and cannot be operated satisfactorily on gasoline. Modern design has reduced the fuel consumption so that it compares favorably with four cycle engines.

The two cycle gas engine is manufactured in a range of sizes from 10 to 100 horsepower. The speed of this type engine is low, usually less than 650 RPM, with corresponding low piston speeds. The torque curve is high, rather flat and with the falling side almost the entire speed range of the engine. (See Fig. 5)

The coefficient of excess energy required in the flywheel is low and with a relatively large flywheel the two cycle single and twin cylinder engine can take the instantaneous peak horsepower of the pumping cycle without requiring large service factors with small flywheel effect. For example, a twin cylinder two cycle engine has two power impulses for each revolution of the crankshaft, as compared with only one impulse every other revolution with a single cylinder four cycle engine. The net result is that the two cylinder two cycle engine is able without excessive flywheel weight to pull the pumping unit through the peaks. The power output of the two cycle engine is steady reducing the shock on the unit gear, structure, and down well equipment.

Starting of two cycle engines is normally on natural gas. The large flywheel two cycle engine can be started by air or a friction wheel starter. Some of the medium and small size engines are made with built-in electric or gas motor starters which have proven very convenient and practical.

Installation of the two cycle engine does not present any particular problem with a properly installed pumping unit. The exhaust pipe size and length are very important for good performance and must be installed according to the manufacturers recommendation. This type of engine is best installed on a tailored base, made for the particular engine. This makes a lower mounted, easily serviced installation. Some two cycle engines are built with the engine base deep enough to clear the flywheel and allow the engine to be installed on cross slide rails. In installations of this type, only heavy cast iron cross rails should be used.

SUMMARY:

Let us summarize the selection and application of the prime mover for oil well pumping. It must first be considered that the calculated polished rod horsepower cannot necessarily be used to determine the prime mover size. The pumping peaks vary considerable with the depth and size of pumping installation. The polished rod horsepower must be multiplied by a factor which will result in large enough prime mover to handle the starting torque and the pumping peaks without overloading the prime mover. Remember that the pumping cycle peaks can be in excess of 300 percent of the average horsepower required.

Sufficient torque to properly cover the starting and pumping peak torques will be found in suitable sizes of high slip electric motors and slow speed gas engines with their high torque and large WR2 of the flywheels.

Service factors must be applied to all types of prime movers to allow an ample reserve of power to prevent heating and overloading. These service factors are products of experience with all types and sizes of prime movers.

Selection of equipment depends upon many factors: No one type of prime mover will fit all conditions and locations. Proper installations result in economy of operation and trouble free operation. The best type of prime mover for permanent installations is either electric motor or open flywheel two cycle gas engines which develop a smooth torque curve without shock.