THE BEAM PUMPING "DESIGN CHAIN"

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ABSTRACT

There are six basic API Loads critical to beam pumping. These loads interact during a complete pumping cycle. Fortunately these loads can be measured and compared against their respective counterparts. A building block approach which combines these basic loads aids in diagnosing operating and design problems.

Standing and traveling valve actions tend to follow a fairly definite pattern during a normal pumping cycle. When abnormal pumping conditions occur, these valves may not conform to their respective normal patterns of opening and closing. A dynamometer is a useful instrument which can be used to record specific loads generated during the pumping cycle. Certain pumping equipment malfunction can be diagnosed by interpreting dynamometer cards. Normal and abnormal valve action can also be detected using the principles associated with dynamometer card interpretation.

There are many basic components which must be considered when designing or operating beam pumping equipment. These must be in harmony with each other to have a relatively trouble free pumping operation. When one of these components malfunctions or is changed, it may be necessary to change other parts of the system to maintain the desired relationship.

Certain operating parameters can be established to promote optimum operating conditions. The use of dimensionless speeds, dimensionless loads and acceleration factors aids in determining the most trouble free portion of the beam pumping "operating window". It is equally important to operate the equipment with the proper counterbalance. Overpumping a well can aggravate an otherwise properly counterbalanced operation. In these situations it is necessary to bring the well outflow into a satisfactory relationship with the well inflow. Redesigning the producing equipment is one way to accomplish that objective. When changing the producing equipment might not be the practical thing to do, intermitting the pumping cycle is another often used and satisfactory method.

THE SIX BASIC BEAM PUMPING LOADS

Six basic loads are critical to the beam pumping cycle: zero load, standing valve load, traveling valve load, peak polished rod load, minimum polished rod load, and the optimum counterbalance effect load at 90° crank angle at the polished rod. Four of these loads, zero, standing valve, traveling valve and counterbalance effect at 90°, can be measured under static conditions using a dynamometer. The peak and minimum polished rod loads are measured under dynamic conditions with a dynamometer. Each of these six loads can be precalculated using API RP11L, "Recommended Practice for Design Calculations for Sucker Rod Pumping Systems (Conventional Units)." The measured loads can be compared with their corresponding precalculated counterparts to diagnose operating and design problems. Figure 1 shows the relative positions of these six loads on the building block diagram. The vertical axis on the building block diagram is linear and represents pounds of load. The horizontal axis is also linear and reflects the relative position of the polished rod during the pumping cycle. Other nomenclature and definitions are included at the end of the paper.

Zero Load, 0

The zero load is the starting point. Not all dynamometers record the zero load on each card. A reference line (R) can be used to determine the position of the zero load on cards on which the zero line is not recorded. Even though it has no height, the zero load is considered to be the first building block. This beginning point is extremely important when determining and calculating the positions of the other critical loads. Figure 2 shows the location of the zero line with respect to the dynamometer card window.

Standing Valve Load, SV

The standing valve load is a misnomer. It does not reflect the load on the standing valve but rather the effects of a good or bad standing valve, or the condition of a pump barrel having a top holddown in case the barrel is split. The actual load on the standing valve test should be the weight of the sucker rods suspended in well fluid. It is one of the two most important loads in dynamometer card interpretation. The standing valve load as reflected on Figure 3 is:

$$SV = Zero + W - (W \times 0.128 \times G)$$

The second building block, W, is the weight of the entire sucker rod string suspended in air. The third building block, W x 0.128 x G, reflects the buoyancy of the well fluid on the sucker rod string.

Traveling Valve Load, TV

The traveling valve load is the other of the two most important loads used in dynamometer card interpretation. It is useful in indicating the condition of the traveling valve and the pump barrel and plunger. The traveling valve load is shown as a combination of four building blocks on Figure 4.

$$TV = W - (W \times 0.128 \times G) + (0.340 \times G \times D^2 \times H)$$

The fourth building block, $0.340 \times G \times D^2 \times H$, is the fluid load on the gross plunger area and is called F_0.

Peak Polished Rod Load, PPRL

The peak polished rod load is the maximum load experienced during the pumping cycle, regardless of whether it occurs on the upstroke or downstroke. The previously mentioned four building blocks plus the maximum dynamic effects on the upstroke are combined to give the peak polished rod load.

$$PPRL = W - (W \times 0.128 \times G) + F_1$$

F_1 is considered to be the fifth building block. It replaces the F_0 building block in locating the peak polished rod load since F_1 contains F_0. The peak polished rod load is shown on Figure 5.

Minimum Polished Rod Load, MPRL

The minimum polished rod load is the minimum load experienced during the pumping cycle, regardless of whether it occurs on the downstroke or upstroke. It represents the buoyed weight of the sucker rod string minus the maximum dynamic effects on the downstroke.

$$MPRL = W - (W \times 0.128 \times G) - F_2$$

F_2, the maximum dynamic effects on the downstroke, is the sixth building block. The location of the minimum polished rod load on the building block diagram is depicted on Figure 6.

Counterbalance Effect Load, CBE

The counterbalance effect is measured under static conditions for convenience but is applied under dynamic conditions. The API RP11L counterbalance effect formula is:

$$CBE = 1.06\ (W_{rf} + 1/2\ F_0)$$

The location of the counterbalance effect on the building block diagram is shown on Figure 7. There is probably as much overall profit to be made by maintaining the proper counterbalance effect on beam pumping wells as on any other item with the possible exception of correctly sizing subsurface pumps.

Composite of the Six Basic Loads

Figure 8 is a composite building block diagram on which are located the six basic beam pumping loads, the associated building blocks, and the derivation of the formula for each load. A dynamometer card has been constructed on this exhibit to show its relationship to the six basic loads.

VALVE ACTION DURING THE PUMPING CYCLE

There are two main valve components in the subsurface rod pump, the standing valve and the traveling valve. The standing valve, as depicted in Figure 9, is the pump intake valve and is normally considered to be a stationary valve. Normally it is a single ball and seat. It acts as a check valve to retain well fluid in the pump barrel on the downstroke portion of the bottomhole pump cycle.

The traveling valve is the pump discharge valve. As shown in Figure 10, it is attached to the plunger of a stationary barrel pump and to the barrel of a traveling barrel pump. Normally it is a single ball and seat. It acts as a check valve to retain well fluid in the tubing on the upstroke portion of the bottomhole pump cycle.

The actions of the standing and traveling valves tend to follow a fairly definite pattern during a normal pumping cycle. This results in the opening and closing of each valve at the appropriate time in the pumping cycle. When abnormal pumping conditions are encountered, valve action may not conform to the normal patterns of opening and closing. The valves then lose their harmonious relationship.

Valve Action on the Downstroke

Valve action would be relatively simple to explain if the valves were located at the surface. They are removed from the surface by a sucker rod string which stretches and recoils. In addition, the independent actions of both valves are seen at the surface on a dynamometer card on which there are also other influences. The description that follows will be based on viewing valve action as it is seen by a dynamometer installed on the polished rod.

Figure 11 shows a trace of the pumping cycle on a dynamometer card. The position of the polished rod at the end of the upstroke is indicated by a large solid dot. A clockwise trace of this dot will indicate the position of the polished rod during the pumping cycle. At the beginning of the downstroke during a normal pumping cycle, the traveling valve is seated and the standing valve is open. The polished rod starts down, but the pump plunger is still traveling up. The speed of the transmission of energy through sucker rods is 16,300 ft/sec. This means that the dynamometer still sees the weight of the fluid in the tubing on the traveling valve and will continue to do so until after the pump plunger is at the top of its stroke and that message is received by the dynamometer. Figure 12 reflects the pressure conditions at the pump at that time. As the pump plunger reaches the end of the upstroke, the polished rod is accelerating on the downstroke. The pump plunger then starts the downstroke and accelerates. The standing valve, which has been open, seats. The traveling valve is still closed. If and when sufficient pressure builds up between the standing and traveling valves to exceed the pressure on top of the traveling valve, the traveling valve opens. The fluid load is transferred from on top of the traveling valve to the standing valve. This can be seen on Figure 13. The pressure relationship at the pump at that time is shown on Figure 14. Normally, this fluid transfer is not instantaneous, but it is fairly rapid.

The traveling valve can open on the downstroke at any polished rod position at which the standing valve is seated, and the pressure between the standing and traveling valves exceeds the pressure above the traveling valve. In the lower part of the downstroke, the fluid load may not be completely transferred from on top of the traveling valve to the standing valve. This is shown on Figure 15 which is supposedly recording the standing valve load, which it is not. In some instances the traveling valve may not open at all on the downstroke. When that happens, the well is usually "pumped off", "gas locked", or has a SV problem.

Valve Action On The Upstroke

Please refer to Figure 16 which shows the polished rod at the end of its downstroke. At that time in a normal pumping cycle, the standing valve is seated, and the traveling valve is open. When the polished rod starts traveling up, the pump plunger is still traveling down due to the time delay caused by the speed of energy transmission through the sucker rod string. As the pump plunger reaches the end of the downstroke, the polished rod is accelerating upward. The pump plunger then starts the upstroke and begins to accelerate. Until the traveling valve seats, the fluid load in the tubing is still on the standing valve. After the traveling valve closes, the pressure between the standing and traveling valves must become less than the pressure beneath the standing valve before the standing valve will open. This pressure relationship at the pump is shown on Figure 17. Figure 18 shows the pressure relationship after the standing valve opens. If the standing valve does not open, the pump will not load, and the normal pumping cycle will be interrupted.

Figure 19 presents the entire sequence of events in the pumping cycle as recorded at the surface by a dynamometer.

COMPONENTS OF THE BEAM PUMPING "DESIGN CHAIN"

The component parts of a beam pumping system are somewhat analogous to a chain. Like the chain, the beam pumping system is no stronger than its weakest part. Unlike a chain, if a part of the beam pumping system is strengthened, it may result in a weak link elsewhere in the system -- one that might not have been weak before. This is why it is essential to balance the various components of the system during the initial design and later by operating practices. There are thirteen major parts of a beam pumping system to consider. These will be discussed one-by-one.

The Producing Reservoir

The main considerations are the kinds and volumes of reservoir fluids, the pressures of the reservoir fluids both in the reservoir and at the pump intake, and the effects that the reservoir fluids will produce as they pass through the producing system. Reservoir fluid inflow and produced fluid outflow should be matched. Undesirable effects result when the producing equipment capacity is not in balance with reservoir fluid inflow. Some of these effects can be loss of production, excessive producing costs, premature equipment failure, and inefficient use of energy.

Bottomhole pressure measuring equipment makes it possible to calculate reservoir and tubing intake pressures within the desired range of accuracy. Through the courtesy of Shell Oil Company, the release of a previously unpublished work by W. E. Gilbert makes it possible to calculate annulus pressures much more accurately than before when gas is bubbling through a liquid column. Accurate and stabilized reservoir and producing bottomhole pressures are essential to the determination of present or future well producing capacities.

As a result of the work of J. V. Vogel, the Inflow Performance Relationship technique (IPR) was simplified so that it can be easily applied in the field. Producing rates can be estimated within the desired range of accuracy using the IPR technique. All that is needed is a shut-in reservoir pressure, a stabilized producing intake pressure, and a stabilized producing rate at the stabilized producing intake pressure.

Downhole Gas Separation

The optimum well for beam pumping will have a rathole below the casing perforations. This will enable the designer to utilize a natural gas anchor. The gas-liquid separation capacity of a natural gas anchor is greater than the capacity of any other gas-liquid, gravity type, subsurface separation device.

If a well produces sand, or if the producing zone is open hole that could cave in and stick the tubing, setting the tubing below the pay zone will probably not be practical.

By definition a natural gas anchor is a casing-tubing-pump intake arrangement that causes liquid plus entrained gas to flow downward in the casing-tubing annulus to reach the pump intake piping. If the downward liquid flow velocity is low enough, most of the gas will counterflow upward, and very little gas will enter the pump intake piping with the liquid. If the liquid contains more than 20 to 40 percent water, the maximum average downward velocity can be 0.5 ft/sec. If the liquid to be pumped is predominantly oil, the maximum allowable velocity, in feet per second, will be approximately equal to 0.5 divided by the viscosity of the liquid in centipoise.

The pump intake piping utilized with a natural gas anchor should be designed to cause as little friction pressure drop as is pract cal, because a pressure drop will cause more gas to evolve from the liquid. The slotted, orange-peeled closed, tubing nipple, screwed into the bottom seating nipple collar, should be long enough to allow a slotted, orange-peeled closed, strainer nipple to be run on the bottom of the pump. It is recommended that the area of the tubing nipple (mud anchor) slots equal or exceed four times the annulus area between the tubing internal diameter (I.D.) and the strainer nipple outside diameter (O.D.). It is recommended that the area of the slots in the strainer nipple equal four times the I.D. area of the strainer nipple. It is further recommended that the strainer nipple be sized to screw into part B22 or S16, API Spec 11AX, assuming that a cup type seating nipple is used. If a mechanical, bottom lock seating nipple is specified, the strainer nipple should be sized to screw into part N12, API Spec 11AX. In other words, an undersized, or oversized strainer nipple is not recommended.

It is normally recommended that a natural gas anchor pump intake be at least 15 ft. below the lowest active casing perforation to insure that the intake is out of the turbulent zone.

Many well fluids contain chemicals that cause foam to be generated in the well bore. It has been reported that natural gas anchors increase the volumetric efficiency of reciprocating pumps in foamy wells.

It has also been reported that a gas producing zone below a liquid producing zone and above the pump intake could decrease the efficiency of a natural gas anchor.

If it is impractical to set the pump intake below the producing zone, and if the liquid does contain enough gas to adversely affect a positive displacement pump's efficiency, a gas anchor should be considered. Industry has come up with various designs, including packer type gas anchors, devices that utilize little cups and rubber tubes, glass ball filled devices, etc. and several versions of poor boy gas anchors. We normally cannot recommend the packer type gas anchor, even though its capacity is exceeded only by the natural gas anchor. This is because we fear that sediment, scale and/or sand will stick the packer, and because the packer type gas anchor will hold more back pressure on the producing formation than a well designed, modified poor boy gas anchor.

Adherence to four basic design rules will allow one to design efficient and effective poor boy gas anchors. First, let us agree on the part names. We call the outer tube the mud anchor, and the tube that normally screws into the bottom of an insert rod pump the dip tube.

Rule Number One. The average velocity of the liquid in the down passage should not exceed 0.5 ft/sec, assuming a viscosity of about one centipoise, or 0.5 ft/sec divided by the viscosity if the fluid is more viscous than water. The volume used in determining this velocity is the pump displacement (P.D.), barrels per day. The formula for finding the minimum required area, in square inches, of the annulus between the mud anchor I.D. and dip tube O.D. is 0.00936 times the P.D. BPD divided by the velocity of the fluid in feet per second.

Rule Number Two. The quieting space in the mud anchor-dip tube annulus, the vertical space between the bottom of the mud anchor slots and the top of the dip tube slots, should have a volume equal to, but not less than one and one-half displacements in cubic inches per upstroke, nor greater than two pump displacements. Pump displacement in cubic inches, is equal to S_p, the net plunger travel in inches times the area of plunger in square inches.

Rule Number Three. The area of the mud anchor slots should equal or exceed four times the actual annulus area. Note that the actual annulus area will normally be greater than the minimum required area found when rule number one was applied. Note that slots are recommended instead of holes, because it is believed that they will be more economical.

Rule Number Four. The area of the dip tube slots should equal or exceed four times the I.D. area of the dip tube. Again, undersized or oversized dip tubes cannot be recommended.

It is normally recommended that the O.D. of steel mud anchors be less than the I.D. of the largest overshot than can be run in the well casing if it is believed that it will ever be necessary to recover the mud anchor by washing over. This drastically limits the poor boy mud anchor capacity that can be secured in wells with small casing. Fiber glass reinforced plastic mud anchors that can be drilled up, or steel designs that can be recovered with spears, should be considered when mud anchor O.D. must approach casing drift diameter. A big advantage of fiber glass mud anchors is corrosion resistance. A disadvantage is low strength. Fiber glass anchors with an O.D. greater than tubing collar O.D. should be centralized with two casing drift diameter O.D. centralizers -- one above the seating nipple and the other on the second tubing joint above the seating nipple. This would also appear to be a good practice when steel mud anchors are run, considering the number that are lost due to casing on tubing collar wear.

The mud anchor and dip tube should be orange-peeled closed. Collars with bull plugs cannot be recommended, because it is difficult to pull a bull plug out of a sand or scale deposit. If removing deposits from the bottom of the mud anchor is an anticipated or known problem, female threads should be cut in the bottom of the mud anchor, and a male plug utilized. We cannot see a need for the mud anchor being more than two feet longer than the dip tube.

The mud anchor slots should start as close to the seating nipple collar as practical. Do not remove more than 0.5 of the mud anchor circumference when cutting slots. Recommended maximum slot width is 0.5 in. This should enable a welder or machinist to place the required slot area within one and one-half to two ft. of the seating nipple. This will minimize the length of the dip tube. Minimizing the length of the dip tube minimizes the friction pressure drop between the dip tube perforations and the pump standing valve. This minimizes gas breakout and maximizes pump volumetric efficiency.

Subsurface Pumps

Only API Spec 11AX pumps are normally considered. API Spec 11AX, "API Specification for Subsurface Sucker Rod Pumps and Fittings", covers three types of rod pumps (API Type R) and tubing pumps (API Type T). The rod pumps are stationary barrel, bottom anchor, seating assembly location B; stationary barrel, top anchor, seating assembly location A; and traveling barrel, bottom anchor, seating assembly location T. Bottom anchor pumps can be run one and one-half times deeper than top anchor pumps. This is because they have to resist the collapsing pressure of the tubing fluid column, while top anchor pumps must resist tubing fluid column pressure as a burst pressure. Pounding fluid tends to burst a top holddown pump barrel or cause a failure at the first engaged thread.

API pumps are constructed for four tubing sizes: 15-1.9 in. O.D., 20-2 3/8 in. O.D.; 25-2 7/8 in. O.D. and 30-3.5 in. O.D. Seven pump bores from 125-1 1/4 in. through 275-2 3/4 in. are available. Sizes increase by 1/4 in. A 178-1 25/32 in. bore is available for a heavy wall barrel, soft packed plunger tubing pump. Larger and smaller pump sizes are available and should be considered, especially the smaller sizes. Rod pumps are available with heavy (API Type H) and thin (API Type W) wall barrels. Heavy wall barrels can withstand more pressure than thin wall barrels and can therefore be run deeper, assuming they are made from the same material. Heavy wall barrels utilize larger I.D. extension nipples above and below the barrel. Calculations indicate the extension nipples can be weaker than the barrel and can be the weak link in deep wells. Heavy wall barrel pumps can be operated as stroke through pumps. This can be an advantage when the well fluid precipitates scale on the barrel of a thin wall pump and would stick the plunger in the barrel. A disadvantage of a heavy wall barrel pump with extension nipples, and of all tubing pumps which have heavy wall barrels and extension nipples, is an increase in clearance and a decrease in compression ratio. A low compression ratio increases the tendency of a reciprocating pump to gas lock when pumping a gassy fluid.

An old rule of thumb states that pump plungers should be three ft. long in wells less than 3,000 ft. deep, three ft. plus one ft. in length per 1,000 ft. between 3,000 ft. and 6,000 ft., and six ft. long in 6,000 ft. and deeper wells. Shorter plungers can and should be used when pumping very viscous oil, but shorter lengths than indicated by the rule of thumb cannot be recommended when pumping low viscosity fluids. Slippage (leakage) between the plunger and barrel is inversely proportional to plunger length, and proportional to clearance in thousands of an inch cubed. For example, if the plunger length is cut in half, the slippage will double, but if the clearance is increased from one to two thousands of an inch, the slippage will be increased by a factor of eight.

Allowable pump setting depths should be secured from the pump supplier. Example allowable setting depths from one supplier for 1-1/2 in. pumps are 6,197 ft. for a top holddown, RWA and RHA pumps, and 9,292 ft. for bottom holddown RWB and RHB pumps. This manufacturer states that bottom holddown, traveling barrel, RWT and RHT pumps can be run as deep as RWB pumps. We think the length of the pull tube should be considered when determining the allowable setting depth of traveling barrel pumps. A study made by another supplier indicates that if the pull tube is longer than seven ft. in a 1-1/2 in., traveling barrel pump set at 9,292 ft., you can expect the pull tube to fail at the point where it screws into the seating assembly. It is concluded that long stroke traveling barrel pumps cannot be run as deep as bottom holddown stationary barrel pumps.

The rod string should push straight down on the plunger during the downstroke. Therefore, at least one full-bore, fluted, rod centralizer should be run above the pump handling pony rod.

Pumps for new installations should be sized by computer programs that consider the numerous variables. A study of computer printouts which considered the variables that we considered several years ago resulted in Table 1. Note that 106-1 1/16 in. pumps are recommended, because this was an API pump size when the study was made. This table is recommended as a starting place when sizing API pumps in the field.

Tubing

Normally, external upset API tubing should be used in beam pumping wells, because there will be rod coupling-on-tubing wear, and early tubing failures can be expected if non-upset API tubing is used.

Thread dope must be used on API tubing threads to keep the joints from leaking. Thread dope does not have an infinite life. If collar leaks begin to appear in old tubing strings, it may be necessary to remove all collars, clean the threads and apply new thread dope.

Torque has been condemned as an acceptable method of making up sucker rod connections, and it has limitations on all threaded joints. Do not solve problems that you do not have, but do consider using a combination of torque and turns on critical API tubing strings.

Pounding fluid can create collar leaks in properly made-up tubing joints, and this is another reason for avoiding fluid pounds.

Tubing anchors within 100 to 200 ft. of the pump seating nipple eliminate tubing stretch from the anchor point to the surface, assuming the anchor is set properly. This should also protect this portion of the tubing string from fluid pound collar leaks and the casing from the anchor to the surface from tubing collar-on-casing wear. Tension tubing anchors are recommended.

Tubing anchors should not be used in wells that have bad casing, and they could cause a problem in wells that produce sand or if scale or salt build-up is a problem. Some tubing anchors contain shear pins that are supposed to shear and allow the tubing to be pulled when the anchor unseating mechanism is fouled. Be sure and remove as many pins as practical before the anchor is run, because it may be necessary to shear the pins while lifting the entire rod string, the fluid in the tubing, and the tubing without yielding the tubing.

If tubing is not anchored, tubing stretch on each pump downstroke will cause the tubing collars to rub against the casing. Evidence is worn collars and casing leaks. It is recommended that the location of the collars relative to the casing be changed each time the tubing is pulled.

Unanchored tubing cork screws from the seating nipple up several hundred feet every time the traveling valve closes and the standing valve opens. This causes rod coupling-on-tubing wear.

The sucker rods above the pump can be buckled on each plunger downstroke if the force required to move the plunger exceeds 21 lb. for 5/8 in. rods, 41 lb. for 3/4 in. rods, etc. This also causes rod coupling-on-tubing wear. This wear causes a large portion of the so called "split tubing" leaks that occur near the pump. Therefore, these tubing joints should be rotated with joints further up the hole on a regular schedule. Further, the force required to push the plunger through the barrel on the downstroke should be supplied with something much stiffer than a 5/8 in. or 3/4 in. sucker rod.

Sinker Bars

We recommend the use of centralized sinker bars above the pump to supply the force needed to open the traveling valve at the start of the downstroke, to push the plunger through the barrel, and to cause the fluid to flow through the traveling valve and plunger into the tubing.

We have evidence that this increases the minimum load, decreases polished rod horsepower, and it should decrease low tubing leaks. We are trying to develop the math that will show the effect of sinker bars on net plunger travel.

We recommend that about 20 percent of the theoretical amount of sinker bars required to open the travel valve be run in wells that are pumping low viscosity liquid. The amount required to push the plunger through the barrel, cause fluid to flow through the traveling valve and plunger into the tubing, and keep the sucker rod string in tension on the downstroke should be run in wells pumping viscous fluid and in wells that utilize unconventional rod strings.

Sucker Rods

The API recognizes three grades of sucker rods at this time -- Grades C, K and D. Grade C is made from carbon steel, has a maximum allowable tensile strength of 115,000 psi, and is, therefore, not susceptible to hydrogen embrittlement. Grade K rods have about the same tensile strength as Grade C rods but contain 1.65 to 2.00 percent nickle, and are therefore, more expensive. A test on a mixed string conducted several years ago indicated that Grade C rods were superior to Grade K rods in corrosive service. Some reports from the field indicate Grade K rods outperform Grade C in some corrosive services. NACE task group T-1F-15 is preparing a supplement to API sucker rod specification 11B and API sucker rod recommended practice 11BR that recommend tests be made in specific areas to determine which is superior.

API Grade D rods have a minimum tensile strength of 115,000 psi and are therefore normally susceptible to hydrogen embrittlement. Therefore, Grade D rods should not normally be used in hydrogen sulfide service. Continuous corrosion inhibition, which we would guess is practically impossible, may allow Grade D rods to escape hydrogen embrittlement failures in hydrogen sulfide service.

Table 4.1, API Spec 11B lists full size couplings for 5/8 in. through 1 1/8 in. sucker rods and recommends minimun tubing sizes. Table 4.2 lists the slimhole couplings that are available for 1/2 in. through one in. sucker rods. Slimhole couplings for 5/8 in. through one in. rods can be run in one size smaller tubing than the respective full sized coupling. This enables operators to run one in. rods in 250-2 7/8 in. O.D. tubing and 7/8 in. rods in 200-2 3/8 in. O.D. tubing, etc. Decreasing the O.D. of the coupling decreases the area available for supporting the pumping loads. How much can the coupling area be reduced relative to the sucker rod area without causing the coupling to become a weak link in the chain? We do not actually know, but we think that a one in. slimhole coupling has about the right cross sectional area. If this is correct, derating factors can be calculated for all the other slimhole couplings. Recommended derating factors are given on Figure 20. Note that the derating factor for 7/8 in. slimhole couplings is 0.75. Again, do not solve problems that you do not have, but if more than 50 percent of your rod string failures are in the connections, and you are running 7/8 in. slimhole couplings, you may find that the majority of the failures are these couplings.

Tapered, 250-2 7/8 in. O.D./200-2 3/8 in. O.D., tubing strings should be considered when 7/8 in. slimhole sucker rod couplings in a straight 200-2 3/8 in. O.D. string are causing excessive pulling jobs.

Torque has been discredited as a sucker rod connection make-up method. When the threads are "properly lubricated", and we do not know what "properly" means, 10 percent of applied torque turns the coupling relative to the pin, and 90 percent of the torque is consumed by friction. Any variation in lubricants or in the surface finish of the threads or mating surfaces could drastically change these percentages. This indicates that torque could never be a precision make-up method for any threaded connection.

Circumferential displacement (C.D.) is now recommended in Section 4 of API RP 11BR, "Care and Handling of Sucker Rods", for making up sucker rod joints. C.D. should also be used for calibrating power tongs. To make up a sucker rod joint using C.D., the pin and coupling threads should be cleaned and lubricated with a lubricant that has passed the NACE screening test. This test states that an acceptable lubricant will allow the lubricated joint to be made up and broken 10 times without galling the threads. Some corrosion inhibitors can pass this test. The clean lubricated joint should then be made up hand tight. A hand tight position, as used in Section 4, is attained when full shoulder abutment is made. The coupling should then be turned the amount specified in Table 4.1, Section 4, API RP 11BR, relative to the pin.

Paragraph 4.8, Section 4, states that new Grades C and K rods should be made up and broken in the field prior to final make-up on initial installation. If full shoulder abutment is not maintained throughout the pumping cycle, lubricant will be pumped out of the joint, and more viscous lubricants should be considered. If the surfaces completely separate, premature failure of the pin and/or coupling will occur, or the joint will unscrew.

Rod strings should be designed to enable the operator to unseat the pump without yielding the bottom section of the rod string. The diameter of the pump plunger determines the fluid load lifted during the pumping cycle, but the I.D. of the seating nipple determines the fluid load that must be lifted to unseat the pump. Friction in the holddown plus sediments in the pump-tubing annulus increase the required pump unseating force.

Old pony rods should not normally be used with new rod strings. A pony rod as long as the polished rod stroke length should be added to the bottom of the smallest rod section each time the rod string is pulled to move the rod coupling-on-tubing wear around. When pony rods added equal one full rod, a pony rod should be removed each time the rods are pulled.

The only couplings covered by API Spec 11B are Class T couplings that have a Rockwell C hardness of 16 minimum and 23 maximum. An API task group is studying hard faced couplings which are available from several manufacturers. Hard faced couplings are favored by some operators over API Class T couplings, because they appear to have more wear resistance. Preliminary results of laboratory tests now being run by an operator to determine the amount of tubing wear that occurs when different brands of hard faced couplings are used with different API grades of tubing, indicate that hard faced couplings drastically increase tubing wear. Hard faced couplings may just move the problem from rod couplings to the tubing. This problem points out the importance of insisting that wells be drilled straight enough to be produced economically.

Section 3, API RP 11BR, recommends a procedure for determing allowable stress on API Grade sucker rods. Manufacturers of non-API rods specify allowable stress on their rods. The allowable stress curve that we use is more conservative than the API curve when the stress ratio, minimum load/peak load, approaches 1.0, and less conservative than API when the stress ratio is less than 0.6.

Tapered strings are used to decrease the stress on the rods above the bottom section. This allows pumps to be run much deeper that would be possible if just one size of rod was run. Tapered rod strings can be operated faster than straight rod strings. Tapered rod strings can weigh less than straight rod strings. This can reduce the required pumping unit gear box size. Rod stretch is proportional to rod string weight. Tapered, light rod strings stretch more than heavy, straight strings when the traveling valve closes on the upstroke.

We think that an API Grade C or D rod string is the correct size when the net plunger travel is 80 percent of the polished rod travel, or greater. If special rods that have an allowable stress greater than Grade D allowable stress are utilized, net plunger travel should equal 70 to 75 percent of polished rod travel. This rule of thumb assumes that the tubing is anchored in deep, high volume wells, that the pumping speed divided by the natural frequency of the rod string is about equal to 0.35, that the fluid load divided by the force that would be required to stretch the rod string one polished rod stroke length is less than about 0.50, and that the acceleration factor is between 0.225 and 0.30 in shallow wells.

At this time, API Spec 11B is not as strong as the API Tubular Goods Specs. This spec. does not insure that the user will receive a quality product at this time. An API task group made up of users and manufacturers is working to improve this document.

Stuffing Boxes and Pumping Tees

Section 13 in Supplement 1 (dated March, 1979) to API Spec 11B covers stuffing boxes and pumping tees. Adherence to this spec. insures that the center line of the stuffing box, pumping tee and well tubing will be aligned within 1.5 in. in 20 ft. and that these parts can be purchased to attach to API Std 5B upset and non-upset tubing.

When a beam pumping unit is set correctly relative to a pumping tee, the polished rod will be in the middle of the pumping tee at the middle of the stroke, with the stuffing box unscrewed and lifted up on the polished rod.

If the stuffing box packing is used to hold the polished rod in the middle of the pumping tee during the stroke, the packing and polished rod will have a very short life.

Polished Rods and Polished Rod Liners

Section 12 of Supplement 1, (dated March, 1979) to API Spec 11B covers polished rods and polished rod liners. Table 12.1 recommends polished rod size versus the size of the top sucker rod. API polished rod lengths are 8, 11, 16 and 22 ft. Upset ends can be furnished on 1 1/8, 1 1/4 and 1 1/2 in. polished rods and are recommended for heavy loads. Upset ends have sucker rods connections which are superior to the pipe thread connections on non-upset polished rods. The surface finish on polished rods is not specified in Section 12, API Spec 11B. It is recommended that a 16 micro-inch, RMS, finish be specified.

Corrosion resistant polished rod liners on carbon steel polished rods are usually more economical than corrosion resistant polished rods. If the polished rod does not travel straight up and down during the pumping cycle, liners may not be practical. A full sucker rod in between the polished rod and any top of the string pony rods will decrease crooked well head induced polished rod failures. The followers in a stuffing box should be changed when out-of-round wear exceed 0.050 in. Polished rods can wear about 1/32 of an inch without drastically decreasing packing life, providing the wear is not corrosion pits.

It is recommended that a coupling and pony rod be screwed on to the top of every polished rod before installation and that this coupling and pony rod remain on the polished rod during operation. Polished rod clamps do slip.

Polished Rod Clamps

Several years ago polished rod clamps were dropped from the API Specs. A task group is now working to reintroduce polished rod clamps in API Spec 11B.

The manufacturers of clamps specify the torque required to tighten their clamps and the forces that will cause clamps to slip on polished rods. This is based on an assumption that the O.D. of the polished rod will be approximately equal to the O.D. the manufacturer assumed when he designed and built the clamp.

Overtightening polished rod clamps may cause the start of a polished rod shear failure.

Polished rods and the inside of the clamp should be cleaned before installation. Do not allow the use of pipe wrenches on polished rod bolt nuts. Replace all pipe wrench cut nuts as soon as practical.

Pumping Units

The scope of API Std 11E, "API Specification for Pumping Units", is to standardize on specific pumping unit structure sizes in combination with established reducer sizes and to standardize on specific reducer sizes. Walking beam design is covered along with formulas for computing the peak torque ratings of gear and chain reducers. Appendix A includes a crank counterbalance rating form and a pumping unit stroke and torque factor form.

Appendixes B, C, and D give the procedure for calculating torque factors for conventional units, Lufkin Mark II type units, and air counterbalance units, and state that the manufacturer is to supply torque factors and polished rod position for each 15^{o} crank position. An API pumping unit is described by listing the peak torque rating in in-lb divided by 1,000, the beam rating in lb. divided by 100, and the maximum stroke length in inches. For example, a unit with a gear box rating of 640,000 in-lb, a 36,500 lb. structure capacity, and a maximum stroke of 144 in. is described by "640-365-144". Manufacturers add letters to further describe their units. For example "C-640D-365-144" indicates a conventional crank counterbalance unit with a 640,000 in-lb, double reduction, gear box, 36,500 lb. structure capacity, and a maximum stroke length of 144 in. This specific unit also has stroke lengths of 124 and 106 in. Books have been written about the advantages and disadvantages of the Appendixes B, C and D type units, and there have been numerous variations of these three basic types of units available on the market. Normally the largest conventional crank counterbalance unit available is a 912. The largest Mark II type unit has a 1,280 gear box. "2,560-470-240" air balance units are available. Small API units, "6.4-32-16" through "57-109-48", are available with beam counterweights. Beam counterweights can be added to conventional crank counterbalanced units. Beam weights may be less expensive than crank weights, but it will take more energy and a larger prime mover to operate the unit at high pumping speeds. Air balanced units may decrease rod failures, but maintaining the air balance system can be a problem. Mark II type units cannot be operated as fast as air balanced units, and air balanced units cannot be operated as fast as conventional units, but this is not normally a problem with either Mark II's or air balanced units. The air balanced and Mark II's use more degrees of crank travel to complete the first one-half of the upstroke. This tends to decrease the peak load. This is an advanatage if rod fatigue is a problem. You do not actually "get something for nothing." The geometry that caused the peak load to decrease also decreases the minimum load, resulting in equal load range for all three, and therefore about the same dynamometer card area and polished rod horsepower. If you could determine the dynamometer card shape that your well would create, you could intelligently select one type of unit over another. If the card will slope up from left to right, a conventional, crank counterbalance, beam pumping unit should be favored. If the card will slope down from left to right, a Mark II type unit should be favored. If the slope approaches zero (flat), an air balanced unit should be first choice.

We used to think that an advantage of some conventional units was that the walking beam could be moved relative to the saddle bearing. Later we figured out that utilizing this feature caused the polished rod to move in an arc. This decreased stuffing box packing life and polished rod life.

API RP 11G, "API Recommended Practice for Installation and Lubrication of Pumping Units", gives excellent guidance in these two areas. Pumping units bases should not move relative to the earth during the pumping cycle, and the unit should not move relative to the base. Maximum allowable vibration amplitude for slow speed equipment (60 to 360 cycles/min) is five mils, peak-to-peak. Maximum allowable amplitude for 601-1,200 cycles/min equipment is three mils, peak-to-peak. If vibration amplitudes exceed these limits, fatigue failures can be anticipated.

Pumping units should be set high enough, relative to the stuffing box, to allow a polished rod liner to be installed on the polished rod, and to allow a dynamometer and an auxiliary hanger to be installed between the carrier bar and the top of the liner with the unit operating at the longest stroke length.

Lubrication of beam pumping units is normally not a problem if up-to-date manufacturer and/or API recommended practices are followed. Manufacturers' recommendations vary slightly and do change with time. Gear box oil samples should be inspected for emulsions, dirt and odor every six months, and water should be drained from the gear box on a regular schedule. If the oil looks bad, it should be changed. Filtering of gear box oil is not recommended. Check to see if the gear box bearings are getting oil if the pumping speed is reduced below 10 SPM. Oil wipers normally wipe oil from the slow speed gear into troughs that serve the gear box bearings. At slow speeds, there may not be sufficient oil on the sides of the slow speed gear. If this occurs, the wipers should be moved to the high speed gear.

The brake should not be used to hold the cranks and weights horizontal with the well load disconnected, because this can drastically overload the gears.

API RP 11ER, "API Recommended Practice for Guarding of Pumping Units", First Edition, March, 1976, was prepared to give guidance on this important subject. Some local codes are more conservative than this RP.

Sheaves and V-Belt Drives

API Spec 1B, "API Specification for Oil-Field V-Belting", Fifth Edition, March, 1978, adequately covers selection, installation and operation of V-belt drives. The designer of beam pumping installations must know V-belt drive basics. New beam pumping units can be purchased with different size sheaves, and sheaves can be purchased to accept different V-belt cross sections. A unit sheave should be selected that will allow as much speed variation, up and down, from the design speed as is practical without violating API Spec 1B rules. Most unit sheaves will have grooves for more belts than are actually needed. Excessive belts increase investment and decrease prime mover bearing life. The number of belts required should be redetermined before replacement belts are ordered. Joined V-belts are not recommended if belt length is less than 285 in. The length of multiple V-belts should not vary more than is indicated in Column 8, Table 2.2, API Spec 1B. The tension in any V-belt drive should be checked on a regular schedule, using the Section 6, API Spec 1B, RP as a guide.

Prime Movers

A 1,200 RPM, NEMA Design D, five to eight percent slip, electric motor is the normally recommended prime mover for beam pumping units. Motor size will usually be about two times polished rod horsepower, assuming the pumping unit is heavily loaded, pump volumetric efficiency is high, etc. Our present sizing procedures usually result in oversized prime movers. It is therefore recommended that existing beam pumping prime mover loads be determined in each beam pumping area and compared with the prime mover sizing formulae in use. Oversized electric motors should be fused to protect the installation, not just the motor.

Grounding of electrical equipment must not be neglected. Bare, solid copper, No. 4 minimum size, ground wires should be run from all electrical equipment to the well casing. The resistance between any piece of equipment and the casing should not exceed one ohm. The ground wire from an electric motor should be connected to the motor frame, not to a foundation bolt. In dry country effective grounding systems must be attached to a ground rod that penetrates permanent moisture, and well casings fit this definition. In addition, paragraph 250-81a of the 1978 National Electrical Code states that an underground water pipe ground shall be supplemented by an additional electrode. A statement on Page 81 of "Electrified Oil Production" by Hogwood and Howell indicates that supplemental electrodes should be located directly under arresters. The above indicates we should consider double grounding electrical installations.

Single and multi-cylinder engines are still used as prime movers on isolated leases and for pumping large volume wells, etc. Single cylinder, two cycle engines can cause inadequate foundations to fail, and pumping unit structure fatigue failures. The torque curve of a gas engine should be studied before the allowable beam pumping unit operated speed range is determined. High prime mover mounts are ideal for electric motor prime movers but can be impractical for multi-cylinder engines.

Flow Lines

Piping that introduces the gas that flows up the casing-tubing annulus into the flow line, and the flow line from the pumping tee to the battery may not normally be thought of as part of the beam pumping installation, but it is important.

The check valve in the line from the annulus to the flow line has to work properly. If it leaks, liquid production can circulate back into the annulus.

If the flow line is too small, it will cause back pressure to be placed on the producing formation and decrease production. If the flow line is too large, what goes into the flow line at the well will not be what comes out at the battery or test facility. Short well tests will be meaningless.

We recommend that conventional swing check valves be sized to have about 0.5 psi pressure drop across the valve with design flow.

We recommend that flow lines be designed so that velocity, in feet per second, is between 16 divided by the square root of the fluid density, in pounds per cubic foot, and 50 divided by the square root of the density.

Flow lines should be monitored for leaks on a regular schedule. Steel flow lines should normally be buried below the frost line, and buried steel lines should be externally coated and cathodically protected.

The casing-tubing annulus should be equipped with a surface valve that will allow the casing pressure and the fluid level to be monitored. This valve can also be used to introduce defoamers, corrosion inhibitors, hot oil, etc. It should be bull plugged closed when not in use. Introducing liquids into the annulus at a higher rate than the annulus self-venting rate, drives the producing liquid level below the pump intake, starves the pump, and causes premature system failures.

Back pressure valves in the flow line upstream from the casing annulus gas piping tie-in can be used to keep the tubing from unloading. Optimum back pressure would be equal to the pump intake pressure. Back pressure on the tubing can cause paraffin deposits in the tubing to come loose, flow up the tubing, and block the back pressure valve, causing the stuffing box packing to blow out. Be very sure the tubing and rods are clean before applying back pressure.

OPTIMUM OPERATING CRITERIA

Several parameters must be considered when establishing optimum operating conditions. The major of these are dimensionless pumping speeds, dimensionless pumping loads and the associated acceleration factors. When used together, the most trouble free portion of the beam pumping "operating window" can be determined. Although there are others, two main pieces of data are used to identify desirable operating conditions. First, and most important is the well test. The next, and a valuable diagnostic indicator, is a dynamometer card.

Well Tests

Stabilized and representative well tests are essential to analyzing beam pumping equipment performance. The frequency of such tests is an individual determination based on the type of well. Tests should be of sufficient frequency to be usable with the dynamometer in analyzing problem wells. A normal test and then one reflecting the fluid volume at the time dynamometer cards are taken are considered the minimum to be able to use dynamometer cards effectively.

Dynamometer Cards

A suite of dynamometer cards developed from API RP 11L2 is included as Figure 21. Although there is no one "typical" dynamometer card, the cards on this figure are typical for the N/N_0' vs. F_0/Sk_r relationship which each represents. A brief discussion of dynamometer card basics is included so the reader can relate to the changes in shapes on the figure.

A dynamometer card is a trace of well load in pounds versus polished rod position. A dynamometer card can also be a trace of well load in pounds versus time. Both traces are made with basically the same objective -- to obtain a graphic representation of the forces acting on the polished rod during a pumping cycle. The major difficulty normally encountered in interpreting dynamometer cards is that the trace represents a composite of forces acting throughout the sucker rod system but which are measured and recorded at the polished rod. It is comparable to listening in on a multi-telephone party line and picking out significant bits of information. There are times when these "party line disturbances" are such that even the most skilled dynamometer card interpreter finds it virtually impossible to determine what is happening. However, by applying some basics, it is possible to narrow the determination to probabilities.

As shown on Figure 22, the dynamometer card has a linear vertical load axis, and a linear horizontal position of polished rod axis. The scale of the vertical axis is pounds per inch. The scale of the horizontal axis is inches per inch, or in one particular case, the horizontal axis can also be expressed in terms of time. The interpreter should determine which horizontal scale is applicable to the particular dynamometer card.

A "healthy" dynamometer card should be taken immediately after changing beam pumping equipment, such as installing a new pump. It must be representative of a situation in which all parts of the system are in good condition and are operating properly. This card is used for comparative purposes when later cards are taken -- as an indicator that certain conditions are either normal or abnormal. It should be compared to the appropriate card on the suite of cards on Figure 21. After establishing that it is a good representative card, it then becomes the reference card against which future dynamometer cards from the same well can be compared as long as the equipment and the operating conditions are not changed. However, it is only one piece of data. All pertinent data should be considered when analyzing pumping well performance. When coupled with other data, a dynamometer card can serve as a valuable indicator to help identify and pinpoint possible causes of problems.

Excessive Overtravel and Undertravel

Excessive overtravel can be recognized on a dynamometer card when the general axis through the card has an abnormally high left to right downward slope through the card. An example of this is the card on Figure 21 having an N/N_o' of 0.40 and an F_o/Sk_r of 0.1. It is difficult to counterbalance a well which generates such a card because of the difficulty of keeping the counterbalance effect on the inside of the trace of the dynamometer card during the middle one-half of the up and down strokes. Overtravel is more pronounced with higher N/N_o' - lower F_o/Sk_r relationships than those in the middle of the range. For that reason, an upper limit of 0.35 is recommended for N/N_o'.

Excessive undertravel can be recognized on a dynamometer card when the general axis through the card has an abnormally high left to right upward slope through the card. The card on Figure 21 having an N/N_o' of 0.10 and an F_o/Sk_r of 0.60 is an extreme example of such a situation. It is difficult to counterbalance a well having such a card due to the effective counterbalance problem. Undertravel is more pronounced in situations having high F_o/Sk_r values. For that reason, an upper limit of 0.50 is recommended for F_o/Sk_r.

Acceleration Factor

When considered with dimensionless pumping speeds and loads, acceleration factors are used to keep from over or undersizing pumping equipment. The reason for using acceleration factors as one of the design and operating parameters is that N/N_0 does not consider or include the stroke length. The formula for an acceleration factor is:

$$c = (S \times N^2)/70{,}500$$

One of the things to consider in designing and operating beam pumping equipment is not to exceed the free fall speed of the sucker rod string. In 1962, W. H. Ritterbusch, Jr. authored a chapter in "Petroleum Production Handbook", published by McGraw-Hill Book Company. In that chapter, he stated, "Always choose a speed below that maximum practical limit permitted by free-rod fall so that the polished-rod clamp and hangar bar will not separate on the downstroke." He included a permissible speed and stroke length curve based on 70 percent of a maximum free fall limit. In 1965, Bethlehem Steel Company published a brochure, "Pumping Unit Selection Charts", in which was stated, "Normally at speeds which exceed 0.7 of the free fall velocity, the polished rod begins to leave the carrier." Lufkin Industries in its 1980-81 catalog also supports 0.7 of the free fall velocity as being the maximum. If the well fluid is fresh water and the well bore is straight, we do not have any reason to question the limit cited in the literature. An acceleration factor of 0.417 can be calculated for that limit. However, we do not support that acceleration factor as the practical upper limit because of the possibility of excessive equipment failure. Most of the time well fluids differ in character from fresh water. Well bores are seldom straight. Even though there are other factors, just considering those two are normally sufficient to discourage setting the maximum acceleration factor at 0.417. Table 2 presents a comparison of the strokes per minute versus stroke length for acceleration factors of 0.417 and 0.3.

For design purposes, we recommend that the acceleration factor be kept between 0.225 and 0.3 in shallow wells. In the event 0.3 conflicts with an N/N_0' of 0.35, N/N_0' dominates.

For operating purposes, the acceleration factor should be at least 0.225 to be sure the equipment is not overdesigned, and not more than 0.3 to provide a safety factor so as not to approach the free fall speed of the sucker rod string. In real world operating situations the free fall speed of the rods and the gear box capacity determine the maximum pumping speed.

A decrease in the well inflow, or any other event that causes the pump displacement to exceed well inflow, may necessitate decreasing the pump displacement. If the well capacity is equal to or less than six-tenths of the pump displacement, changing the pumping speed, pump diameter, and/or polished rod stroke length, usually in that order, are means of decreasing the pump displacement. However, if the well capacity is equal to or greater than six-tenths of the pump displacement, do not consider changing any of those three. In that case, pump displacement can be controlled by time clocking if the prime mover is an electric motor. If time clocking is used, an effort should be made to confine any fluid pound to the first one-quarter of the downstroke.

Figure 23 is a nomograph which makes it possible to determine the maximum pumping speed and the maximum polished rod stroke length for both tapered and nontapered sucker rod strings. The example problem on the nomograph is for a nontapered rod string having an $F_c = 1.0$. N/N_o is equal to N/N_o' when F_c is equal to 1.0. When F_c is greater than 1.0, which is the case in tapered rod strings, the nomograph should be entered at the top of the chart, "Pump Setting Depth". Then a vertical line should be drawn from the actual pump setting depth until it intersects the F_c value for the particular rod string being considered. At that point a horizontal line should be drawn until it intersects the acceleration factor line c = 0.3. Then from that point a horizontal line intersects the corresponding strokes per minute, and a vertical line intersects the polished rod stroke length to consider.

Optimum Counterbalance Effect

One of the most important items in optimizing a beam pumping operation is properly counterbalancing the pumping unit. The optimum counterbalance effect is measured at 90^{o} crank angle at the polished rod. It can also be precalculated. The optimum CBE formula will be developed using a dynamometer card to reflect the upstroke and downstroke peak net torques.

Deriving the formula will involve calculating the well load torque and counterbalance torque at the upstroke peak net torque crank angle, and the well load torque and counterbalance torque at the downstroke peak net torque crank angle. Then by setting the resulting upstroke peak net torque equal to the downstroke peak net torque, it is possible to solve for the CBE that makes both peak net torques theoretically equal.

Optimum CBE Formula

Before deriving the optimum CBE formula, the following basic rules should be followed:

1. For well load calculations:

    a. When the polished rod is traveling up, the well load torque is always positive.

    b. When the polished rod is traveling down, the well load torque is always negative.

2. For rotary counterbalance calculations:

    a. When the rotary counterbalance is being lifted, the counterbalance torque is positive.

    b. When the rotary counterbalance is falling, the counterbalance torque is negative.

3. For considering structural unbalance:

    a. When the structural unbalance is positive, its value is subtracted from the measured well load.

b. When the structural unbalance is negative, its value is added to the measured well load.

c. Beam weights are considered as being positive structural unbalance for conventional (Class One Lever) pumping units.

Step 1. Calculate the upstroke peak net torque.

a. Determine the crank angle on the upstroke, Θ_1, at which the peak net torque occurs. This will require a torque calculation be made at crank angles on either side of the point where the peak net torque occurs.

b. Then calculate the well load torque at Θ_1.

$$\text{WL Torque @ } \Theta_1 = (\text{WL @ } \Theta_1 - \text{SU}) \times \overline{\text{TF}} \text{ @ } \Theta_1$$

c. Next calculate the counterbalance torque at Θ_1.

$$\text{CB Torque @ } \Theta_1 = (\text{CBE @ } 90^\circ - \text{SU}) \times \overline{\text{TF}} \text{ @ } 90^\circ \times \text{LACF @ } \Theta_1$$

d. Now calculate the upstroke peak net torque by subtracting the counterbalance torque at Θ_1 from the well load torque at Θ_1.

$$\text{Upstroke Peak Net Torque} = [(\text{WL}_{\Theta_1} - \text{SU}) \times \overline{\text{TF}}_{\Theta_1}] - [(\text{CBE}_{90^\circ} - \text{SU}) \times \overline{\text{TF}}_{90^\circ} \times \text{LACF}_{\Theta_1}]$$

Figure 24 shows the locations of these torque calculations on the dynamometer card.

Step 2. Calculate the downstroke peak net torque.

a. Determine the crank angle on the downstroke, Θ_2, at which the peak net torque occurs. This will require a torque calculation be made at crank angles on either side of the point where the peak net torque occurs.

b. Then calculate the counterbalance torque at Θ_2.

$$\text{CB Torque @ } \Theta_2 = (\text{CBE @ } 90^\circ - \text{SU}) \times \overline{\text{TF}} \text{ @ } 90^\circ \times \text{LACF @ } \Theta_2$$

c. Next calculate the well load torque at Θ_2.

$$\text{WL Torque @ } \Theta_2 = (\text{WL @ } \Theta_2 - \text{SU}) \times \overline{\text{TF}} \text{ @ } \Theta_2$$

d. Now calculate the downstroke net peak net torque by subtracting the well load torque at Θ_2 from the counterbalance torque at Θ_2.

$$\text{Downstroke Peak Net Torque} = [(CBE_{90^o} - SU) \times \overline{TF}_{90^o} \times LACF_{\Theta_2}] - [(WL_{\Theta_2} - SU) \times \overline{TF}_{\Theta_2}]$$

Figure 25 shows the locations of these torque calculations on the dynamometer card.

Step 3. Set the upstroke peak net torque equal to the downstroke peak net torque.

$$\overbrace{[(WL_{\Theta_1} - SU) \times \overline{TF}_{\Theta_1}] - [(CBE_{90^o} - SU) \times \overline{TF}_{90^o} \times LACF_{\Theta_1}]}^{\text{UPSTROKE PEAK NET TORQUE}} = \underbrace{[(CBE_{90^o} - SU) \times \overline{TF}_{90^o} \times LACF_{\Theta_2}] - [(WL_{\Theta_2} - SU) \times \overline{TF}_{\Theta_2}]}_{\text{DOWNSTROKE PEAK NET TORQUE}}$$

Step 4. Solve for the optimum counterbalance effect at 90^o crank angle at the polished rod by simplifying the equation developed in Step 3.

$$CBE = \frac{\overbrace{[(WL_{\Theta_1} - SU) \times \overline{TF}_{\Theta_1}]}^{\text{UPSTROKE WELL LOAD TORQUE}} + \overbrace{[(WL_{\Theta_2} - SU) \times \overline{TF}_{\Theta_2}]}^{\text{DOWNSTROKE WELL LOAD TORQUE}}}{\overline{TF}_{90^o} \times (LACF_{\Theta_1} + LACF_{\Theta_2})} + SU$$

This is the optimum counterbalance effect formula which will give a counterbalance effect load at 90^o crank angle at the polished rod which should make the upstroke and downstroke peak net torques equal.

After calculating the optimum counterbalance effect load, it should be checked for accuracy by recalculating the upstroke and downstroke peak net torques using the optimum value. They should now be equal if the optimum counterbalance effect load is correct.

Time Clocking Beam Pumping Wells

A study made several years ago indicated that at least one-half of the pumping wells surveyed had too large a subsurface pump installed. The results of such installations are devastating fluid pounds when wells are overpumped. Too large a pump is desirable in some instances, but the majority of the time it is not by design. In situations by design, other parts of the system are changed to compensate for the oversized pump. Too many times too large a pump is as a result of continuing an undesirable practice. It is still possible to live with too large a pump until the correct size can be installed. Some interim measures are to reduce the pump displacement by reducing the strokes per minute, shortening the stroke, decreasing back pressure on the tubing-casing annulus thereby increasing pump submergence, and by intermitting the producing period.

Time clocking is probably the most common type of intermitting. Figure 26 illustrates one of the easiest and most accurate ways to time clock a well. In order to use this method, good casing liquid levels are essential. A fluid level well sounder can be used to determine the liquid level in the tubing-casing annulus. As long as the liquid level build-up is on the straight line portion of the curve, there should be no loss of production due to intermitting the producing interval. However, when the build-up is on the curved portion of the curve, there will be a loss in production. The example problem on Figure 26 illustrates this loss in production.

If a well is pumping continuously and is pounding fluid more than one-quarter of the way down on the downstroke, it may be a candidate for time clocking. One procedure to determine the amount of pumping time is as follows:

1. Operate the pumping unit until the fluid level has stabilized. If the well is pumped off, it is not a candidate for time clocking by this procedure. Neither is the well a candidate if during the pumping stabilization period the fluid pound stabilizes in the upper one-quarter part of the downstroke.

2. Shut down the well for ten minutes. This ten minute period is called T_1.

3. Start the pumping unit and simultaneously start a stop watch.

4. Continue pumping until the fluid pound is one-quarter of the way down on the downstroke.

5. When that point is reached, stop the stop watch and record the time. This elapsed time is called T_2.

6. Using a fifteen minute percentage timer, the correct pumping schedule is:

$$\text{Pumping Time} = 24\left(\frac{T_2}{T_1 + T_2}\right)\text{hrs/day}$$

EXAMPLE DESIGN PROBLEM AND SOLUTION

A real-life situation beam pumping equipment design problem and its optimum solution are included to present and illustrate pertinent design criteria. Time and space limitations will permit the inclusion of only the major mechanical and operational features. Other important items such as safe soil loading, foundation design, proper grounding, and the actual installation process have not been included, but they are important and should be designed with care.

The design problem will be separated into four separate problems for clarity.

Problem One

Determine the producing bottomhole pressure.

Problem Two

Determine the shut-in bottomhole pressure.

Problem Three

Determine the desired liquid production by calculating the well liquid capacity at a reduced producing bottomhole pressure of 185 psig, or 200 psia.

Problem Four

Determine the conventional beam pumping equipment required to produce the desired liquid found in solving Problem Three.

Basic Data

1. A waterflood producer has been completed at a plugged back depth of 8,872 ft. It has 7 in. O.D., 26 lb/ft casing cemented through the pay zone. The pay is perforated from 8,730 ft. to 8,870 ft. The midpoint of the perforation is at 8,800 ft.

2. The well has 6,034 ft. of 2 7/8 in. O.D. tubing installed. The pump intake is at 6,000 ft. from the surface.

3. The well is currently being produced with a conventional beam pumping unit.

4. The daily liquid production is 180 bbl. of 34^o API oil and 52.5 bbl. of 1.02 specific gravity water.

5. Sour gas is being produced up both the tubing and the tubing-casing annulus. The tubing gas production is 20 mcfpd, and the annulus gas production is 100 mcfpd. Its specific gravity, G, is 0.80. The amount of hydrogen sulfide can be described as moderate.

6. The producing casing pressure is 60 psia.

7. The producing fluid level in the annulus is 5,000 ft. from the surface.

8. The shut-in casing pressure is 300 psia, and the shut-in tubing pressure is 40 psia.

9. The shut-in fluid level in the annulus is 3,000 ft. from the surface.

10. The well does not produce any sand.

Problem One

Determine the producing bottomhole pressure.

Solution to Problem One

Before commencing to solve this problem --

Prepare a producing well sketch, Exhibit 1. Do not neglect this step! Determine when the Gilbert S curve, Figure 27, will be used. It will be used first to determine the pressure at 6,000 ft., the pump intake. It will next be used at 8,800 ft., the midpoint of the perforations.

1. Determine the pressure at the pump intake, P_x, using the following:

$$P_x = [(P_c + P_{ab})/C_g] + [(D_x - FL) \times S.G. \times 0.433 \times F_x]$$

$P_c + P_{ab}$ = Casing Pressure, psia = 60 psia

C_g = Gas Gradient Correction Factor, Figure 28 = 0.87

D_x = Distance from Surface to Pressure Point = 6,000 ft.

FL = Distance from Surface to Fluid Level = 5,000 ft.

S.G. = Specific Gravity of the Oil
= $141.5/(131.5 + 34^o) = 0.855$

0.433 = Fresh Water Gradient, lb/in^2-ft

F_x = Liquid Gradient Correction Factor from Figure 27

Determine a liquid gradient correction factor from Figure 27.

a = Area of Flow Conduit
= Casing-Tubing Annulus
= $[(6.276)^2 - (2.875)^2] \times (\pi/4) = 24.44 in.^2$

$$Q_1 = \text{Gas Flow Rate, mcfpd} = 100 \text{ mcfpd}$$

Let $F_{x1} = 1.0$

$$P_{x1} = (60/0.87) + [(6{,}000 - 5{,}000) \times 0.855 \times 0.433 \times 1.0]$$
$$= 69 + 370.2 = 439.2 \text{ psia}$$

$$Q_1/aP^{0.4} = 100/[24.44 \times (439.2)^{0.4}]$$
$$= 100/(24.44 \times 11.4) = 0.359$$

$$F_{x2} = 0.555$$

$$P_{x2} = 69 + (370.2 \times 0.555) = 274.5 \text{ psia}$$

$$Q_1/aP^{0.4} = 0.433$$

$$F_{x3} = 0.515$$

$$P_{x3} = 69 + (370.2 \times 0.515) = 259.7 \text{ psia}$$

$$Q_1/aP^{0.4} = 0.443$$

$$F_{x4} = 0.51$$

$P_{x4} = 69 + (370.2 \times 0.51) = \underline{257.8 \text{ psia}}$, which is the pressure at the pump intake.

Note: The preceeding procedure should be repeated until two successive trial pressures are within the limits of accuracy desired. Normally, the last P_x should be within five percent of the previously calculated P_x.

2. Now determine the pressure at the midpoint of the perforations, P_{wf}. First, find the specific gravity of the oil-water mixture.

$$S.G._{o+w} = 0.855 \times [180/(180 + 52.5)] + 1.02 \times (52.5/232.5)$$
$$= 0.662 + 0.230 = 0.892$$

Then find the pressure at the midpoint of the perforations.

$$P_{wf} = 257.8 + [(8{,}800 - 6{,}000) \times S.G. \times 0.433 \times F_x]$$

Let $F_{x1} = 1.0$

$$P_{x1} = 257.8 + (2{,}800 \times 0.892 \times 0.433 \times 1.0)$$
$$= 257.8 + 1{,}081.5 = 1{,}339.3 \text{ psia}$$

$$a = (6.276)^2 \times (\pi/4) = 30.94 \text{ in.}^2$$

$$Q_2 = \text{Gas Flow Rate, mcfpd} = 120 \text{ mcfpd}$$

$$Q_2/aP^{0.4} = 120/[30.94 \times (1{,}339.3)^{0.4}]$$
$$= 120/(30.94 \times 17.81) = 0.218$$

$F_{x2} = 0.655$

$P_{x2} = 257.8 + (1{,}081.5 \times 0.655) = 966.2$ psia

$Q_2/aP^{0.4} = 0.248$

$F_{x3} = 0.63$

$P_{x3} = 257.8 + (1{,}081.5 \times 0.63) = \underline{939.1 \text{ psia}}$, which is the producing bottomhole pressure, P_{wf}, at the midpoint of the perforations.

Problem Two

Determine the shut-in bottomhole pressure.

Solution to Problem Two

Before commencing the solution, prepare a shut-in well sketch, Exhibit No. 2. Do not neglect this step!

1. Now commence to calculate the shut-in bottomhole pressure at the midpoint of the perforations, $\overline{P}_r$.

$\overline{P}_r = [(P_c + P_{ab})/C_g]$ + oil column pressure + mixed liquid column pressure

$= (300/0.92) + [1{,}000 \times 0.855 \times 0.433] + [4{,}800 \times 0.892 \times 0.433]$

$= 326.1 + 370.2 + 1{,}853.9 = \underline{2{,}550.2 \text{ psia}}$, which is the shut-in bottomhole pressure at the midpoint of the perforations.

Problem Three

Determine the desired liquid production by calculating the well liquid capacity at a reduced producing bottomhole pressure of 185 psig, or 200 psia.

Solution to Problem Three

Use the single IPR curve developed by J. V. Vogel, Figure 29. Find the liquid capacity of the well at a producing bottomhole pressure of 200 psia.

1. Calculate the dimensionless number which represents the producing bottomhole pressure at the midpoint of the perforations with a liquid production of 232.5 BPD, as a fraction of the maximum shut-in bottomhole pressure at the midpoint of the perforations.

$$P_{wf}/\overline{P}_r = 939.1/2{,}550.2 = 0.368$$

2. Using Vogel's IPR curve, determine the dimensionless number which represents the producing rate at 939.1 psia, as a fraction of the maximum producing rate.

$$q_{o(939.1\ \text{psia})} / q_{o(\text{max.})} = 0.82$$

3. Now calculate the dimensionless number which represents the producing bottomhole pressure at 200 psia, as a fraction of the shut in bottomhole pressure, both at the midpoint of the perforations.

$$P_{wf}/\overline{P}_r = 200/2{,}550.2 = 0.0784$$

4. Using Vogel's IPR curve, determine the dimensionless number which represents the producing rate at 200 psia, as a fraction of the maximum producing rate.

$$q_{o(200\ \text{psia})} / q_{o(\text{max.})} = 0.98$$

5. Calculate the producing rate when the producing bottomhole pressure is 200 psia.

$$\frac{q_{o(200\ \text{psia})}}{q_{o(939.1\ \text{psia})}} = \frac{X\ \text{BLPD}}{232.5\ \text{BLPD}}$$

$$\frac{0.98}{0.82} = \frac{X\ \text{BLPD}}{232.5\ \text{BLPD}}$$

X = 277.9 BLPD, which is desired liquid production at a producing bottomhole pressure of 200 psia at the midpoint of the perforations.

Problem Four

Determine the conventional beam pumping equipment required to produce the desired liquid found in solving Problem Three.

Assumptions for Problem Four

1. Assume a volumetric efficiency of 70 percent. Correspondingly, the pump displacement required will be 278 BLPD/0.70 = 397 BLPD.

2. The pump intake cannot be placed below the casing perforations and must be at least 15 feet above the top of the perforations to remain out of the turbulent zone. Assume the pump intake depth to be 8,730 ft. minus 15 ft., or 8,715 ft.

3. Producing casing pressure will increase as the relationship $(BLPD)^2$ increases. For example, C.P. at 278 BLPD will equal 60 psia x $(278/232.5)^2$ = 85.8 psia. C_g for (8,730 ft. - 15 ft. = 8,715 ft.) will be 0.786. The pressure at the pump intake if the well pumps off will be 85.8/0.786 = 109.2 psia. Fluid column pressure from 8,715 ft. to 8,800 ft. will be 200 psi - 109 psi = 81 psi. It will only be equal to (8,800 ft. - 8,715 ft.) x 0.433 x 1.02, or equal to 37.5 psi, assuming dead salt water. Therefore, the pump intake pressure will be greater than 109 psia + (81 psi - 38 psi), or 156 psia. If the pump intake pressure had been calculated to be less than the gas column pressure, the required capacity would have to be decreased.

4. Table 1 indicates a 1.75 in. pump should be considered. It also indicates that API Grade C rods are impractical. Since Grade D rods should not be used in this hydrogen sulfide environment, an alternative is to investigate Oilwell E rods, or equivalent. These rods are said to allow a maximum dynamic stress of 50,000 psi, regardless of the load range, providing the well is effectively inhibited. The minimum yield strength for Oilwell E rods is 60,000 psi. A 0.9 safety factor limits this to a maximum stress of 54,000 psi.

5. Design for an $N/N_o' \leq 0.35$ and a $F_o/Sk_r \leq 0.50$. These two factors will control, and c, the acceleration factor, will be less than 0.225.

6. S_p will be less than 0.80 S if the Oilwell stated capabilities of the E rods are fully utilized. Assume an $S_p/S \leq 0.75$ for Oilwell E rods.

7. Centralized sinker bars should be run above the pump.

8. The tubing should be anchored above the pump.

Solution to Problem Four

1. Find the pump displacement.

$$PD = 278/0.70 = 397 \text{ BLPD} = 0.1166\, S_p N D^2$$

Assume $S_p = 0.75$ S
$D = 1.75$ in.

Select stroke lengths, S, from Table 2.2 Supplement 1, dated December, 1972, API Std 11E, and manufacturers' catalogs.

2. Find the values for N, c, and N/N_o.

SN = 397/(0.1166 x 0.75 x 3.0625) = 1,482.4

S	N	c	N/N_o
124	11.95	0.251	0.425
144	10.29	0.216	0.366
145	10.22	0.215	0.364
168	8.82	0.185	0.314

3. Select S = 145 and N = 10.22.

   Since $N/N_o \div F_c = N/N_o'$, assume that $N/N_o' = 0.33$ since an F_c of 1.1 is also a reasonable value to assume.

4. Select a rod string.

$$G = 1.02$$
$$F_o = 0.34GD^2H = 0.34 \times 1.02 \times 3.0625 \times 8{,}715 = 9{,}256 \text{ lb.}$$

   With $N/N_o' = 0.33$, $S_p/S = 0.75$, Figure 4.1, API RP 11L indicates:

$$F_o/Sk_r = 0.44$$
$$Sk_r = F_o/0.44 = 9{,}256/0.44 = 21{,}036 \text{ lb.}$$

$$k_r = Sk_r/S = 21{,}036/145 = 145.08$$

$$1/k_r = 0.00689 = E_rL$$
$$E_r = 0.00689/L = 0.00689/8{,}715 = 0.791 \times 10^{-6} \text{ in/lb-ft}$$

   From Table 4.1, API RP 11L, select a rod string with an $E_r \geq 0.791 \times 10^{-6}$ in/lb-ft. A 76 rod string meets this requirement. A 76 rod string has $E_r = 0.795 \times 10^{-6}$ in/lb-ft, $W_r = 1.855$ lb/ft, and $F_c = 1.088$.

5. Check the calculated stress against the allowable stress.

$$W = W_rL = 1.855 \times 8{,}715 = 16{,}166 \text{ lb.}$$
$$W_{rf} = W[1.0 - (0.128 \times G)] = W[1.0 - (0.128 \times 1.02)] = 16{,}166 \times 0.8694 = 14{,}055 \text{ lb.}$$

$$F_o = 9.256 \text{ lb.}$$

$$1/k_r = 0.795 \times 10^{-6} \times 8{,}715 = 6.928 \times 10^{-3} \text{ in/lb}$$

$$Sk_r = S/(1/k_r) = 20{,}928 \text{ lb.}$$

$$F_o/Sk_r = 0.4423$$

$$N/N_o = 10.22 \times (8{,}715/245{,}000) = 0.364$$

$$N/N_o' = 10.22 \times (8{,}715/245{,}000) \times 1.088 = 0.334$$

From Figure 4.2, API RP 11L,

$$F_1/Sk_r = 0.68$$

$$\begin{aligned} PPRL &= W_{rf} + [(F_1/Sk_r) \times Sk_r] \\ &= 14{,}055 + (0.68 \times 20{,}928) \\ &= 28{,}286 \text{ lb.} \end{aligned}$$

$$\begin{aligned} \text{Stress on Top Rod} &= PPRL/a_{7/8"} \\ &= 28{,}286/0.601 \\ &= 47{,}065 \text{ psi, which is less than 50,000 psi maximum dynamic stress limit for Oilwell E rods.} \end{aligned}$$

So, Oilwell E rods would not be overloaded.

6. Check to see if the pump can be unseated. Assume a maximum unseating stress $\leq 60{,}000 \times 0.9 \leq 54{,}000$ psi.

Calculate F_o for a seating nipple having D = 2.28 in.

$$\begin{aligned} F_o &= 0.34\ GD^2H \\ &= 0.34 \times 1.02 \times (2.28)^2 \times 8{,}715 \\ &= 15{,}711 \text{ lb.} \end{aligned}$$

Calculate the weight of 3/4" rods in well fluid.

$$\begin{aligned} \text{Weight} &= 1.63 \times 8{,}715 \times 0.625 \times 0.8694 \\ &= 7{,}719 \text{ lb.} \end{aligned}$$

Calculate the 3/4 in. buoyancy on 7/8" rods.

$$\begin{aligned} \text{Buoyancy} &= \text{Area}_{3/4" \text{ rod}} \times \text{Length}_{7/8" \text{ rods}} \times 0.433 \times G \\ &= 0.442 \times 8{,}715 \times 0.375 \times 0.433 \times 1.02 \\ &= 638 \text{ lb.} \end{aligned}$$

Calculate the load on the top 3/4 in. rod when unseating the pump.

$$\begin{aligned} \text{Load} &= 15{,}711 + 7{,}719 - 638 \\ &= 22{,}792 \text{ lb.} \end{aligned}$$

Calculate the unseating stress in the top 3/4" rod.

$$\begin{aligned} \text{Stress} &= 22{,}792/0.442 \\ &= 51{,}566 \text{ psi, which is less than 54,000 psi, which is 0.9 of the minimum yield strength of E rods.} \end{aligned}$$

So, pump can be unseated.

Determine the pull on the top 7/8" rod with a stress of 54,000 psi on the top 3/4" rod.

$$\begin{aligned} \text{Pull} &= (54{,}000 \times 0.442) + \\ &\quad (0.375 \times 8{,}715 \times 2.22 \times 0.8694) + 638 \\ &= 23{,}868 + 6{,}307.7 + 638 \\ &= 30{,}814 \text{ lb.} \end{aligned}$$

7. Redetermine the pumping speed.

$$\begin{aligned} PD &= 397 \text{ BLPD} = 0.1166\, S(S_p/S)D^2N \\ (S_p/S)N &= 397/0.1166SD^2 \\ &= 397/0.1166 \times 145 \times (1.75)^2 \\ &= 7.667 \end{aligned}$$

$$F_c = 1.088$$

$$F_o/Sk_r = 0.4423$$

Redefine N so that $(S_p/S)N$ will equal or exceed 7.667.

N	N/N_o'	S_p/S	$(S_p/S)N$
10.21	0.3338	0.75	7.658
10.22	0.334	0.75	7.665
10.23	0.3345	0.753	7.703

Select N = 10.22 SPM.

8. Fill out API RP 11L Calculation Sheet. (Exhibit 3)

9. Select surface equipment.

   a. Gear Box selection

      Calculated Peak Torque = 687,631 in-lb

      Assume a 912,000 in-lb gear box.
      687,631/912,000 = 0.754

      From Figure 30, new unit efficiency = 0.85
      Minimum gear box torque = 687,631/0.85
      = 808,978 in-lb

      Select a unit with a 912,000 in-lb gear box.

   b. Minimum beam capacity

$$\begin{aligned} \text{Capacity} &\geq \text{PPRL} \times 1.10 \\ &= 28{,}286 \times 1.1 \\ &= 31{,}115 \text{ lb.} \end{aligned}$$

The nearest Table 2.2, Supplement 1, API Std 11E capacity is 36,500 lb.

c. Maximum stroke length

$$\begin{aligned} \text{Length} &\geq 145 \times 1.10 \\ &= 159.5 \text{ in.} \end{aligned}$$

Select a 912-365-168 pumping unit. Refer to manufacturers' catalogs. One manufacturer's designation is C912D-365-168.

d. Counterbalance to order

$$\begin{aligned} \text{Counterbalance} &= \text{CBE} \times 1.10 \\ &= 19{,}804 \times 1.10 \\ &= 21{,}784 \text{ lb. at the polished rod at } 90^{o} \\ &\quad \text{with a 145 in. stroke.} \end{aligned}$$

Consider ordering sufficient counterbalance to operate with a 168 in. stroke.

e. Prime mover selection

Find the beam pumping unit horsepower efficiency factor.

First, find the ratio of 4,960 times PRHP divided by the API gear box torque rating.

$$\begin{aligned} (4{,}960 \times \text{PRHP})/912{,}000 &= (4{,}960 \times 33.74)/912{,}000 \\ &= 0.1835 \end{aligned}$$

Then using Figure 31, find the beam pumping unit horsepower efficiency factor.

$$\text{Efficiency Factor} = 0.57$$

$$\begin{aligned} \text{Brake horsepower required} &= 33.74/0.57 \\ &= 59.19 \text{ HP} \end{aligned}$$

Assume an electric motor, NEMA Design D, and a cyclic load factor of 0.75.

$$\begin{aligned} \text{Motor to Order} &= 59.19/0.75 \\ &= 78.92 \text{ HP} \end{aligned}$$

Unless experience indicates a larger motor is necessary, order a 75 HP motor.

f. V-belt drive selection

One manufacturer's gear box reduction for the selected unit is 28.72. Using that value and the smallest and largest recommended sheaves, the resulting SPM can be calculated as follows:

GBPD Sheaves Available	Smallest Rec. PMPD	Resulting SPM	Largest Rec. PMPD	Resulting SPM
34"-10C	9"	10.32	17"	19.50
51"-10C	9"	6.88	17"	13.00
55 1/4"-10C (NR)	9"	6.35	17"	12.00
34"-8D	13"	14.91	17"	19.50
37"-8D	13"	13.70	17"	17.92
47.6"-8D	13"	10.65	17"	13.93
55.2"-8D	13"	9.18	17"	12.01

Eliminate the 34"-10C, 34"-8D, 37"-8D, and 47.6"-8D. Select the 51"-10C because the SPM range (6.88 to 13.00 SPM) is best.

Select the PMPD sheave size to pump 10.22 SPM.

PMPD Sheave = (10.22 SPM x 28.72 GBR x 51 in. GBPD) ÷ 1,120 RPM
= 13.37 in.

Select a 13" PMPD sheave.

Calculate V-belt drive design HP.

Initial installation = (Peak crankshaft torque x SPM) ÷ 70,000
= (687,631 x 10.22)/70,000
= 100.39 HP

Maximum design HP = (912,000 x 13.00)/70,000
= 169.37 HP

This requires 17 in. PMPD sheaves which are not readily available. Considering a 16 in. PMPD sheave, the design HP = (912,000 x 12.23)/70,000 = 159.34 HP

Using Figures 32 and 33, determine the number of C-section belts required.

Initial installation = 100.39 HP
HP per belt for 13 in., C-section belt = 19+
Number of belts = 100.39/19
= 5.28, or rounded to <u>6 belts</u>

Maximum design
HP per belt for 16 in., C-section belt = 24.5
Number of belts = 159.34/24.5
= 6.50, or rounded to <u>7 belts</u>

Note: Neither design calls for or requires the filling of all ten grooves in the unit sheave.

10. Selection of subsurface equipment

a. Gas anchor

A Poor Boy gas anchor must be used since the pump intake must be above the casing perforations.

Assuming the tubing at the pump will be 2 7/8 in. O.D., the dip tube will be 1 1/4 in. nominal.

I.D. of dip tube = 1.38 in.
O.D. of dip tube = 1.66 in.
I.D. area = 1.49 $in.^2$
O.D. area = 2.17 $in.^2$

Minimum annulus area = (397 BLPD/100) x 1.87
= 7.424 $in.^2$

Minimum mud anchor I.D. area = 7.42 + 2.17
= 9.59 $in.^2$

Minimum mud anchor I.D. = $(9.59/0.7854)^{0.5}$
= 3.49 in.

Select 4.0 in. O.D., 3.548 in. I.D.

I.D. Area = 9.90 $in.^2$

Pump displacement = 108.6 x $(1.75)^2$ x 0.7854
= 261.21 $in.^3$

Length of quieting space = 261.21 x [1.5/(9.90 - 2.17)]
= 391.82/7.73
= 50.69 in.

Area of mud anchor slots = 7.73 x 4
= 30.92 $in.^2$
= Sixteen 4 in. x 1/2 in. slots

Area of dip tube slots = 1.49 x 4
= 5.96 $in.^2$
= Twenty-four 2 in. x 1/8 in. slots

Construct a poor boy gas anchor work plan, Figure 34.

b. Subsurface pump

25-175 RHBC 16-6-2 (Section II, API Spec 11AX)

To meet NACE Standard MR-01-75 for moderate H_2S, no sand.

c. Sinker bars

Sinker bar factor = 0.45 in.^{2} (Table 3)

G = 1.02

L = 8,715 ft.

Theoretical weight = 0.45 x 8,715 x 0.433 x 1.02
= 1,732 lb.

Assume 20 percent of 1,732 lb. is needed = 0.20 x 1,732
= 346 lb.

One and one-half in. polished rods can be run in 2 7/8 in. O.D. tubing. (Table 4.2, API Std 11B states that one in. rods with slimhole couplings can be run in 2 7/8 in. O.D. tubing. Table 12.1, Supplement 1, dated March, 1979, to API Spec 11B states that 1 1/2 in. polished rods should be run with one in. sucker rods.)

Weight of 1 1/2 in. P.R. in air = $(1.5)^2$ x 0.7854 x 490/144
= 6.01 lb/ft

Weight in 1.02 G liquid = 6.01 x 0.869
= 5.23 lb/ft

Ft. of 1 1/2 in. P.R. required = 346/5.24
= 66.2 ft.

Select three 22 ft. x 1 1/2 in. polished rods. Figure 35 indicates the sinker bars must be centralized, because more than 40.7 ft. of 1 1/2 in. polished rods will buckle due to their own weight.

d. Sucker rod string

Use Oilwell E, or equivalent, rods.
62.5% of 3/4 in. = 5,247 ft.; round to 5,250 ft.
37.5% of 7/8 in. = 3,268 ft.; round to 3,275 ft.

Use one 3/4 in. Oilwell E pony rod and centralizer above the pump. Also add the 66 ft. of 1 1/2 in. polished rods as sinker bars (calculated above), centralizers, and one in. handling pony rods, or specify K-bars, or equivalent.

Use three crossover boxes.

7/8 in. space-out Oilwell E pony rods.

Note: Be sure the pony rods are the same material as the sucker rods.

e. Polished rod

Select a 22 ft. x 1 1/4 in. polished rod. (Table 12.1, API Spec 11B)

f. Tubing string

There are 6,034 ft. of 2 7/8 in. O.D. tubing now in the well; 8,715 ft. are required. Order 8,715 - 6,034 = 2,681 ft., rounded to 2,800 ft., of 2 7/8 in. O.D., J-55, 6.50 lb/ft tubing.

g. Economics

Due to constantly changing prices, an economic study is not included. However, three different units, tubing, rods and pump combinations were compared at the time this solution was prepared:

(1) 912 conventional unit, 2 7/8 in. tubing, 1.75 in. pump and a 76 string of Oilwell E rods.

(2) 912 conventional unit, 3 1/2 in. tubing, 1.75 in. pump and a string of 96 API Grade D rods.

(3) 1,280 air balanced unit, 2 7/8 in tubing, 1.50 in. pump and a string of 85 API Grade D rods.

At that time the economic study indicated that the 912 conventional unit with Oilwell E rods should be selected, because it was approximately $10,000 less than the other two unit combinations.

The full capabilities of the Oilwell E rods are utilized by the system selected. Oilwell does not recommend subjecting its E rods to stresses greater than 60,000 psi (the minimum yield strength of API Grade C sucker rods), even when unseating or pulling stuck pumps. These rods must be protected from corrosion.

h. Vibration analysis

The final system should be operated at 10.2 - 145 in. SPM, with a 1.75 in. pump, 76 E rods, have an N/N_o' of 0.334 and an F_o/Sk_r of 0.442. Figure 21 can be used to visualize the shape of the dynamometer card that will be generated. To construct that card to scale will require using other dimensionless load numbers. Exhibit 4 is a dynamometer card for this installation, constructed on a building block exhibit with the respective loads proportional to each other.

CONCLUSION

We hope this paper makes it possible for personnel directly involved with the selection and operation of beam pumping equipment to have a better understanding of the criteria associated with each part of the system. We also hope there will be a better understanding of the various loads acting on the pumping system during the pumping cycle. There are many opportunities for increasing production, reducing operating costs and increasing efficiencies by operating beam pumping equipment in an optimum manner. With current technology, it is now possible to size beam pumping equipment much more accurately on initial installations and to determine that such equipment is also sized correctly on existing installations. The design method presented in the paper is highly recommended for suitable wells.

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NOMENCLATURE AND DEFINITIONS

a	Cross section area of casing-tubing annulus, square inches
BLPD	Barrels liquid per day
BOPD	Barrels oil per day
c	Acceleration factor, $S(N)^2/70,500$
CBE	Counterbalance effect measured at the polished rod at the 90^o crank angle, pounds
C_g	Gas gradient correction factor
C.P.	Casing pressure, psig
D	Pump plunger diameter, inches
D_x	Depth from the surface to the pressure point under consideration, feet
E_r	Elastic constant of sucker rod string, inches per pound foot

E_t	Elastic constant for tubing string, inches per pound foot
F_c	Frequency factor, a constant of proportionality which depends on the sucker rod string design
FL	Distance from the surface to the fluid level, feet
F_o	Static fluid load on the gross plunger area, pounds; pounds per foot multiplied by H, the net lift in feet
F_x	Liquid gradient correction factor
F_1	Fluid load on the gross plunger area plus maximum upstroke dynamic effects, pounds
F_2	Maximum downstroke dynamic effects, pounds
F_3	Polished rod horsepower factor
F_o/Sk_r	Dimensionless sucker rod stretch load
F_1/Sk_r	A function of N/N_o and F_o/Sk_r
F_2/Sk_r	A function of N/N_o and F_o/Sk_r
F_3/Sk_r	A function of N/N_o and F_o/Sk_r
G	Specific gravity
GBPD	Gear box sheave pitch diameter, inches
GBR	Gear box speed reduction factor
H	Net lift, approximated by the distance from the surface to the operating fluid level in the tubing-casing annulus, feet
HP	Horsepower
I.D.	Inside diameter, inches
k_r	Spring constant of the total sucker rod string, and represents the load in pounds required to stretch the total sucker rod string one inch
k_t	Spring constant of the unanchored portion of the tubing, and represents the load in pounds required to stretch the unanchored portion of the tubing (between the anchor and the standing valve) one inch
$1/k_r$	Elastic constant for the total sucker rod string, inches per pound; also equals $E_r \times L$

$1/k_t$	Elastic constant for the unanchored portion of the tubing string, inches per pound, measured from the standing value to the tubing anchor; also equals $E_t \times L_{ua}$
L	Length of the sucker rod string, feet
LACF	Lever arm correction factor; the absolute sine value of an angle
mcfpd	Thousand cubic feet of gas per day
MPRL	Minimum load at the polished rod during the pumping cycle, pounds
N	Pumping speed, strokes per minute
N_o	Natural frequency of a nontapered sucker rod string, strokes per minute
N_o'	Natural frequency of a tapered sucker rod string, strokes per minute
N/N_o	Dimensionless pumping speed factor for a nontapered sucker rod string
N/N_o'	Dimensionless pumping speed factor for a tapered sucker rod string
0	Zero load or the zero line on a dynamometer card when the load on the dynamometer is zero
O.D.	Outside diameter, inches
P_{ab}	Atmospheric pressure, psia
P_c	Casing pressure, psig
PD	Bottomhole pump displacement assuming 100% volumetric efficiency, barrels per day; also equals $0.1166 \times S_p \times N \times D^2$
PMPD	Prime mover sheave pitch diameter, inches
PPRL	Peak load at the polished rod during the pumping cycle, pounds
$\overline{P}_r$	Reservoir pressure, psia
PRHP	Horsepower at the polished rod
psi	Pounds per square inch
psia	Pounds per square inch absolute
psig	Pounds per square inch gauge
PT	Peak torque, inch-pounds

P_{wf}	Producing bottomhole pressure, psia
P_x	Pressure at the pressure point D_x under consideration, psia
q_o	Liquid producing rate at some value less than maximum, barrels per day
$q_{o(max.)}$	Maximum producing rate at 100% drawdown pressure rate with reservoir pressure at maximum, barrels per day
$q/q_{o(max.)}$	Producing rate as a fraction of maximum producing rate
$Q/aP^{0.4}$	Ordinate from Figure 27 where Q = mscfpd, a = in.^{2} and P = psia
R	Reference line drawn on every card by the dynamometer reference stylus
RPM	Revolutions per minute
S	Polished rod stroke length, inches
S.G.	Specific gravity
Sk_r	Pounds of static load necessary to stretch the total sucker rod string an amount equal to the polished rod stroke length
S_p	Bottomhole pump stroke, inches
SPM	Pumping speed, strokes per minute
SU	Structural unbalance, pounds
SV	Standing valve; standing valve load, pounds
T_a	Adjustment for peak torque for values of W_{rf}/Sk_r other than 0.3
$\overline{TF}$	Torque factor, inches
TV	Traveling valve; traveling valve load, pounds
T_1	The ten minute shut-in period used in time clocking calculations
T_2	Elapsed time used in time clocking calculations, minutes
V	Downward fluid velocity in a gas anchor, feet per second
W	Total weight of the sucker rod string in air, pounds
WL	Well load measured at the polished rod, pounds
W_{rf}	Total weight of the sucker rod string in well fluid, pounds

W_{rf}/Sk_r Weight of the sucker rod string in well fluid compared to the weight necessary to stretch the sucker rod string one polished rod stroke length

0.1166 Pump constant for a one inch pump

0.128 Weight of a cubic foot of fresh water, 62.4 pounds, divided by the weight of a cubic foot of steel, 489 pounds

0.34 Weight of a column of fresh water in a cylinder having a diameter of one inch and a height of one foot, pounds

0.433 Weight of a column of fresh water having a volume defined by a cross sectional area of one square inch and a height of one foot, pounds

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FIGURES, TABLES AND EXHIBITS

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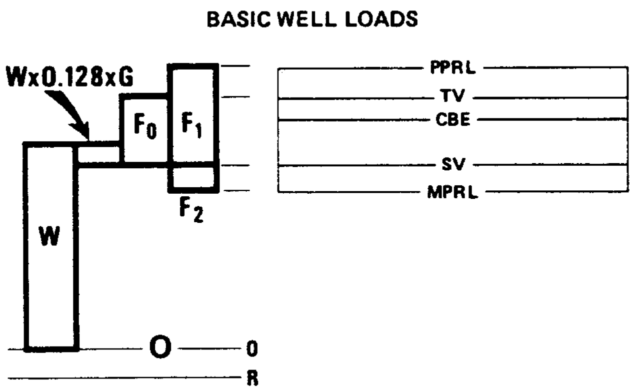
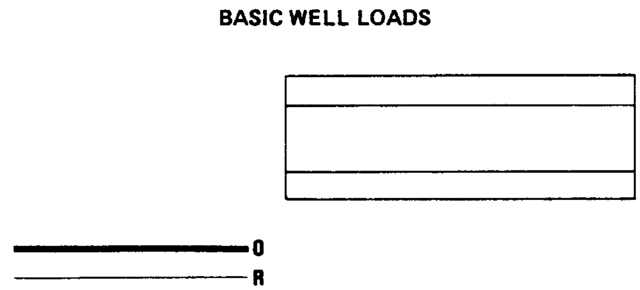


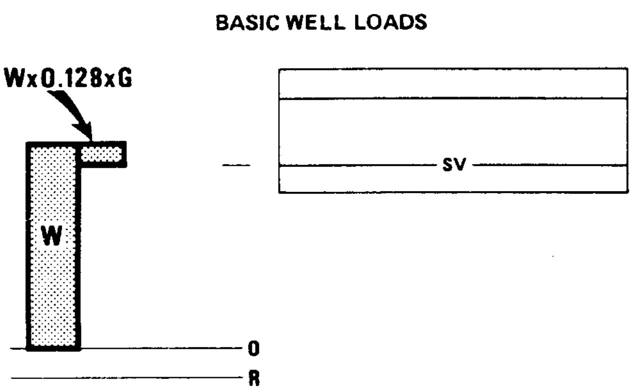
BUILDING BLOCK APPROACH

FIGURE 1



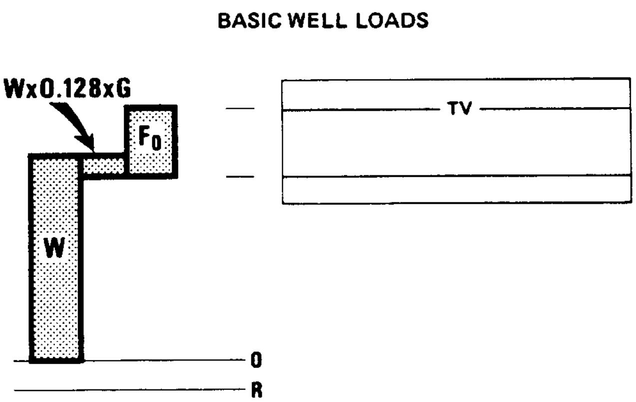
ZERO LOAD

FIGURE 2



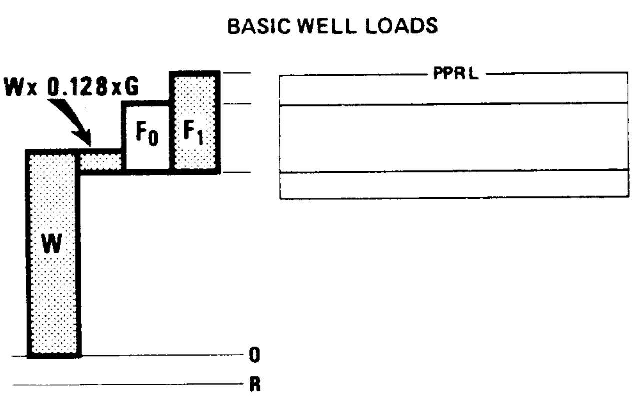
STANDING VALVE LOAD

FIGURE 3



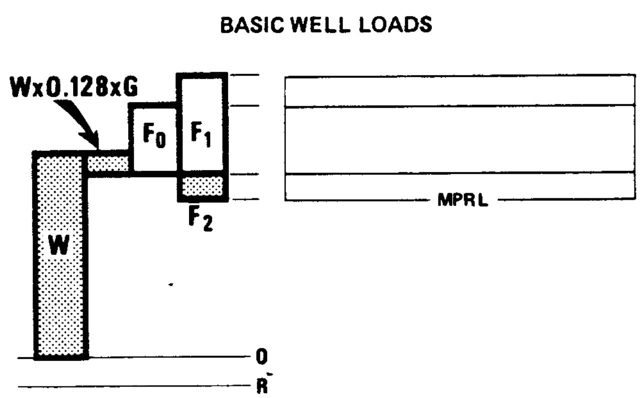
TRAVELING VALVE LOAD

FIGURE 4



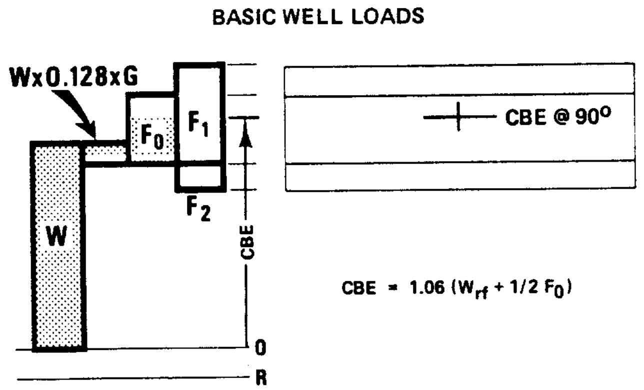
PEAK POLISHED ROD LOAD

FIGURE 5



MINIMUM POLISHED ROD LOAD

FIGURE 6



CALCULATED COUNTERBALANCE EFFECT
AT THE POLISHED ROD
AT 90° CRANK ANGLE
(CONVENTIONAL UNITS)

FIGURE 7

STANDING VALVE

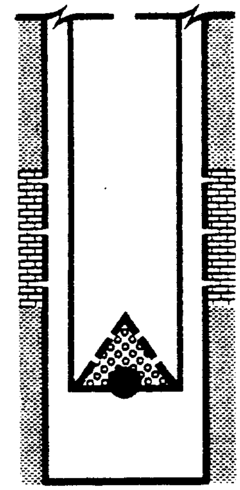


FIGURE 9

TRAVELING VALVE

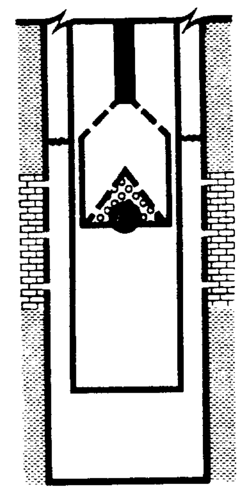


FIGURE 10

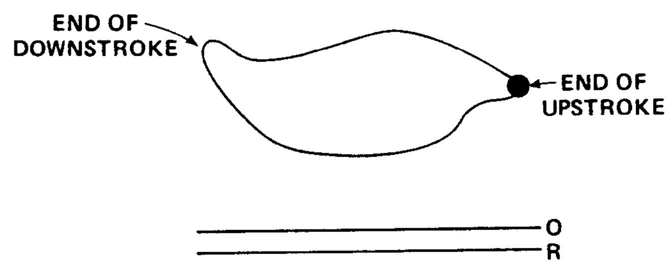
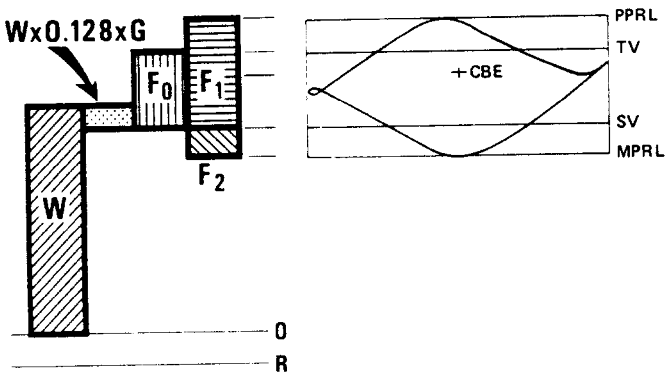


FIGURE 11

SIX BASIC BEAM PUMPING LOADS



$\boxed{W} = W_r \times L$

$W_{rf} = \boxed{W} - \boxed{W \times 0.128 \times G}$; $W_{rf} = W\,(1.0 - 0.128 \times G)$;

$W_{rf} = SV$

$\boxed{F_0} = 0.340 \times G \times D^2 \times H$; $TV = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_0}$;

$TV = W_{rf} + F_0$

$\boxed{F_1} = \frac{F_1}{Skr} \times Skr$; $PPRL = \boxed{W} - \boxed{W \times 0.128 \times G} + \boxed{F_1}$;

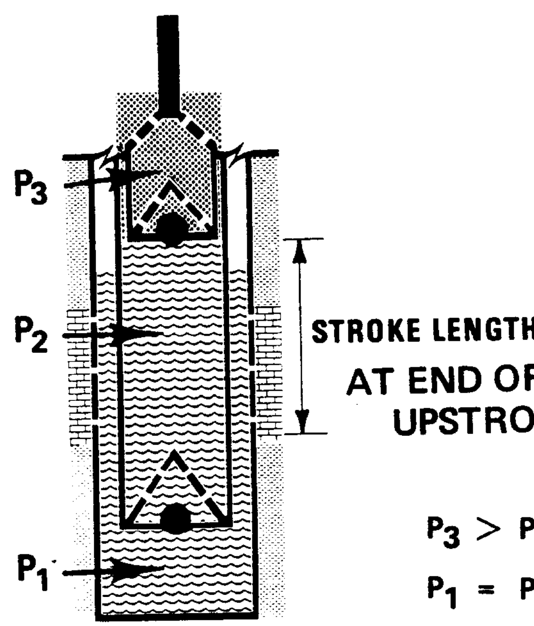
$PPRL = W_{rf} + F_1$

$\boxed{F_2} = \frac{F_2}{Skr} \times Skr$; $MPRL = \boxed{W} - \boxed{W \times 0.128 \times G} - \boxed{F_2}$;

$MPRL = W_{rf} - F_2$

$CBE = 1.06\,(W_{rf} + 1/2\ F_0)$

FIGURE 8



AT END OF UPSTROKE:

$P_3 > P_2$

$P_1 = P_2$

END OF UPSTROKE

FIGURE 12

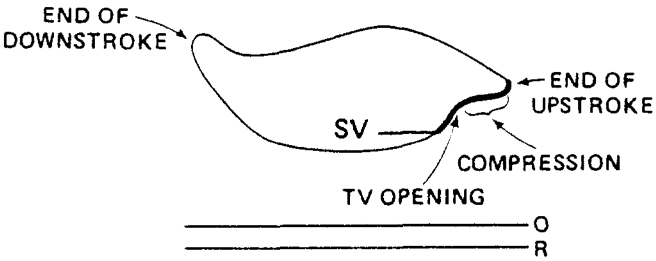


FIGURE 13

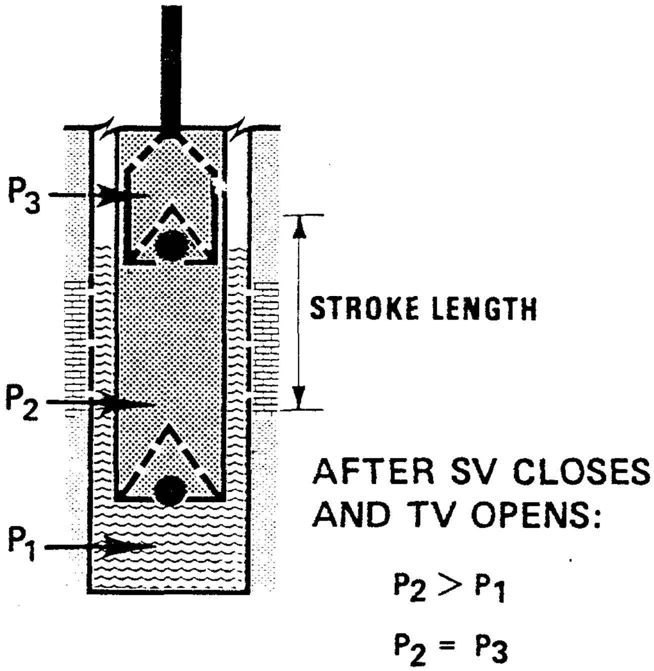


FIGURE 14

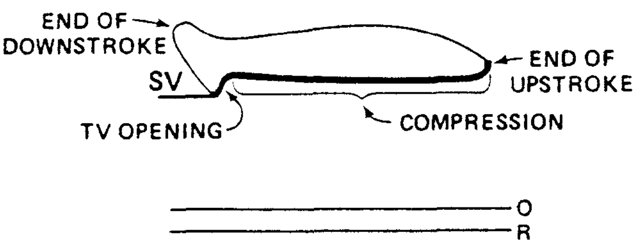


FIGURE 15

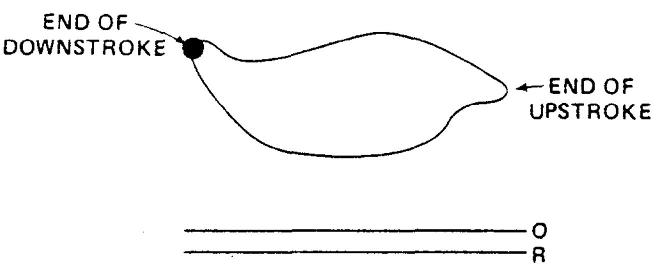


FIGURE 16

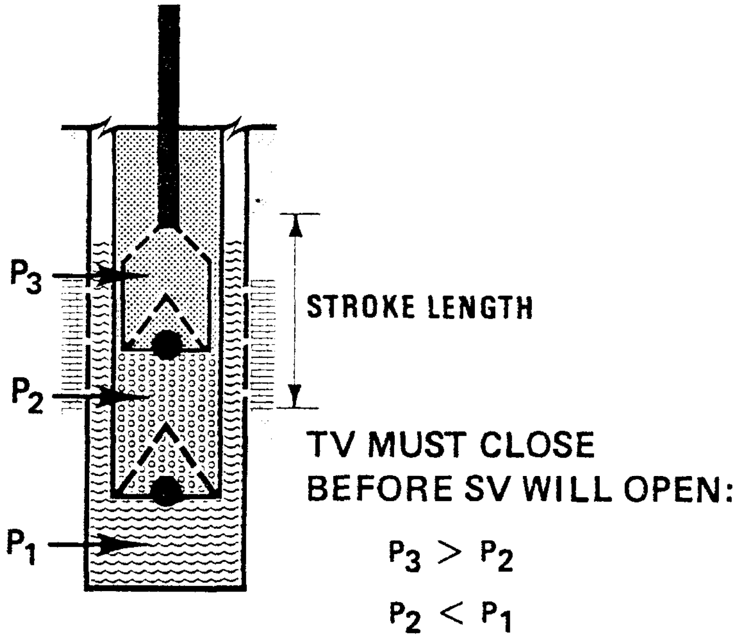


FIGURE 17

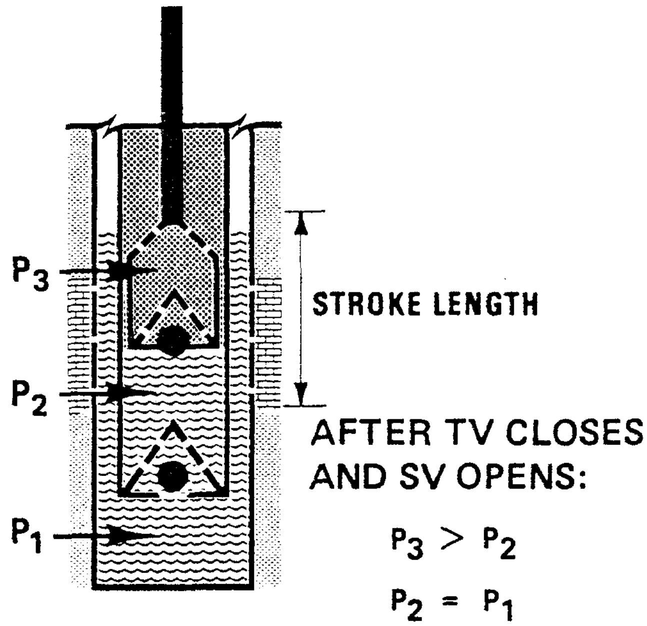
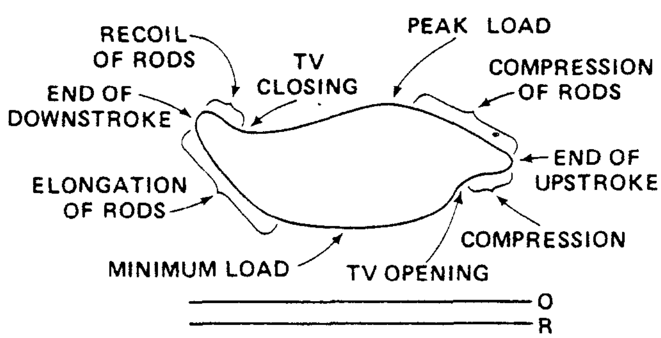
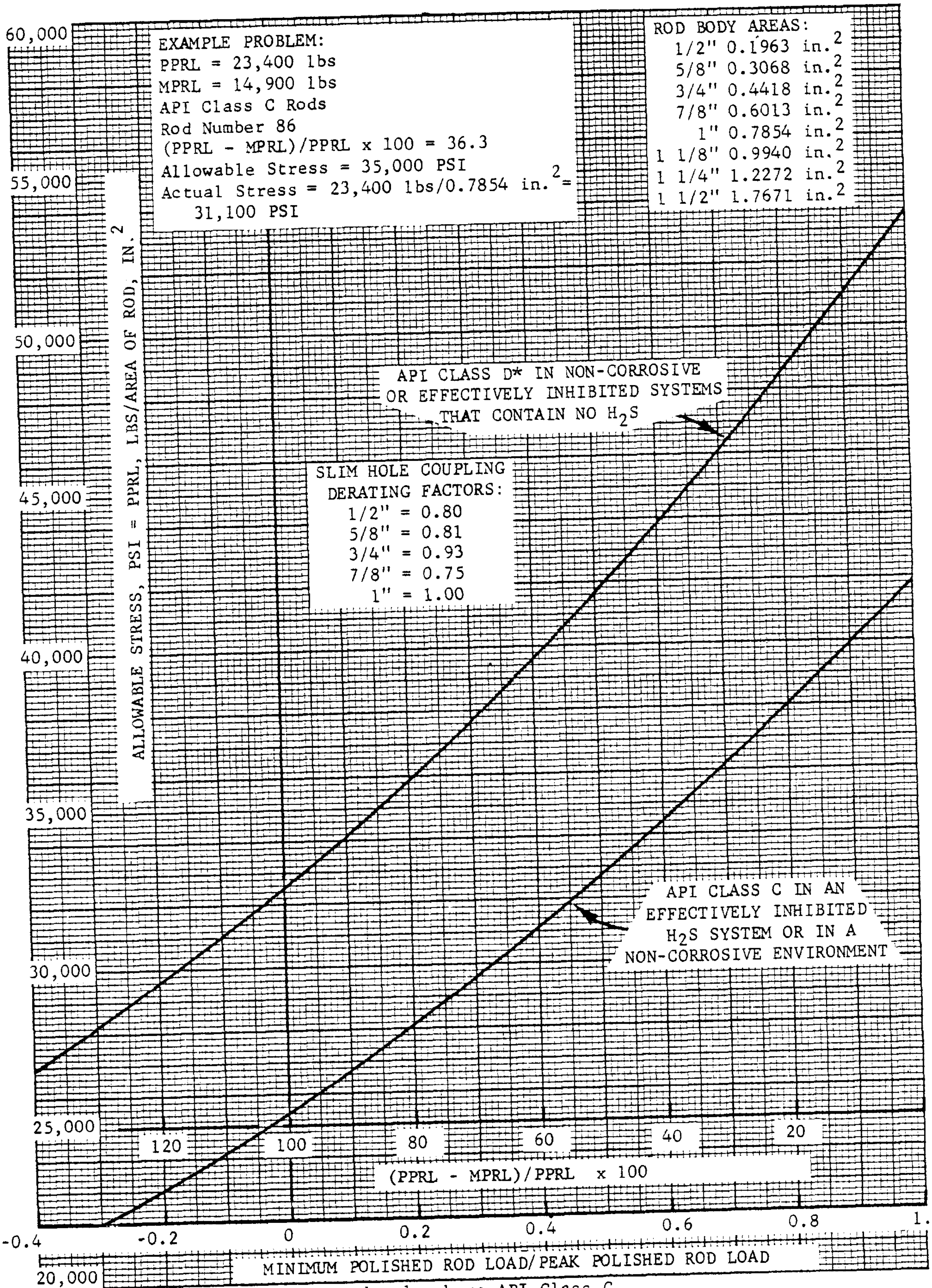


FIGURE 18



"TYPICAL" PUMPING CYCLE

FIGURE 19



*To be used only where API Class C rod capabilities are exceeded.

FIGURE 20

REPRESENTATIVE DYNAMOMETER CARDS

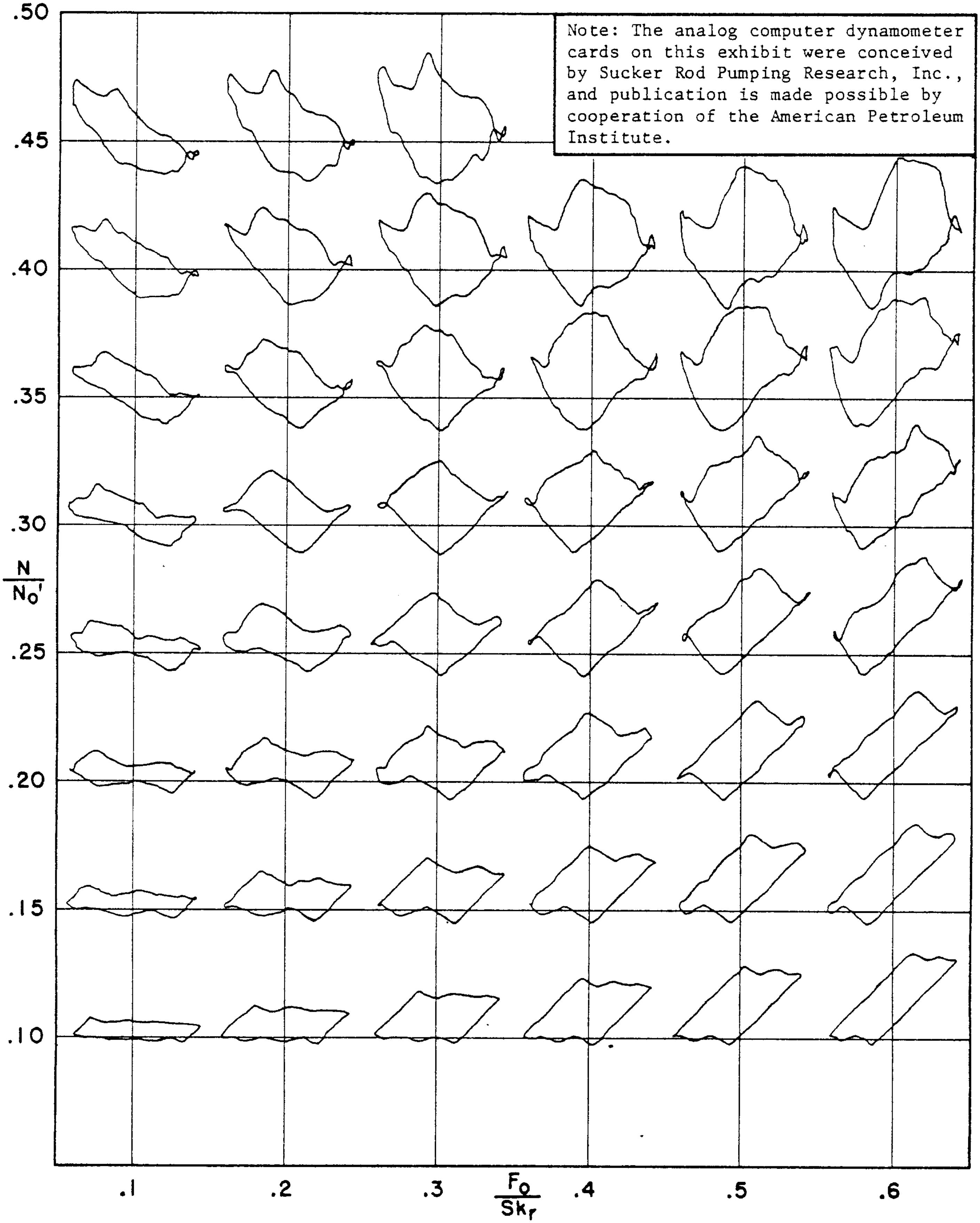


FIGURE 21

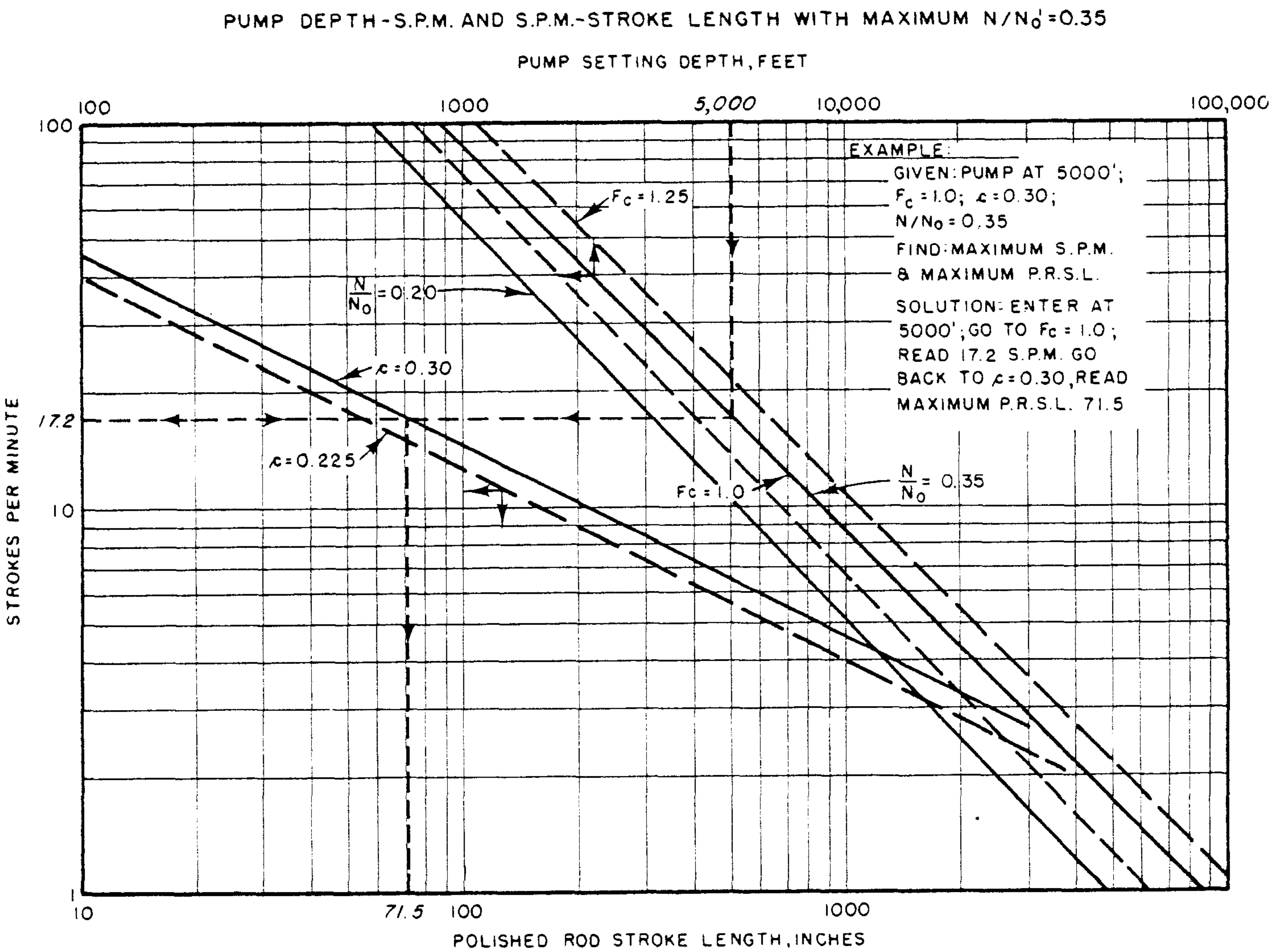


FIGURE 23

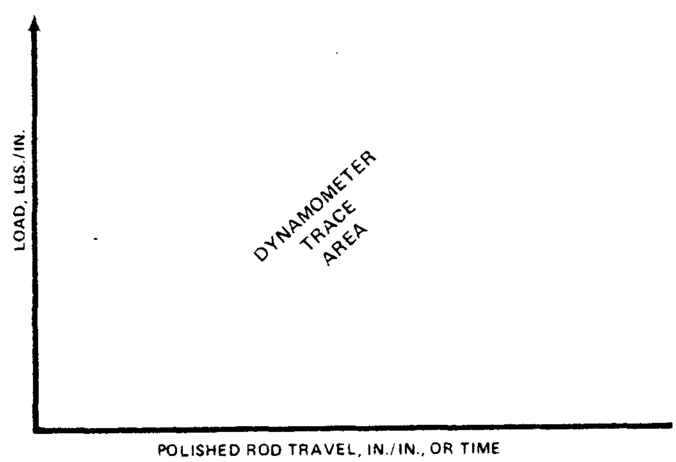


FIGURE 22

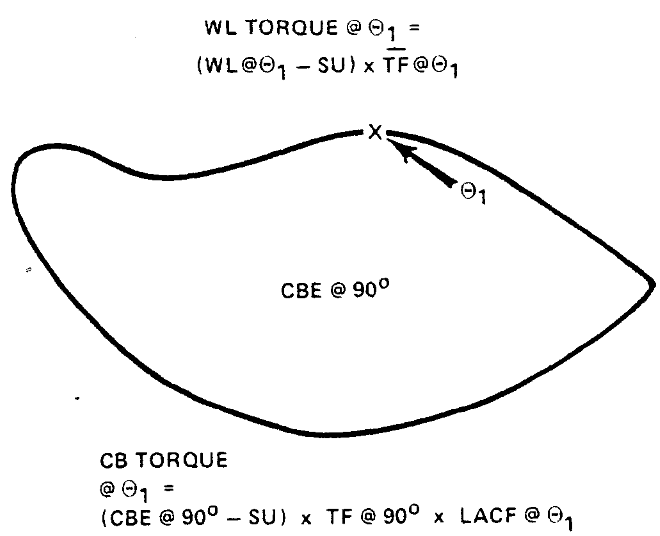
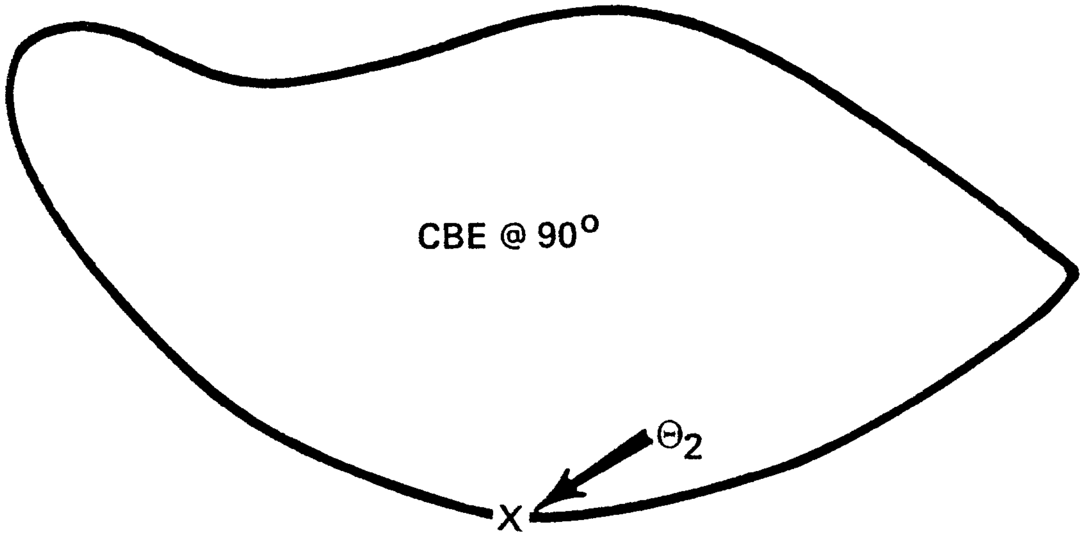


FIGURE 24

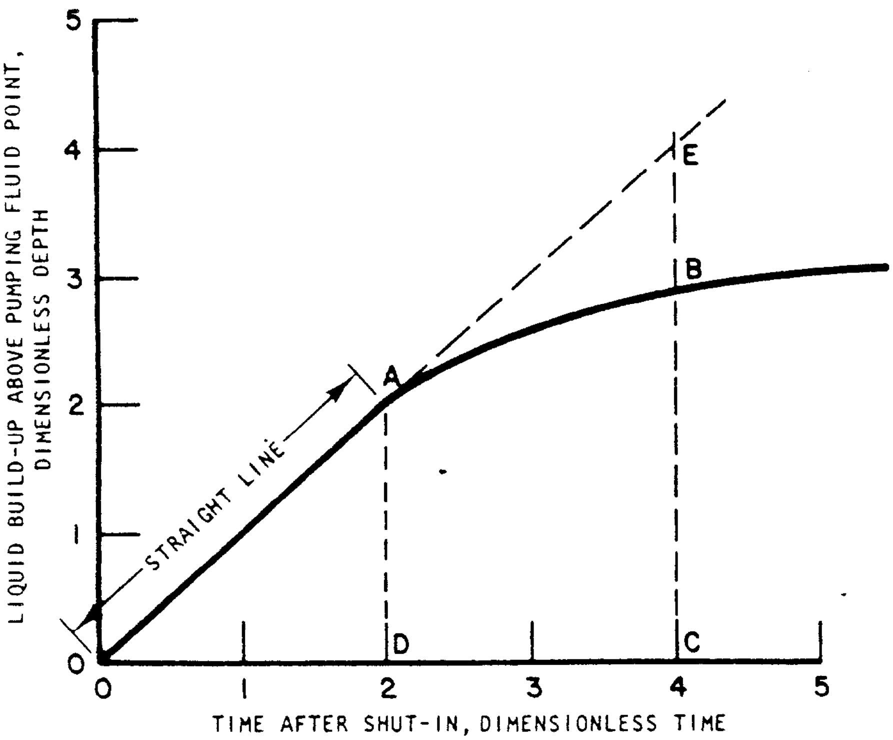
CB TORQUE @ Θ_2 =
(CBE @ 90^o – SU) x TF @ 90^o x LACF @ Θ_2



WL TORQUE @ Θ_2 =
(WL @ Θ_2 – SU) x $\overline{TF}$ @ Θ_2

FIGURE 25

TIME CLOCKING PUMPING WELLS



EXAMPLE:

1. PERCENT PRODUCTION LOST IF UNIT STARTS PUMPING BEFORE BUILD-UP REACHES A = 0
2. PERCENT PRODUCTION LOST IF PUMPING STARTS AT B = $100 \times \frac{\text{AREA ABE}}{\text{AREA OEC}}$
3. IN EXAMPLE, PERCENT LOST = $100 \times \frac{45.67 \text{ UNITS}}{369.77 \text{ UNITS}}$

   = 12.35 PERCENT

FIGURE 26

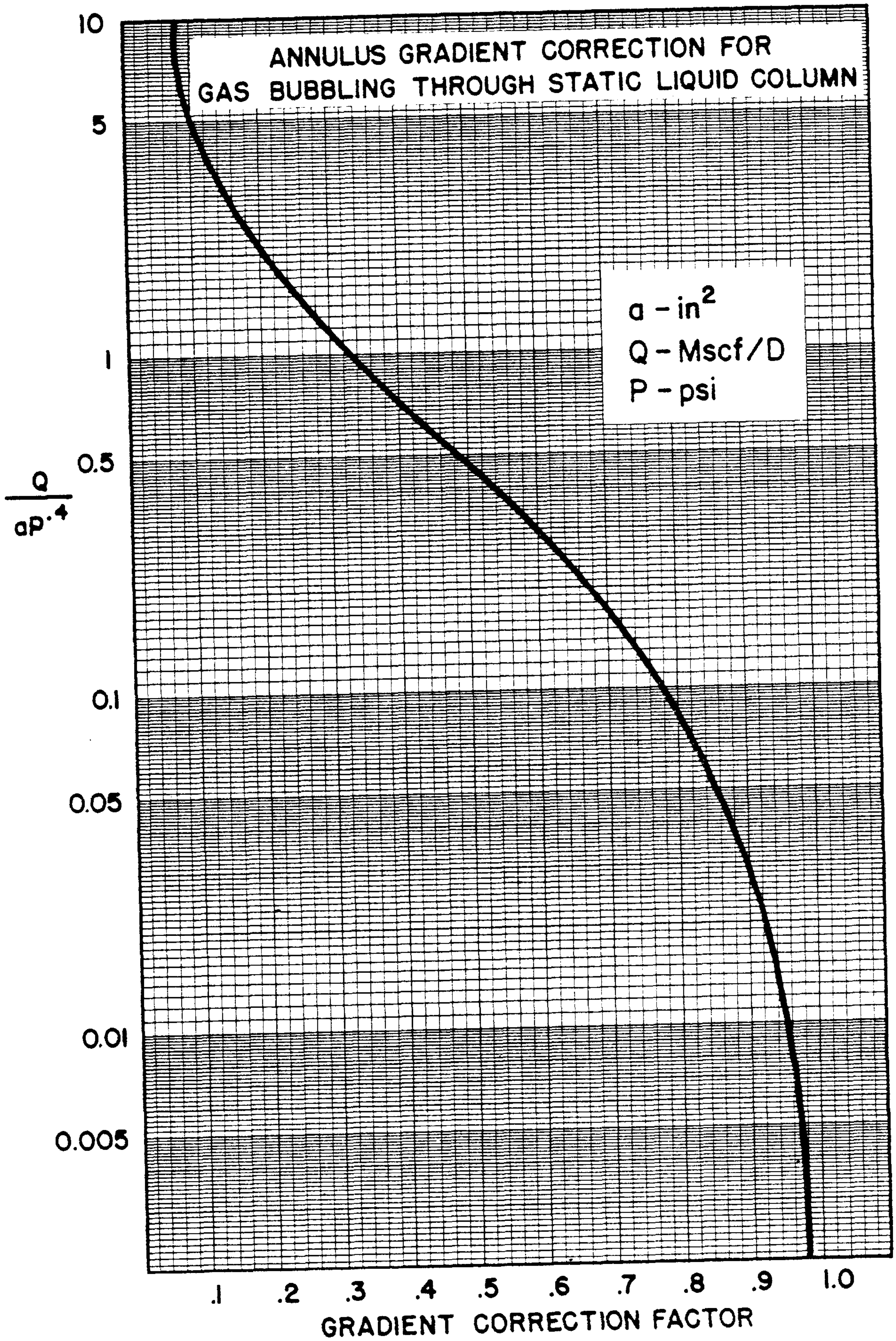


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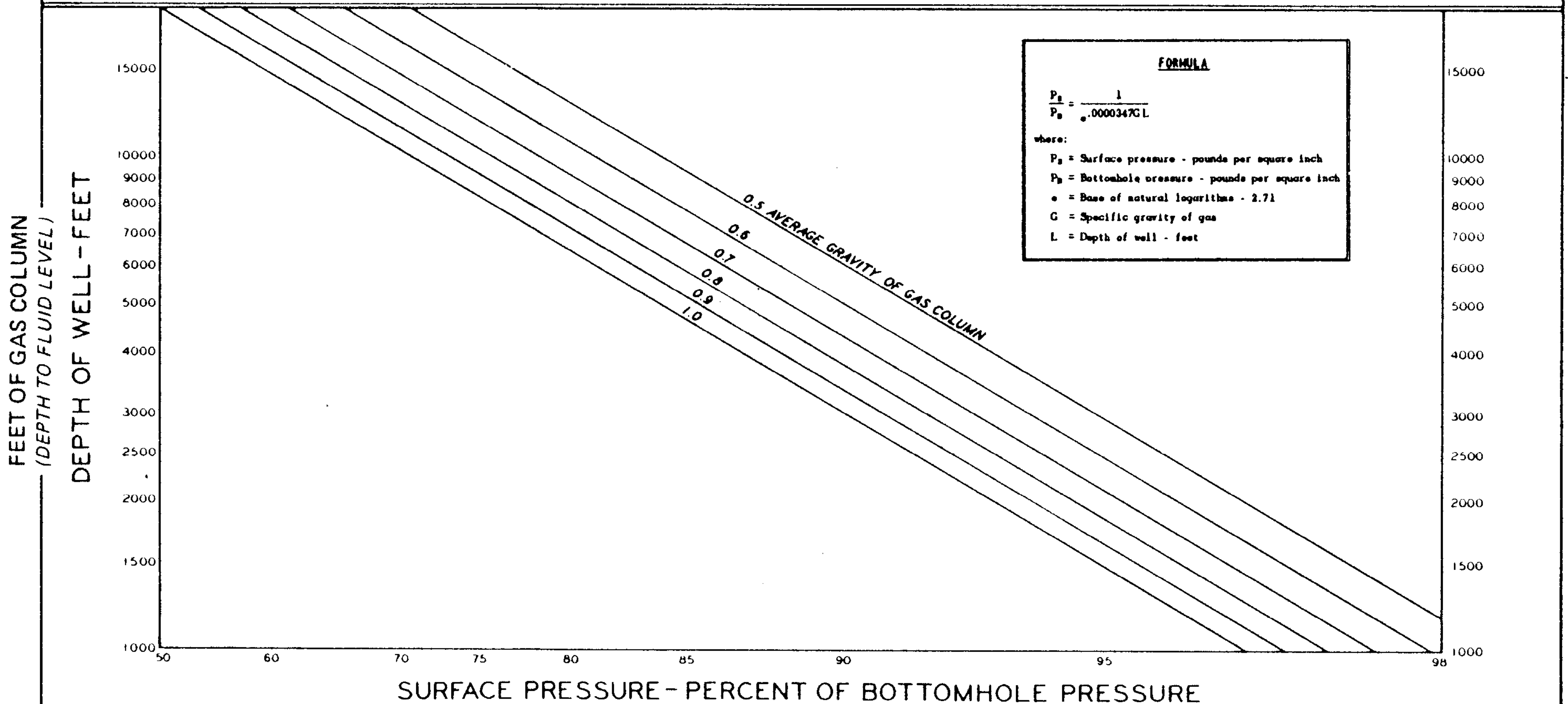
FIGURE 27

GAS GRADIENT CORRECTION FACTOR CURVES

Pressure Loss Due to the Weight of a Column of Gas

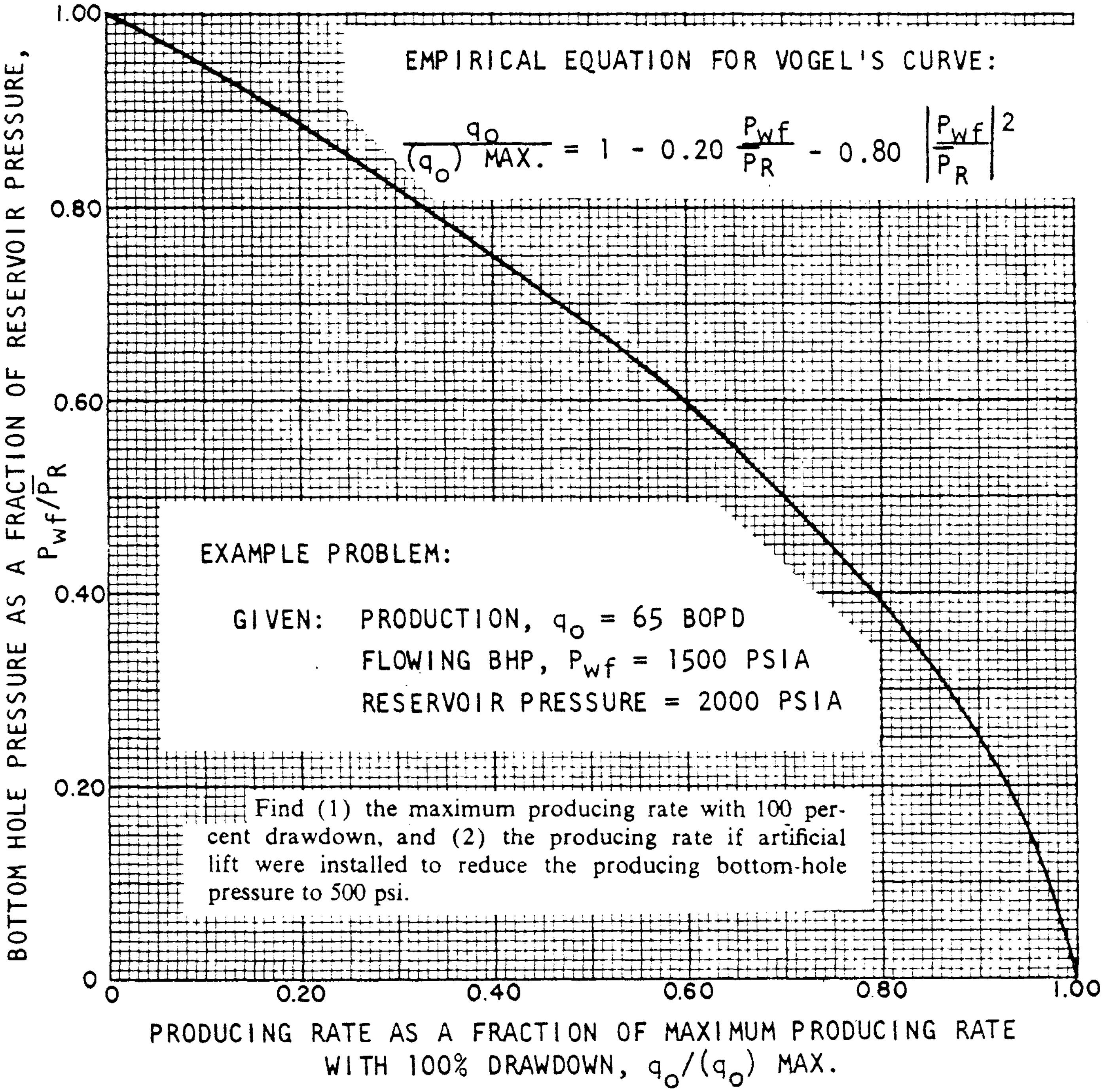


GAS GRADIENT CORRECTION FACTOR, C_g

Modified and Reproduced courtesy ARMCO Inc.

FIGURE 28

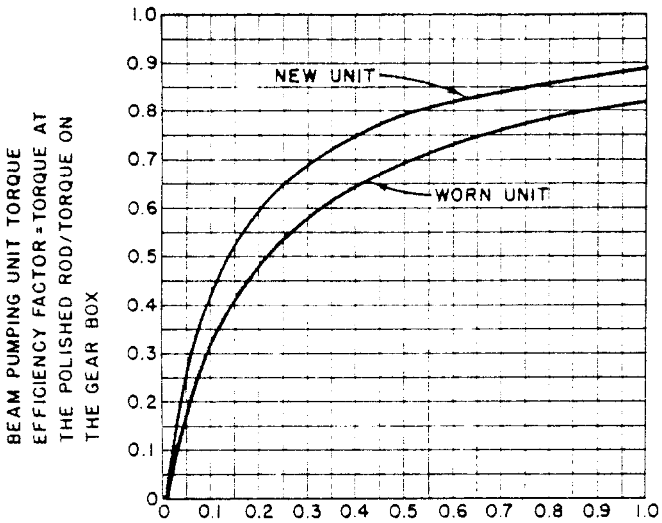
VOGEL'S CURVE FOR INFLOW PERFORMANCE RELATIONSHIP



The solution is: (1) with p_{wf} = 1,500 psi, $p_{wf}/\bar{p}_R$ = 1,500/2,000 = 0.75. From Fig. 5, when $p_{wf}/\bar{p}_R = 0.75$, $q_o/(q_o)_{max} = 0.40$, $65/(q_o)_{max} = 0.40$, $(q_o)_{max} = 162$ BOPD; (2) with p_{wf} = 500 psi, $p_{wf}/\bar{p}_R$ = 500/2,000 = 0.25. From Fig. 5, $q_o/(q_o)_{max} = 0.90$, $q_o/162 = 0.90$, q_o = 146 BOPD.

(AFTER J. V. VOGEL, SPE TRANSACTIONS, VOL. 243, 1968. COPYRIGHTED, 1968 BY THE SOCIETY OF PETROLEUM ENGINEERS. USED BY PERMISSION.)

FIGURE 29



BEAM PUMPING UNIT TORQUE EFFICIENCY FACTOR

FIGURE 30

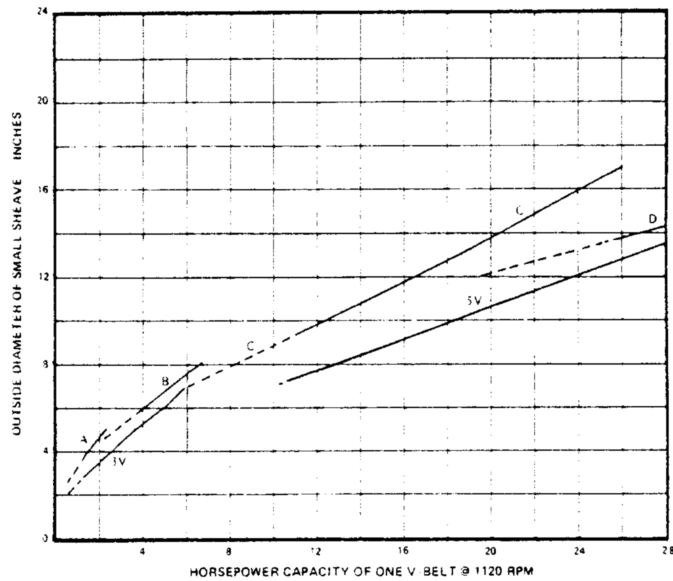
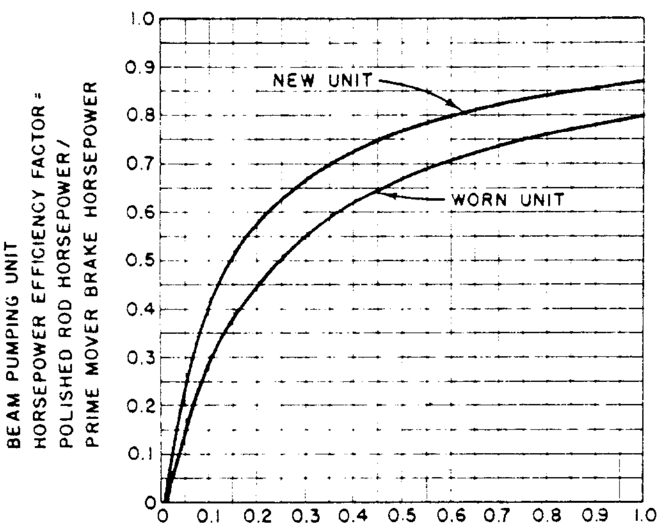


FIGURE 32



BEAM PUMPING UNIT HORSEPOWER EFFICIENCY FACTOR

FIGURE 31

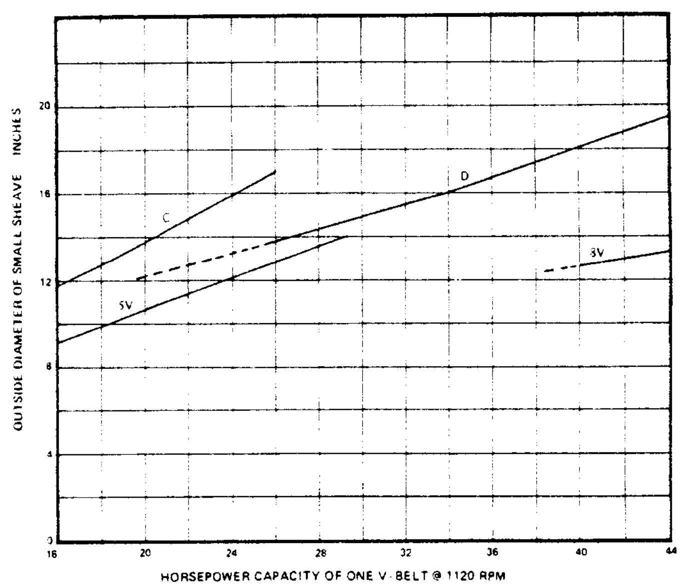


FIGURE 33

EXAMPLE DESIGN PROBLEM

POOR BOY GAS ANCHOR

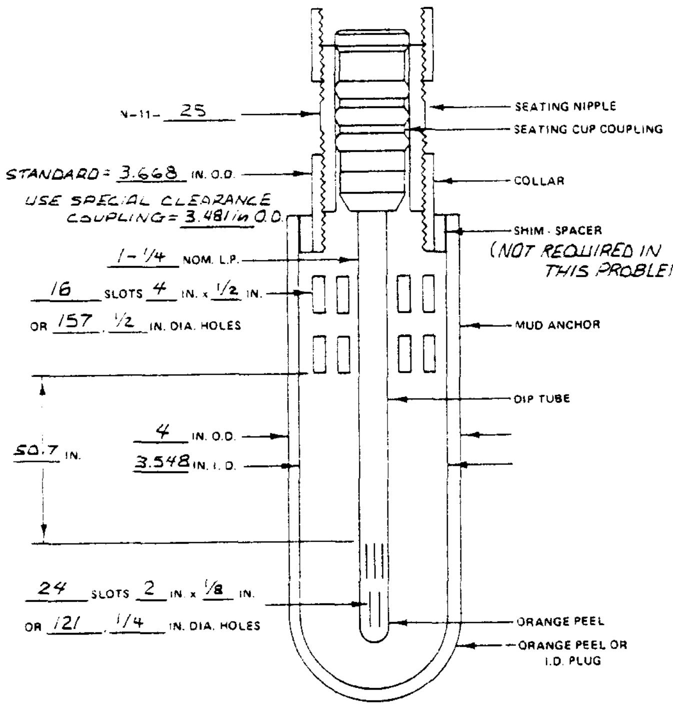


FIGURE 34

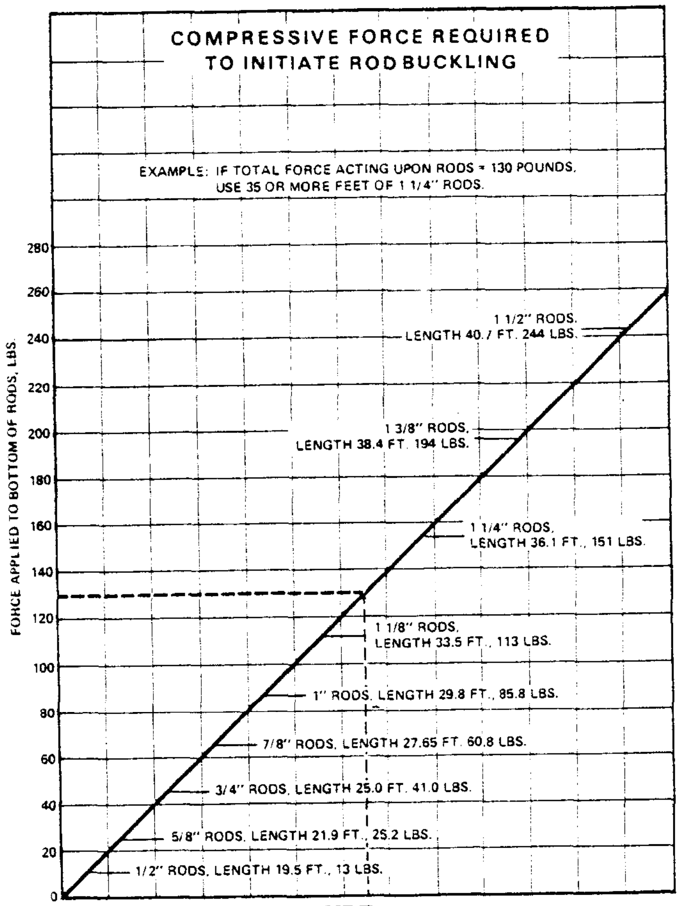


FIGURE 35

TABLE 1

PUMP PLUNGER SIZES RECOMMENDED FOR OPTIMUM DESIGN

PUMP DEPTH AND FLUID LEVEL	FLUID PRODUCTION, BARRELS PER DAY - 100% VOLUMETRIC EFFICIENCY - 1.00 SPECIFIC GRAVITY												
FEET	25	50	75	100	200	300	400	500	600	700	800	900	1000
1000	1.06	1.06	1.25	1.50	1.75 1.50	1.75 1.50	2.25 2.00	2.25 2.00	2.25 2.50	2.25 2.50	2.75 2.50	2.75 2.50	2.75
2000	1.06	1.06	1.25	1.50	1.75 1.50	1.75 1.50	2.25 2.00	2.25 2.00	2.25 2.50	2.25 2.50	2.75 2.50	2.75 2.50	2.75
3000	1.06	1.06	1.25	1.50	1.75 1.50	1.75 1.50	2.25 2.00	2.25 2.00	2.25 2.50	2.25 2.50	2.75 2.50	2.75 2.50	2.75
4000	1.06	1.06	1.25	1.50	1.75 1.50	1.75 1.50	2.25 2.00	2.25 2.00	2.25 2.50	2.25 2.50	2.75 2.50	2.75 2.50	2.75
5000	1.06	1.06	1.25	1.50	1.50	1.50	2.00	2.00	2.25 2.50	2.50	 2.50	 2.50	
6000	1.06	1.06	1.25	1.25	1.25	1.75 1.50	1.75 2.00	 2.00	 2.25	 2.25			
7000	1.06	1.06	1.06	1.25	1.50 1.25	1.50	 2.00	 2.00	 2.25	 2.25			
8000	1.06	1.06	1.25	1.25	1.50 1.25	 1.75	 1.75	 2.00					
9000	 1.06	 1.06	 1.06	 1.06	 1.50	 1.75							
10000	 1.06	 1.06	 1.06	 1.06	 1.50	 1.75							

LEGEND:

IN THIS TABULATION SURFACE PUMPING STROKES UP TO 120 INCHES ONLY ARE CONSIDERED. 2", 2-1/2" AND 3" NOMINAL TUBING ARE CONSIDERED. TOP LINE INDICATES PLUNGER DIAMETER, INCHES, TO USE WITH API CLASS C RODS. IF TOP LINE IS BLANK, CAPABILITIES OF CLASS C RODS WILL BE EXCEEDED.

BOTTOM LINE INDICATES PLUNGER DIAMETER, INCHES, TO USE WITH API CLASS D RODS. IF BOTTOM LINE IS BLANK, AND TOP LINE IS NOT, USE PLUNGER DIAMETER INDICATED ON TOP LINE.

IF BOTH LINES ARE BLANK, CAPABILITIES OF CLASS D RODS WILL BE EXCEEDED.

TABLE 2

STROKES PER MINUTE vs. STROKE LENGTH

STROKE LENGTH, IN.	SPM @ 70% OF FREE FALL SPEED OF SUCKER RODS	SPM* WHEN ACCELERATION FACTOR = 0.3
16	42.9	36.4
24	35.0	29.7
30	31.3	26.6
36	28.6	24.2
42	26.5	22.7
48	24.7	21.0
54	23.3	19.8
64	21.4	18.2
74	19.9	16.9
86	18.5	15.7
100	17.1	14.5
120	15.7	13.3
144	14.3	12.1
168	13.2	11.2
192	12.4	10.5
216	11.7	9.9
240	11.1	9.4
300	9.9	8.4

*Recommended maximum for design purposes.

TABLE 3

SINKER BAR FACTOR TABLE

		COLUMNS 3 a&b [(SEAT CONTACT O.D. AREA/I.D. AREA) MINUS 1.0]		COLUMN 2 x COLUMN 3a	COLUMN 2 x COLUMN 3b	
COLUMN 1 PLUNGER DIAMETER	COLUMN 2 PLUNGER AREA, IN.^{2}	HARBISON-FISCHER DATA	O'BANNON DATA	SINKER BAR FACTORS, IN.^{2}	SINKER BAR FACTORS, IN.^{2}	RECOMMENDED SINKER BAR FACTORS, IN.^{2}
1 1/16"	0.886	0.235	0.33	0.209	0.293	0.30
1 1/4"	1.227	0.219	0.26	0.269	0.319	0.30
1 1/2"	1.767	0.216	0.24	0.382	0.424	0.40
1 3/4"	2.405	0.156	0.19	0.375	0.458	0.45
2"	3.142	0.139	0.17	0.436	0.535	0.50
2 1/4"	3.976	0.120	0.15	0.477	0.595	0.55
2 1/2"	4.909	0.099	0.13	0.490	0.640	0.60
2 3/4"	5.940	0.099	0.13	0.588	0.772	0.70
3 3/4"	11.045	-	0.13	-	1.433	1.40

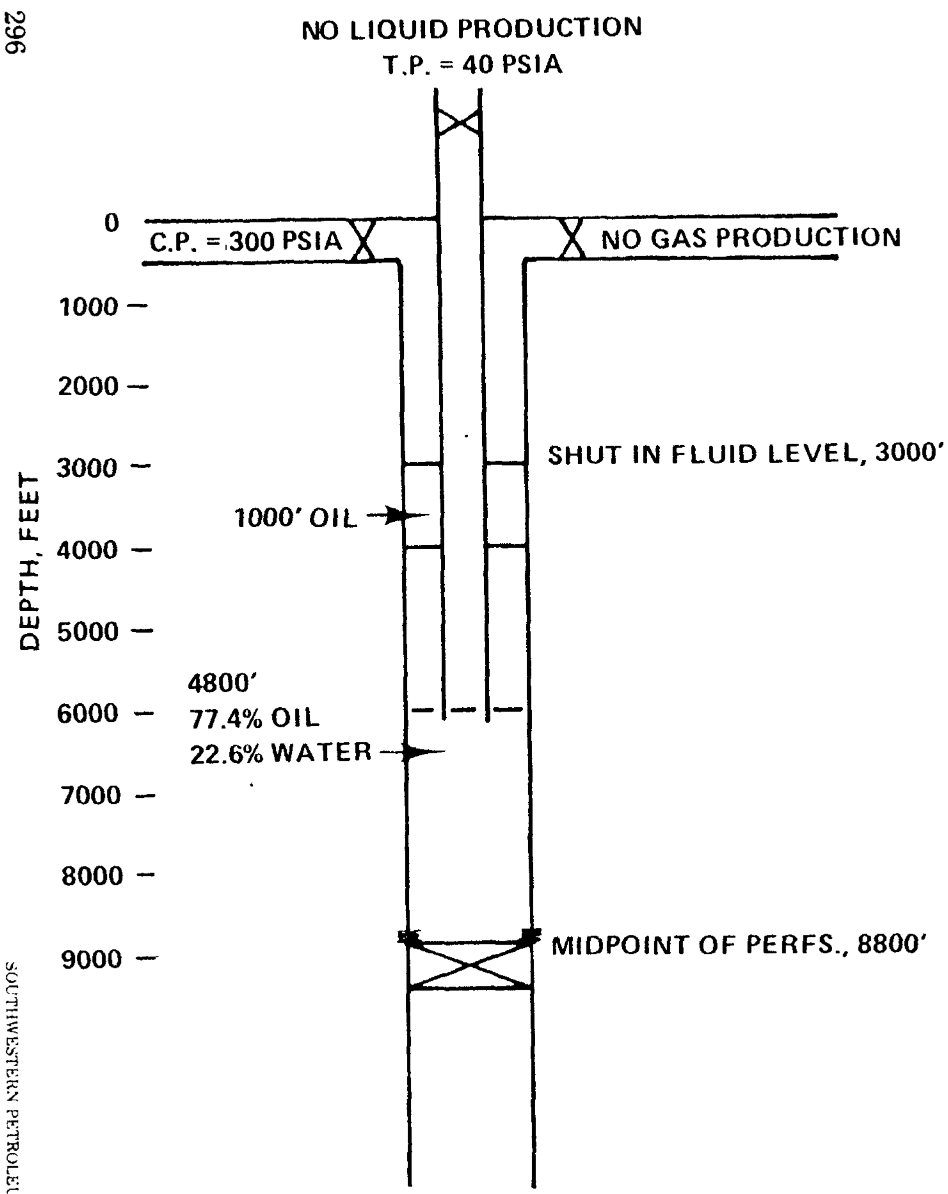


EXHIBIT 2

SHUT IN WELL

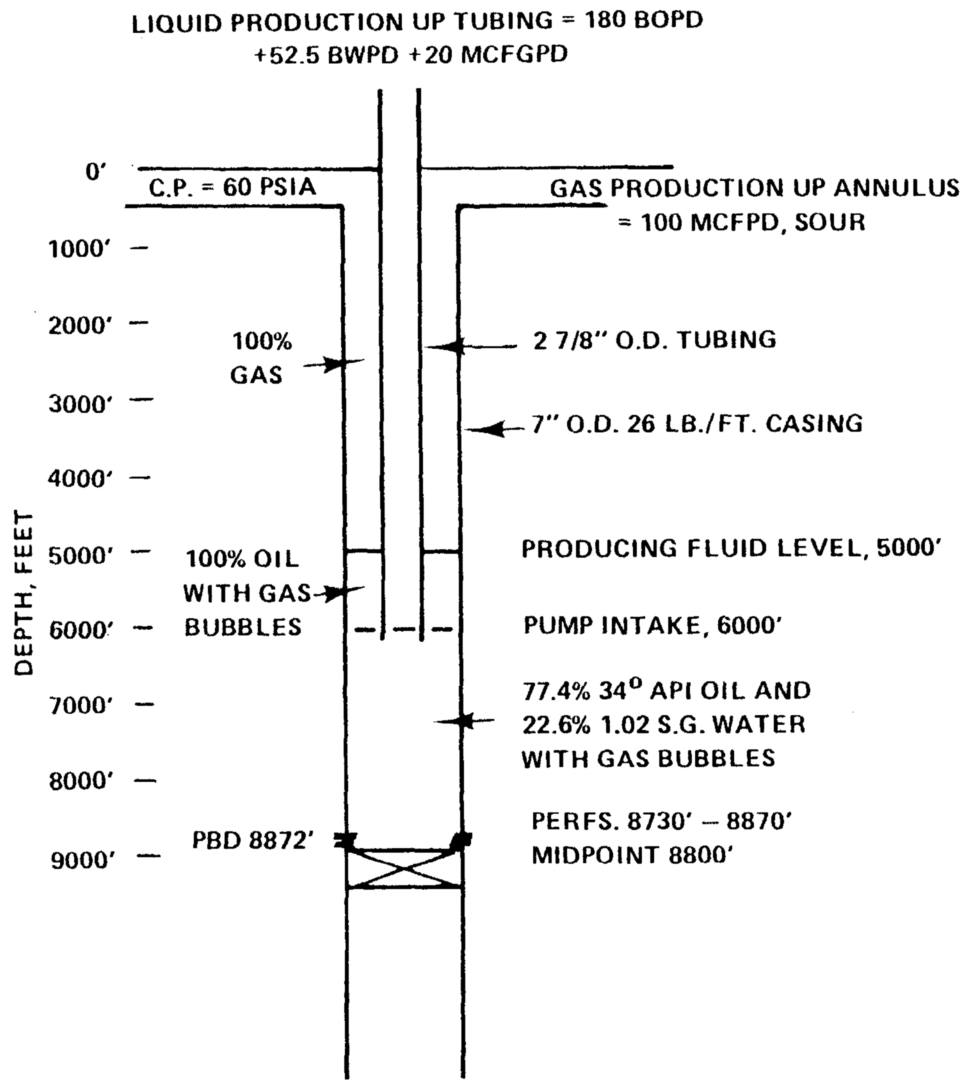


EXHIBIT 1

PRODUCING WELL

DESIGN CALCULATIONS SHEET
CONVENTIONAL SUCKER ROD PUMPING SYSTEM

Well Example Design Problem Calculated By ________ Date ________

Known or Assumed Data:
278 BFPD ÷ 0.70 Vol. Efficiency = 397 PD, Bbls. per day.
Fluid Level, H = 8,715 Ft. Pump Depth, L = 8,715 Ft.
Tubing Size 2 7/8 In. Is it anchored? Yes 8,615 No ____ Pumping Speed, N = 10.22 SPM
Length of Stroke, S = 145 In. Plunger Diameter, D = 1.75 In.
Specific Gravity of Fluid, G = 1.02 Sucker Rods 76
API Class: C, D, S.S., K, H.T. (Circle one) Oilwell E

Record Factors from Tables 1 & 2:

1. W_r = 1.855 (Table 4.1, Column 3)
2. E_r = 0.795×10^{-6} (Table 4.1, Column 4)
3. F_c = 1.088 (Table 4.1, Column 5)
4. E_t = 0.221×10^{-6} (Table 4.2, Column 5)

Calculate Non-Dimensional Variables:

5. $F_o = .340 \times G \times D^2 \times H$ = .340 × 1.02 × 3.0625 × 8,715 = 9,256 lbs. (Gross Plunger Load)
6. $1/K_r = E_r \times L$ = 0.795×10^{-6} × 8,715 = 5.928×10^{-3} In./Lb. (line 2 × L)
7. $Sk_r = S \div 1/k_r$ = 145 ÷ 6.928×10^{-3} = 20,928 Lbs. (S/line 6)
8. F_o/Sk_r = 9,256 ÷ 20,928 = 0.4423 (line 5/line 7)
9. N/N_o = NL ÷ 245,000 = 10.22 × 8,715 ÷ 245,000 = 0.364
10. $N/N_o' = N/N_o \div F_c$ = 0.364 ÷ 1.088 = 0.334 (line 9/line 3)
11. $1/K_t = E_t \times L$ = 0.221×10^{-5} × 100 = 0.0221×10^{-3} In./Lb. (line 4 × L)

Solve for S_p and PD:

12. S_p/S = 0.75 (Figure 4.1) (line 10 to line 8 to answer)
13. $S_p = [(S_p/S) \times S] - [F_o \times 1/k_t]$ = [0.75 (line 12) × 145 (S)] - [9,256 (line 5) × 0.0221×10^{-3} (line 11)] = 108.6 In.
14. $PD = 0.1166 \times S_p \times N \times D^2$ = 0.1166 × 108.6 (line 13) × 10.22 (N) × 3.0625 (D^2) = 396.3 Bbls. per day.

Determine Non-Dimensional Parameters:

15. $W = W_r \times L$ = 1.855 × 8,715 = 16,166 Lbs. (line 1 × L)
16. $W_{rf} = W[1 - (.128G)]$ = 16,166 [1-(.128 × 1.02)] = 14,055 Lbs.
17. W_{rf}/Sk_r = 14,055 ÷ 20,928 = 0.672 (line 16/line 7)

Record Non-Dimensional Factors from Figures 3 through 7:

18. F_1/Sk_r = 0.68 (Figure 4.2) (line 9 to line 8 to answer)
19. F_2/Sk_r = 0.245 (Figure 4.3) (line 9 to line 8 to answer)
20. $2T/S^2k_r$ = 0.44 (Figure 4.4) (line 9 to line 8 to answer)
21. F_3/Sk_r = 0.43 (Figure 4.5) (line 9 to line 8 to answer)
22. T_a = Torque Adjustment for Peak Torque for Values of W_{rf}/Sk_r other than 0.3

   a. % = 0.3 (Figure 4.6) (Intersection of lines 10 and 8 is %)

   b. T_a = 1.03 (Figure 10.8) (From % on Fig. 4.6 to W_{rf}/Sk_r from line 17 to T_a)

Solve for Operating Characteristics:

23. PPRL $= W_{rf} + [(F_1/Sk_r) \times Sk_r]$ = 14,055 (line 16) + [0.68 (line 18) × 20,928 (line 7)] = 28,286 Lbs.
24. MPRL $= W_{rf} - [(F_2/Sk_r) \times Sk_r]$ = 14,055 (line 16) - [0.245 (line 19) × 20,928 (line 7)] = 8,928 Lbs.
25. PT $= (2T/S^2k_r) \times Sk_r \times S/2 \times T_a$ = 0.44 (line 20) × 20,928 (line 7) × 72.5 (S/2) × 1.03 (line 22) = 687,631 Lb.In.
26. PRHP $= (F_3/Sk_r) \times Sk_r \times S \times N \times 2.53 \times 10^{-6}$ = 0.43 (line 21) × 20,928 (line 7) × 145 (S) × 10.22 (N) × 2.53×10^{-6} = 33.74
27. CBE $= 1.06 (W_{rf} + 1/2 F_o)$ = 1.06 × (14,055 (line 16) + 4,628 (line 5/2)) = 19,804 Lbs.
28. (PPRL - MPRL) × 100/PPRL = ________ % [(Line 23 - Line 24) × 100/Line 23]
29. Actual Rod Stress = 28,286 (line 23) ÷ 0.601 (area of top rod) = 47,065 psi
30. Allowable Rod Stress = 50,000 (From Figure 5.1) psi (From Oilwell Supply)
31. If line 29 is greater than line 30, Overload = ______ (Line 29) - ______ (Line 30) = ______ psi, and system should be redesigned.

16a. TV = $W_{rf} + F_o$ = 14,055 + 9,256 = 23,311 lb.

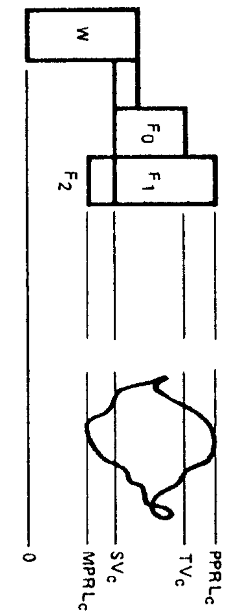
(Revised: 2-16-73)

TABLE 3.1, API SPEC. 1B

STANDARD GROOVE DIMENSIONS FOR V-BELT SHEAVES
(See Fig. 3.1)
All Dimensions In Inches, Except as shown

1	2	3	4	5	6	7	8	9	10	11
	Standard Groove Outside Diameter		Groove Angle (Degrees) α	Standard Groove Dimensions						
Cross Section	Minimum* Recommended	Range		b_g	h_g Min.	a	R_B Min.	d_B ±0.0005	S_g**	S_e
A	3.25	2.85 to & Incl. 5.65 Over 5.65	34 38	0.494 0.504 ±0.005	0.460	0.125	0.148 0.149	0.4375	0.625 ±0.025	0.375 +0.090 −0.062
A-B	5.75	3.75 to & Incl. 7.35 Over 7.35	34 38	0.612 0.625 ±0.006	0.612	A=0.375 B=0.175	0.230 0.226	0.5625	0.750 ±0.025	0.500 +0.120 −0.065
B	5.75	4.95 to & Incl. 7.35 Over 7.35	34 38	0.637 0.650 ±0.006	0.550	0.175	0.189 0.190	0.5625	0.750 ±0.025	0.500 +0.120 −0.065
C	9.40	7.4 to & Incl. 8.4 Over 8.4 to & Incl. 12.4 Over 12.4	34 36 38	0.879 0.887 0.895 ±0.007	0.750	0.200	0.274 0.276 0.277	0.7812	1.000 ±0.025	0.688 +0.160 −0.070
D	13.6	12.6 to & Incl. 13.6 Over 13.6 to & Incl. 17.6 Over 17.6	34 36 38	1.259 1.271 1.283 ±0.008	1.020	0.300	0.410 0.410 0.411	1.1250	1.438 ±0.025	0.875 +0.220 −0.080
E	21.8	18.8 to & Incl. 24.8 Over 24.8	36 38	1.527 1.542 ±0.010	1.270	0.400	0.476 0.477	1.3438	1.750 ±0.025	1.125 +0.280 −0.090
3V	2.65	Up through 3.49 Over 3.49 to and Incl. 6.00 Over 6.00 to & Incl. 12.00 Over 12.00	36 38 40 42	0.350 ±0.005	0.340	0.025	0.181 0.183 0.186 0.188	0.3438	0.406 ±0.015	0.344 +0.094 −0.031
5V	7.10	Up through 9.99 Over 9.99 to & Incl. 16.00 Over 16.00	38 40 42	0.600 ±0.005	0.590	0.050	0.329 0.332 0.336	0.5938	0.688 ±0.015	0.500 +0.125 −0.047
8V	12.50	Up through 15.99 Over 15.99 to & Incl. 22.40 Over 22.40	38 40 42	1.000 ±0.005	0.990	0.100	0.575 0.580 0.585	1.0000	1.125 ±0.015	0.750 +0.250 −0.062

*Below these diameters, horsepower ratings decrease sharply and result in less economical drives.
**See footnote under Table 3.2



OPTIMUM DESIGN DYNAMOMETER CARD
EXHIBIT 4

TABLE A.1, API SPEC. 1B

CLASSICAL V-BELT SHEAVE SIZES GENERALLY LISTED IN MANUFACTURERS' CATALOGS*

1	2	3	4	5	6	7	8	9	10	11	12	13
Combination A-B Section				B Section			C Section			D Section		
Diameter				Diameter			Diameter			Diameter		
Outside	Pitch Using A Section	Pitch Using B Section	Grooves	Outside	Pitch	Grooves	Outside	Pitch	Grooves	Outside	Pitch	Grooves
3.75	3.0	--	1 thru 6	20.35	20.0	2 thru 6,8,10	7.4	7.0	2 thru 6	12.6	12.0	4 thru 6, 8,10,12
3.95	3.2	--	1 thru 6									
4.15	3.4	--	1 thru 6	25.35	25.0	2 thru 6,8,10	7.9	7.5	2 thru 6	13.6	13.0	4 thru 6, 8,10,12
4.35	3.6	--	1 thru 6									
4.55	3.8	--	1 thru 6	30.35	30.0	2 thru 6,8,10	8.4	8.0	2 thru 6 8,10	14.1	13.5	4 thru 6, 8,10,12
4.75	4.0	--	1 thru 6									
4.95	4.2	--	1 thru 6	38.35	38.0	2 thru 6,8,10	8.9	8.5	2 thru 6, 8,10	14.6	14.0	4 thru 6, 8,10,12
5.15	4.4	4.8	1 thru 6									
5.35	4.6	5.0	1 thru 6				9.4	9.0	2 thru 6, 8,10,12	15.1	14.5	4 thru 6, 8,10,12
5.55	4.8	5.2	1 thru 6									
5.75	5.0	5.4	1 thru 6				9.9	9.5	2 thru 6, 8,10,12	15.6	15.0	4 thru 6, 8,10,12
5.95	5.2	5.6	1 thru 6									
6.15	5.4	5.8	1 thru 6				10.4	10.0	2 thru 6, 8,10,12	16.1	15.5	4 thru 6, 8,10,12
6.35	5.6	6.0	1 thru 6									
6.55	5.8	6.2	1 thru 6				10.9	10.5	2 thru 6, 8,10,12	16.6	16.0	4 thru 6, 8,10,12
6.75	6.0	6.4	1 thru 6									
6.95	6.2	6.6	1 thru 6				11.4	11.0	2 thru 6, 8,10,12	18.6	18.0	4 thru 6, 8,10,12
7.15	6.4	6.8	1 thru 6									
7.75	7.0	7.4	1 thru 6				12.4	12.0	2 thru 6, 8,10,12	20.6	20.0	4 thru 6, 8,10,12
8.95	8.2	8.6	1 thru 6									
9.75	9.0	9.4	1 thru 6				13.4	13.0	2 thru 6, 8,10,12	22.6	22.0	4 thru 6, 8,10,12
11.35	10.6	11.0	1 thru 6									
12.75	12.0	12.4	1 thru 6				14.4	14.0	2 thru 6, 8,10,12	27.6	27.0	4 thru 6, 8,10,12
15.75	15.0	15.4	1 thru 6									
18.75	18.0	18.4	1 thru 6				16.4	16.0	2 thru 6, 8,10,12	33.6	33.0	4 thru 6, 8,10,12
							18.4	18.0	2 thru 6, 8,10,12	40.6	40.0	4 thru 6, 8,10,12
							20.4	20.0	2 thru 6, 8,10,12	48.6	48.0	5,6,8,10, 12
							24.4	24.0	2 thru 6, 8,10,12	58.6	58.0	5,6,8,10, 12
							27.4	27.0	2 thru 6, 8			
							30.4	30.0	2 thru 6, 8,10,12			
							36.4	36.0	3 thru 6, 8,10,12			
							44.4	44.0	3 thru 6, 8,10,12			
							50.4	50.0	3 thru 6, 8,10,12			

*NOTE: *This information is shown here as an aid to the drive designer. It does not constitute a rigid standard, and is not intended to preclude future additions or deletions of sheave sizes.*

PUMP DESIGNATION

2.1. The basic types of pumps and letter designation covered by this specification are as follows:

Type of Pump	Letter Designation: Metal Plunger Pumps, Heavy-Wall Barrel	Metal Plunger Pumps, Thin-Wall Barrel	Soft-packed Plunger Pumps, Heavy-Wall Barrel	Soft-packed Plunger Pumps, Thin-Wall Barrel
Rod Pumps				
Stationary Barrel, Top Anchor	RHA	RWA	....	RSA
Stationary Barrel, Bottom Anchor	RHB	RWB	....	RSB
Traveling Barrel, Bottom Anchor	RHT	RWT	....	RST
Tubing Pumps	TH	....	TP	....

2.2 Complete pump designations include: (1) nominal tubing size, (2) basic bore diameter, (3) type of pump, including type of barrel and location and type of seating assembly, (4) barrel length, (5) plunger length, and (6) total length of extensions when used, as follows:

XX - XXX X X X X X - X - X

- Total length of extensions, whole feet.
- Nominal plunger length, feet.
- Barrel length, feet.
- **Type seating assembly;** C — Cup type
  M — Mechanical type
- **Location of seating assembly;** A — Top
  B — Bottom
  T — Bottom, traveling barrel
- **Type barrel;** H — Heavy-wall, W — Thin-wall: **For metal plunger pumps**
  S — Thin-wall, P — Heavy-wall: **For soft-packed plunger pumps**
- **Type pump;** R — Rod
  T — Tubing
- **Pump bore (basic):**
  125 — 1¼ in. *(31.8 mm)*
  150 — 1½ in. *(38.1 mm)*
  175 — 1¾ in. *(44.5 mm)*
  178 — $1\frac{25}{32}$ in. *(45.2 mm)*
  200 — 2 in. *(50.8 mm)*
  225 — 2¼ in. *(57.2 mm)*
  250 — 2½ in. *(63.5 mm)*
  275 — 2¾ in. *(69.9 mm)*
- **Tubing size;**
  15 — 1.900 in. OD. *(48.3 mm)*
  20 — 2⅜ in. OD. *(60.3 mm)*
  25 — 2⅞ in. OD. *(73.0 mm)*
  30 — 3½ in. OD. *(88.9 mm)*

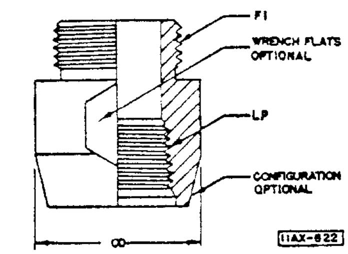
Example: A 1¼ in. *(31.8 mm)* bore rod type pump with a 10 ft. *(3.048 m)* heavy wall barrel and 2 ft. *(0.610 m)* of extensions, a 4 ft. *(1.219 m)* plunger, and a bottom cup type seating assembly for operation in 2⅜ in. *(60.3 mm)* tubing, would be designated as follows:

20-125 RHBC 10-4-2

2.3 In addition to the pump designation described in Par. 2.2, it is necessary for the purchaser to provide the following additional information:

a. Barrel material
b. Plunger material
c. Plunger clearance (fit)
d. Valve material
e. Length of each extension

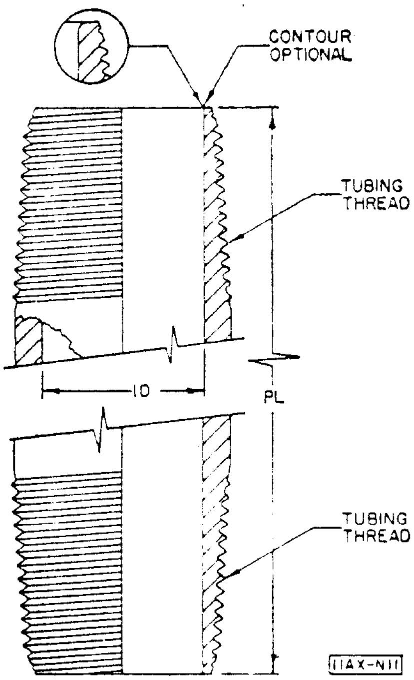
NOTE: *Metallic Materials for Subsurface Sucker Rod Pumps for Hydrogen Sulfide Environments* are listed in NACE Std MR-01-76.



B22, API SPEC. 11AX
All Dimensions in Inches (Followed by Equivalent in mm)
B22 — BUSHING, SEAT, BARREL CAGE

1	2	3	4	5
Dimensional Symbol	Part Number			
	B22-15	B22-20	B22-25	B22-30
FI	1.2500-14 *(31.750-14)*	1.4704-14 *(37.348-14)*	1.8024-14 *(45.781-14)*	2.1095-11½ *(53.581-11½)*
[1]LP	¾ nom.	1 nom.	1¼ nom.	1½ nom.
OD	1.438 max. *(36.53 max.)*	1.750 max. *(44.45 max.)*	2.250 max. *(57.15 max.)*	2.750 max. *(69.85 max.)*

[1]Line pipe thread. See Std 5B for details.

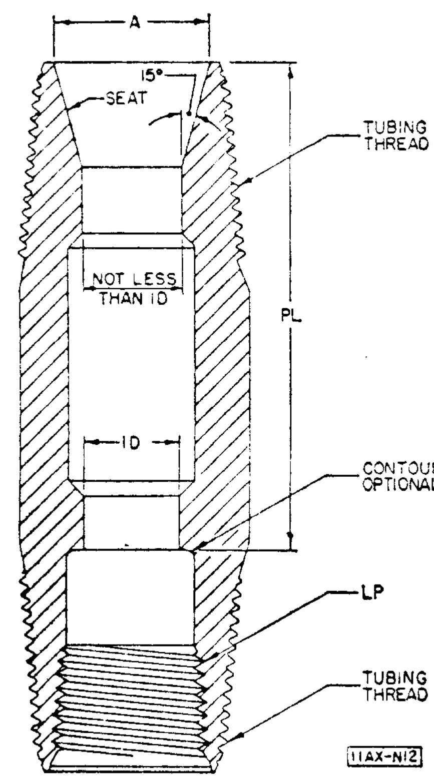


N11, API SPEC. 11AX
All Dimensions in Inches (Followed by Equivalent in mm)
N11 — NIPPLE, SEATING, CUP TYPE (ROD PUMP)

1	2	3	4	5
Dimensional Symbol	Part Number			
	N11-15	N11-20	N11-25	N11-30
[1]Tubing Thread	[2]1.900-10 IJ *(48.3-10 IJ)*	2⅜-8 EU *(60.3-8 EU)*	2⅞-8 EU *(73.0-8 EU)*	3½-8 EU *(88.9-8 EU)*
ID $^{+.010}_{-.000}$ *($^{+.25}_{-.00}$)*	1.460 *(37.08)*	1.780 *(45.21)*	2.280 *(57.91)*	2.780 *(70.61)*
PL min.	6 *(152.4)*	6 *(152.4)*	6 *(152.4)*	6 *(152.4)*

[1]See API Std 5B for tubing thread details.

[2]Upper connection may be 1.900-10 IJ *(48.3-10 IJ)* box thread, thus eliminating need for C34-15 coupling.



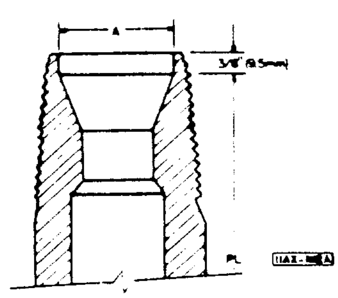
N12, API SPEC. 11AX
N12 — NIPPLE, SEATING, MECHANICAL BOTTOM LOCK

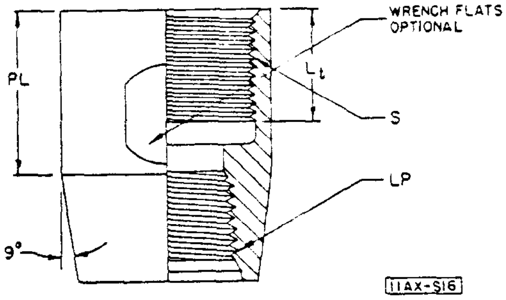
1	2	3	4	5
Dimensional Symbol	Part Number			
	N12-15	N12-20	N12-25	N12-30
[1]Tubing Thread	[3]1.900-10 IJ *(48.3-10 IJ)*	2⅜-8 EU *(60.3-8 EU)*	2⅞-8 EU *(73.0-8 EU)*	3½-8 EU *(88.9-8 EU)*
A	1.475 *(37.47)*	1.688 *(42.88)*	2.188 *(55.58)*	2.688 *(68.28)*
ID	1.125 *(28.58)*	1.375 *(34.93)*	1.750 *(44.45)*	2.250 *(57.15)*
PL $^{+.000}_{-.016}$ *($^{+.00}_{-.41}$)*	3.656 *(92.86)*	4.352 *(110.54)*	5.102 *(129.59)*	6.164 *(156.75)*
[2]LP nom.	1	1½	2	2½

[1]See API Std 5B for tubing thread details.

[2]Line pipe threads. See Std 5B for details.

[3]Upper connection may be 1.900-10 IJ *(48.3-10 IJ)* box thread, thus eliminating need for C34-15 coupling.





S16, API SPEC. 11AX
S16 — SEATING CUP COUPLING, BOTTOM ANCHOR

1	2	3	4	5
Dimensional Symbol	Part Number			
	S16-15	S16-20	S16-25	S16-30
S	1.1894-14 (30.211-14)	1.1894-14 (30.211-14)	1.5604-14 (39.634-14)	2.0035-11½ (50.889-11½)
*LP nom.	¾	1	1¼	1½
L_t min.	1⅛ (28.6)	1⅛ (28.6)	1¼ (31.8)	1¼ (31.8)
PL	....	2 3/16 (55.6)	2¼ (57.2)	1 15/16 (49.2)

*Line pipe thread. See API Std 5B for details.

TABLE 3.1, API SPEC. 11B
GENERAL DIMENSIONS AND TOLERANCES FOR SUCKER RODS AND PONY RODS
All dimensions in inches except rod lengths which are in feet. See Fig. 3.1.

1	2	3	4	5	6	7	8	9	10
Size of Rod	Nominal Diameter of Pin	Outside Diameter of Pin Shoulder and Box D_f	Width of Wrench Square $\pm\frac{1}{32}$ W_s	Length of Wrench Square[1] W_l	Diameter of Bead D_u	Total Length of Rod Box, min. L_b	Length of Box-and-Pin Rod[2] ±2.0 in.	Length of Box-and-Pin Rod[3] ±2.0 in.	Length of Box-and-Pin and Pin-and-Pin Pony Rods[2, 3] ±2.0 in.
½	¾	$1.000^{+0.005}_{-0.010}$	⅝	¾		—	—	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12
⅝	15/16	[4]$1.250^{+0.005}_{-0.010}$	⅞	1¼	Not	2⅛	25	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12
¾	1 1/16	$1.500^{+0.005}_{-0.010}$	1	1¼	to	2⅜	25	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12
⅞	1 3/16	$1.625^{+0.005}_{-0.010}$	1	1¼	Exceed	2⅜	....	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12
1	1⅜	$2.000^{+0.005}_{-0.010}$	1 5/16	1½	D_f	3	....	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12
1⅛	1 9/16	2.250 ± 0.015	1½	1⅝		3¼	....	25, 30	1⅓, 2, 3, 4, 6, 8, 10, 12

[1]Minimum length exclusive of fillet.
[2]The length of box-and-pin rods shall be measured from contact face of pin shoulder to contact face of box.
[3]The length of pin-and-pin rods shall be measured from contact face of pin shoulder to contact face on the field end of the coupling.
[4]**Dimension D_f of ⅝ in. box-and-pin rods shall be 1.375 ±0.015.**

TABLE 4.1, API SPEC. 11B
FULLSIZE COUPLINGS AND SUBCOUPLINGS
(All dimensions in Inches, See Fig. 4.1)

1	2	3	4	5	6
Nominal Coupling Size*	Outside Diameter W	Length Min. N_L	Length of Wrench Flat W_L**	Dist. Between Wrench Flats $0-\frac{1}{32}$ W_f	Used With Min. Tubing Size
⅝	1½	4	1¼	1⅜	2 1/16 OD
¾	1⅝	4	1¼	1½	2⅜ OD
⅞	1 13/16	4	1¼	1⅝	2⅞ OD
1	2 3/16	4	1½	1⅞	3½ OD
1⅛	2⅜	4½	1⅝	2⅛	3½ OD

*Also size of rod with which coupling is to be used.
**Minimum length exclusive of fillets.

TABLE 4.2, API SPEC. 11B

SLIMHOLE COUPLINGS AND SUBCOUPLINGS

(All dimensions in Inches, See Fig. 4.1)

1	2	3	4
Nominal Coupling Size*	Outside Diameter .005 - .010 *W*	Length Min. N_L	Used With Min. Tubing Size
½	1	2¾	1.660 OD
⅝	1¼	4	1.990 OD
¾	1½	4	2 1/16 OD
⅞	1⅝	4	2⅜ OD
1	2	4	2⅞ OD

*Also size of rod with which coupling is to be used.

TABLE 12.1, ASI SPEC. 11B

(See API Spec. 11B for polished-rod thread details.)

1	2	3	4
Polished-Rod Size (OD), in.	[1]Length, ft.	Thread Size (Nominal Pin Dia., in.)	Size Sucker Rod with which used
1	8, 11, 16	¾	½
[2]1⅛	8, 11, 16, 22	15/16, 1 1/16	⅝, ¾
[2]1¼	11, 16, 22	1 3/16	⅞
1½	16, 22	1⅜	1
[3]1½ (upset)	16, 22	1 9/16	1⅛

[1]Polished rods in lengths greater than 22 ft may be furnished by agreement between purchaser and manufacturer.

[2]1⅛ and 1¼ in. polished rods may be furnished with an upset on one end if so specified on the purchase order.

[3]The upset on 1½ in. polished rods to be made on one end only with a shoulder diameter equal to dimension D_f (2.250±.015 in.) in accordance with Spec 11B, and the length of this shoulder parallel to the body of the rod shall be ½ in. minimum.

TABLE 12.2, API SPEC. 11B

POLISHED-ROD LINERS

1	2	3
Liner Size, (OD), in.	*Threaded End Connection (UN-Class 2A)	Size Polished Rod with which used (OD), in.
1⅜	1⅜ — 16	1⅛
1½	1½ — 16	1¼
1¾	1¾ — 16	1½

***See Handbook H28, Screw-Thread Standards for Federal Service; obtainable from Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.**

TABLE 4.1, API RP 11BR

SUCKER ROD JOINT CIRCUMFERENTIAL DISPLACEMENT VALUES

All dimension in inches

1	2		3	
	Running New Grade D Displacement Values		Rerunning Grades C, D, & K Displacement Values	
Rod Size	Minimum	Maximum	Minimum	Maximum
½	6/32	8/32	4/32	6/32
⅝	8/32	9/32	6/32	8/32
¾	9/32	11/32	7/32	17/64
⅞	11/32	12/32	9/32	23/64
1	14/32	16/32	12/32	14/32
1⅛	18/32	21/32	16/32	19/32

NOTE: *Above displacement values were established through calculations and strain gage tests.*

TABLE 2.2, SUPPLEMENT 1, API Std 11E
PUMPING UNIT SIZE RATINGS

1	2	3	4
Pumping Unit Size	Reducer Rating, in.-lb	Structure Capacity, lb	Max. Stroke Length, in.
6.4— 32— 16	6,400	3,200	16
6.4— 21— 24	6,400	2,100	24
10— 32— 24	10,000	3,200	24
10— 40— 20	10,000	4,000	20
16— 27— 30	16,000	2,700	30
16— 53— 30	16,000	5,300	30
25— 53— 30	25,000	5,300	30
25— 56— 36	25,000	5,600	36
25— 67— 36	25,000	6,700	36
40— 89— 36	40,000	8,900	36
40— 76— 42	40,000	7,600	42
40— 89— 42	40,000	8,900	42
40— 76— 48	40,000	7,600	48
57— 76— 42	57,000	7,600	42
57— 89— 42	57,000	8,900	42
57— 95— 48	57,000	9,500	48
57—109— 48	57,000	10,900	48
57— 76— 54	57,000	7,600	54
80—109— 48	80,000	10,900	48
80—133— 48	80,000	13,300	48
80—119— 54	80,000	11,900	54
80—133— 54	80,000	13,300	54
80—119— 64	80,000	11,900	64
114—133— 54	114,000	13,300	54
114—143— 64	114,000	14,300	64
114—173— 64	114,000	17,300	64
114—143— 74	114,000	14,300	74
114—119— 86	114,000	11,900	86
160—173— 64	160,000	17,300	64
160—143— 74	160,000	14,300	74
160—173— 74	160,000	17,300	74
160—200— 74	160,000	20,000	74
160—173— 86	160,000	17,300	86
228—173— 74	228,000	17,300	74
228—200— 74	228,000	20,000	74
228—213— 86	228,000	21,300	86
228—246— 86	228,000	24,600	86
228—173—100	228,000	17,300	100
228—213—120	228,000	21,300	120
320—213— 86	320,000	21,300	86
320—256—100	320,000	25,600	100
320—305—100	320,000	30,500	100
320—213—120	320,000	21,300	120
320—256—120	320,000	25,600	120
320—256—144	320,000	25,600	144
456—256—120	456,000	25,600	120
456—305—120	456,000	30,500	120
456—365—120	456,000	36,500	120
456—256—144	456,000	25,600	144
456—305—144	456,000	30,500	144
456—305—168	456,000	30,500	168
640—305—120	640,000	30,500	120
640—256—144	640,000	25,600	144
640—305—144	640,000	30,500	144
640—365—144	640,000	36,500	144
640—305—168	640,000	30,500	168
640—305—192	640,000	30,500	192
912—427—144	912,000	42,700	144
912—305—168	912,000	30,500	168
912—365—168	912,000	36,500	168
912—305—192	912,000	30,500	192
912—427—192	912,000	42,700	192
912—470—240	912,000	47,000	240
912—427—216	912,000	42,700	216
1280—427—168	1,280,000	42,700	168
1280—427—192	1,280,000	42,700	192
1280—427—216	1,280,000	42,700	216
1280—470—240	1,280,000	47,000	240
1280—470—300	1,280,000	47,000	300
1824—427—192	1,824,000	42,700	192
1824—427—216	1,824,000	42,700	216
1824—470—240	1,824,000	47,000	240
1824—470—300	1,824,000	47,000	300
2560—470—240	2,560,000	47,000	240
2560—470—300	2,560,000	47,000	300
3648—470—240	3,648,000	47,000	240
3648—470—300	3,648,000	47,000	300

TABLE 4.1, API RP 11L
ROD AND PUMP DATA
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1⅛	1	⅞	¾	⅝	½
44	All	0.726	1.990×10^{-6}	1.000						100.0
54	1.06	0.908	1.668×10^{-6}	1.138					44.6	55.4
54	1.25	0.929	1.633×10^{-6}	1.140					49.5	50.5
54	1.50	0.957	1.584×10^{-6}	1.137					56.4	43.6
54	1.75	0.990	1.525×10^{-6}	1.122					64.6	35.4
54	2.00	1.027	1.460×10^{-6}	1.095					73.7	26.3
54	2.25	1.067	1.391×10^{-6}	1.061					83.4	16.6
54	2.50	1.108	1.318×10^{-6}	1.023					93.5	6.5
55	All	1.135	1.270×10^{-6}	1.000					100.0	
64	1.06	1.164	1.382×10^{-6}	1.229				33.3	33.1	33.5
64	1.25	1.211	1.319×10^{-6}	1.215				37.2	35.9	26.9
64	1.50	1.275	1.232×10^{-6}	1.184				42.3	40.4	17.3
64	1.75	1.341	1.141×10^{-6}	1.145				47.4	45.2	7.4
65	1.06	1.307	1.138×10^{-6}	1.098				34.4	65.6	
65	1.25	1.321	1.127×10^{-6}	1.104				37.3	62.7	
65	1.50	1.343	1.110×10^{-6}	1.110				41.8	58.2	
65	1.75	1.369	1.090×10^{-6}	1.114				46.9	53.1	
65	2.00	1.394	1.070×10^{-6}	1.114				52.0	48.0	
65	2.25	1.426	1.045×10^{-6}	1.110				58.4	41.6	
65	2.50	1.460	1.018×10^{-6}	1.099				65.2	34.8	
65	2.75	1.497	0.990×10^{-6}	1.082				72.5	27.5	
65	3.25	1.574	0.920×10^{-6}	1.037				88.1	11.9	
66	All	1.634	0.883×10^{-6}	1.000				100.0		
75	1.06	1.566	0.997×10^{-6}	1.191			27.0	27.4	45.6	
75	1.25	1.604	0.973×10^{-6}	1.193			29.4	29.8	40.8	
75	1.50	1.664	0.935×10^{-6}	1.189			33.3	33.3	33.3	
75	1.75	1.732	0.892×10^{-6}	1.174			37.8	37.0	25.1	
75	2.00	1.803	0.847×10^{-6}	1.151			42.4	41.3	16.3	
75	2.25	1.875	0.801×10^{-6}	1.121			46.9	45.8	7.2	
76	1.06	1.802	0.816×10^{-6}	1.072			28.5	71.5		
76	1.25	1.814	0.812×10^{-6}	1.077			30.6	69.4		
76	1.50	1.833	0.804×10^{-6}	1.082			33.8	66.2		
76	1.75	1.855	0.795×10^{-6}	1.088			37.5	62.5		
76	2.00	1.880	0.785×10^{-6}	1.093			41.7	58.3		
76	2.25	1.908	0.774×10^{-6}	1.096			46.5	53.5		
76	2.50	1.934	0.761×10^{-6}	1.097			50.8	49.2		
76	2.75	1.967	0.751×10^{-6}	1.094			56.5	43.5		
76	3.25	2.039	0.722×10^{-6}	1.078			68.7	31.3		
76	3.75	2.119	0.690×10^{-6}	1.047			82.3	17.7		
77	All	2.224	0.649×10^{-6}	1.000			100.0			
85	1.06	1.884	0.873×10^{-6}	1.261		22.2	22.4	22.4	33.0	
85	1.25	1.943	0.841×10^{-6}	1.253		23.9	24.2	24.3	27.6	
85	1.50	2.029	0.791×10^{-6}	1.232		26.7	27.4	26.8	19.2	
85	1.75	2.128	0.738×10^{-6}	1.201		29.6	30.4	29.5	10.5	

TABLE 4.1, API RP 11L (CONTINUED)
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1⅛	1	⅞	¾	⅝	½
86	1.06	2.058	0.742×10^{-6}	1.151		22.6	23.0	54.3		
86	1.25	2.087	0.732×10^{-6}	1.156		24.3	24.5	51.2		
86	1.50	2.133	0.717×10^{-6}	1.162		26.8	27.0	46.3		
86	1.75	2.185	0.699×10^{-6}	1.164		29.4	30.0	40.6		
86	2.00	2.247	0.679×10^{-6}	1.161		32.8	33.2	33.9		
86	2.25	2.315	0.656×10^{-6}	1.153		36.9	36.0	27.1		
86	2.50	2.385	0.623×10^{-6}	1.138		40.6	39.7	19.7		
86	2.75	2.455	0.610×10^{-6}	1.119		44.5	43.3	12.2		
87	1.06	2.390	0.612×10^{-6}	1.055		24.3	75.7			
87	1.25	2.399	0.610×10^{-6}	1.058		25.7	74.3			
87	1.50	2.413	0.607×10^{-6}	1.062		27.7	72.3			
87	1.75	2.430	0.603×10^{-6}	1.066		30.3	69.7			
87	2.00	2.450	0.598×10^{-6}	1.071		33.2	66.8			
87	2.25	2.472	0.594×10^{-6}	1.075		36.4	63.6			
87	2.50	2.496	0.588×10^{-6}	1.079		39.9	60.1			
87	2.75	2.523	0.582×10^{-6}	1.082		43.9	56.1			
87	3.25	2.575	0.570×10^{-6}	1.084		51.6	48.4			
87	3.75	2.641	0.556×10^{-6}	1.073		61.2	38.8			
87	4.75	2.793	0.522×10^{-6}	1.038		83.6	16.4			
88	All	2.904	0.497×10^{-6}	1.000		100.0				
96	1.06	2.382	0.670×10^{-6}	1.222	19.1	19.2	19.5	42.3		
96	1.25	2.435	0.655×10^{-6}	1.224	20.5	20.5	20.7	38.3		
96	1.50	2.511	0.633×10^{-6}	1.223	22.1	22.5	22.8	32.3		
96	1.75	2.607	0.606×10^{-6}	1.213	24.8	25.1	25.1	25.1		
96	2.00	2.703	0.578×10^{-6}	1.196	27.1	27.9	27.4	17.6		
96	2.25	2.806	0.549×10^{-6}	1.172	29.6	30.7	29.8	9.8		
97	1.06	2.645	0.568×10^{-6}	1.120	19.6	20.0	60.3			
97	1.25	2.670	0.563×10^{-6}	1.124	20.8	21.2	58.0			
97	1.50	2.707	0.556×10^{-6}	1.131	22.5	23.0	54.5			
97	1.75	2.751	0.548×10^{-6}	1.137	21.5	25.0	50.4			
97	2.00	2.801	0.538×10^{-6}	1.141	26.8	27.1	45.7			
97	2.25	2.856	0.528×10^{-6}	1.143	29.4	30.2	40.4			
97	2.50	2.921	0.515×10^{-6}	1.141	32.5	33.1	34.4			
97	2.75	2.989	0.503×10^{-6}	1.135	36.1	35.3	28.6			
97	3.25	3.132	0.475×10^{-6}	1.111	42.9	41.9	15.2			
98	1.06	3.068	0.475×10^{-6}	1.043	21.2	78.8				
98	1.25	3.076	0.474×10^{-6}	1.045	22.7	77.3				
98	1.50	3.089	0.472×10^{-6}	1.048	23.8	76.2				
98	1.75	3.103	0.470×10^{-6}	1.051	25.7	74.3				
98	2.00	3.118	0.468×10^{-6}	1.055	27.7	72.3				
98	2.25	3.137	0.465×10^{-6}	1.058	30.1	69.9				
98	2.50	3.157	0.463×10^{-6}	1.062	32.7	67.3				
98	2.75	3.180	0.460×10^{-6}	1.066	35.6	64.4				
98	3.25	3.231	0.453×10^{-6}	1.071	42.2	57.8				
98	3.75	3.289	0.445×10^{-6}	1.074	49.7	50.3				
98	4.75	3.412	0.428×10^{-6}	1.064	65.7	34.3				
99	All	3.676	0.393×10^{-6}	1.000	100.0					

TABLE 4.1, API RP 11L (CONTINUED)
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1¼	1⅛	1	⅞	¾	⅝
107	1.06	2.977	0.524×10^{-6}	1.184	16.9	16.8	17.1	49.1		
107	1.25	3.019	0.517×10^{-6}	1.189	17.9	17.8	18.0	46.3		
107	1.50	3.085	0.506×10^{-6}	1.195	19.4	19.2	19.5	41.9		
107	1.75	3.158	0.494×10^{-6}	1.197	21.0	21.0	21.2	36.9		
107	2.00	3.238	0.480×10^{-6}	1.195	22.7	22.8	23.1	31.4		
107	2.25	3.336	0.464×10^{-6}	1.187	25.0	25.0	25.0	25.0		
107	2.50	3.435	0.447×10^{-6}	1.174	26.9	27.7	27.1	18.2		
107	2.75	3.537	0.430×10^{-6}	1.156	29.1	30.2	29.3	11.3		
108	1.06	3.325	0.447×10^{-6}	1.097	17.3	17.8	64.9			
108	1.25	3.345	0.445×10^{-6}	1.101	18.1	18.6	63.2			
108	1.50	3.376	0.441×10^{-6}	1.106	19.4	19.9	60.7			
108	1.75	3.411	0.437×10^{-6}	1.111	20.9	21.4	57.7			
108	2.00	3.452	0.432×10^{-6}	1.117	22.6	23.0	54.3			
108	2.25	3.498	0.427×10^{-6}	1.121	24.5	25.0	50.5			
108	2.50	3.548	0.421×10^{-6}	1.124	26.5	27.2	46.3			
108	2.75	3.603	0.415×10^{-6}	1.126	28.7	29.6	41.6			
108	3.25	3.731	0.400×10^{-6}	1.123	34.6	33.9	31.6			
108	3.75	3.873	0.383×10^{-6}	1.108	40.6	39.5	19.9			
109	1.06	3.839	0.378×10^{-6}	1.035	18.9	81.1				
109	1.25	3.845	0.378×10^{-6}	1.036	19.6	80.4				
109	1.50	3.855	0.377×10^{-6}	1.038	20.7	79.3				
109	1.75	3.867	0.376×10^{-6}	1.040	22.1	77.9				
109	2.00	3.880	0.375×10^{-6}	1.043	23.7	76.3				
109	2.25	3.896	0.374×10^{-6}	1.046	25.4	74.6				
109	2.50	3.911	0.372×10^{-6}	1.048	27.2	72.8				
109	2.75	3.930	0.371×10^{-6}	1.051	29.4	70.6				
109	3.25	3.971	0.367×10^{-6}	1.057	34.2	65.8				
109	3.75	4.020	0.363×10^{-6}	1.063	39.9	60.1				
109	4.75	4.120	0.354×10^{-6}	1.066	51.5	48.5				

*Rod No. shown in first column refers to the largest and smallest rod size in eighths of an inch. For example, Rod No. 76 is a two-way taper of 7/8 and 6/8 rods. Rod No. 85 is a four-way taper of 8/8, 7/8, 6/8, and 5/8 rods. Rod No. 109 is a two-way taper of 1¼ and 1⅛ rods. Rod No. 77 is a straight string of 7/8 rods, etc.

TABLE 4.2, API RP 11L
TUBING DATA

1	2	3	4	5
Tubing Size	Outside Diameter, in.	Inside Diameter, in.	Metal Area, sq. in.	Elastic Constant, in. per lb ft E_t
1.900	1.900	1.610	0.800	0.500×10^{-6}
2⅜	2.375	1.995	1.304	0.307×10^{-6}
2⅞	2.875	2.441	1.812	0.221×10^{-6}
3½	3.500	2.992	2.590	0.154×10^{-6}
4	4.000	3.476	3.077	0.130×10^{-6}
4½	4.500	3.958	3.601	0.111×10^{-6}

TABLE 4.3, API RP 11L

SUCKER ROD DATA

1	2	3	4
Rod Size	Metal Area, Sq in.	Rod Weight in air, lb per ft W_r	Elastic Constant, in. per lb ft E_r
½	0.196	0.72	1.990×10^{-6}
⅝	0.307	1.13	1.270×10^{-6}
¾	0.442	1.63	0.883×10^{-6}
⅞	0.601	2.22	0.649×10^{-6}
1	0.785	2.90	0.497×10^{-6}
1⅛	0.994	3.67	0.393×10^{-6}

$\frac{F_1}{Sk_r}$, PEAK POLISHED ROD LOAD

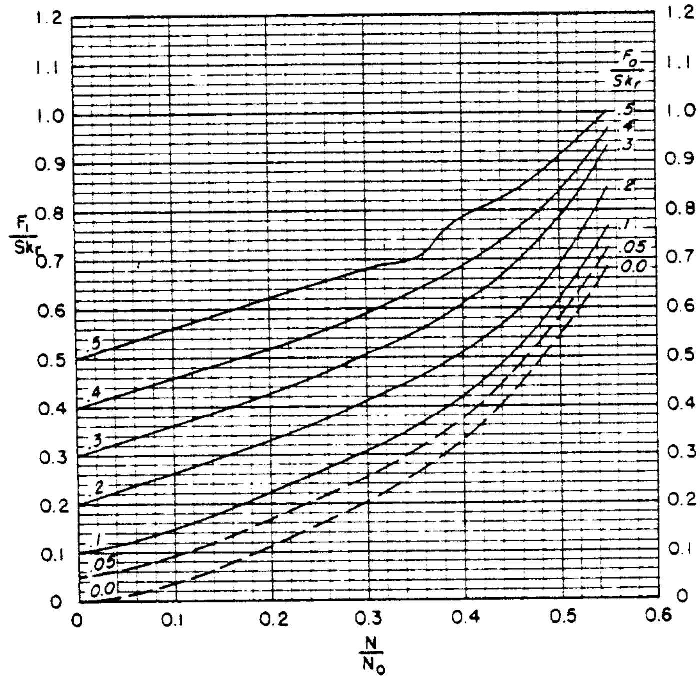


FIGURE 4.2, API RP 11L

$\frac{S_p}{S}$, PLUNGER STROKE FACTOR

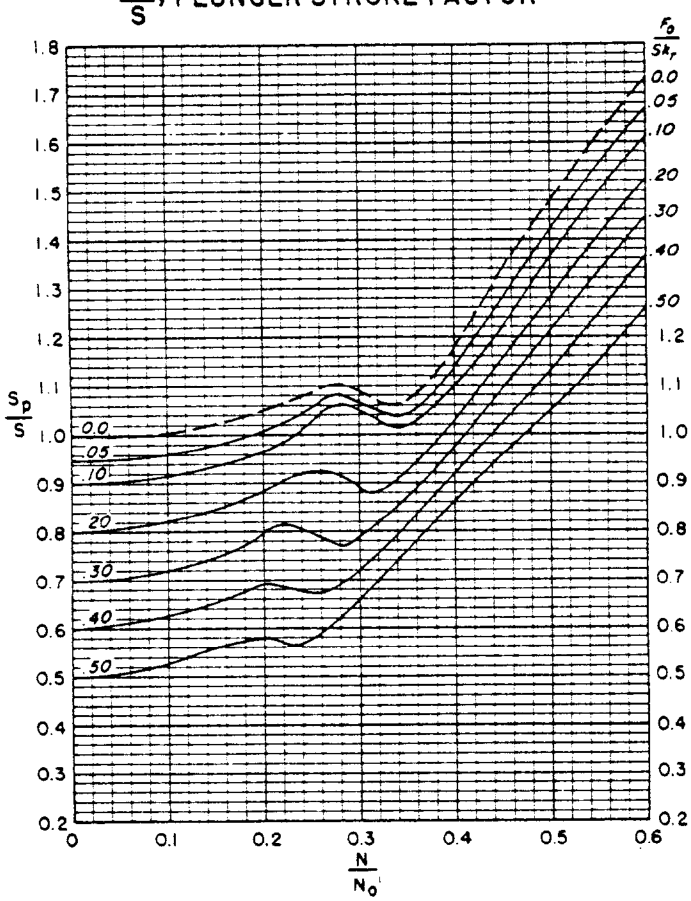


FIGURE 4.1, API RP 11L

$\frac{2T}{S^2 k_r}$, PEAK TORQUE

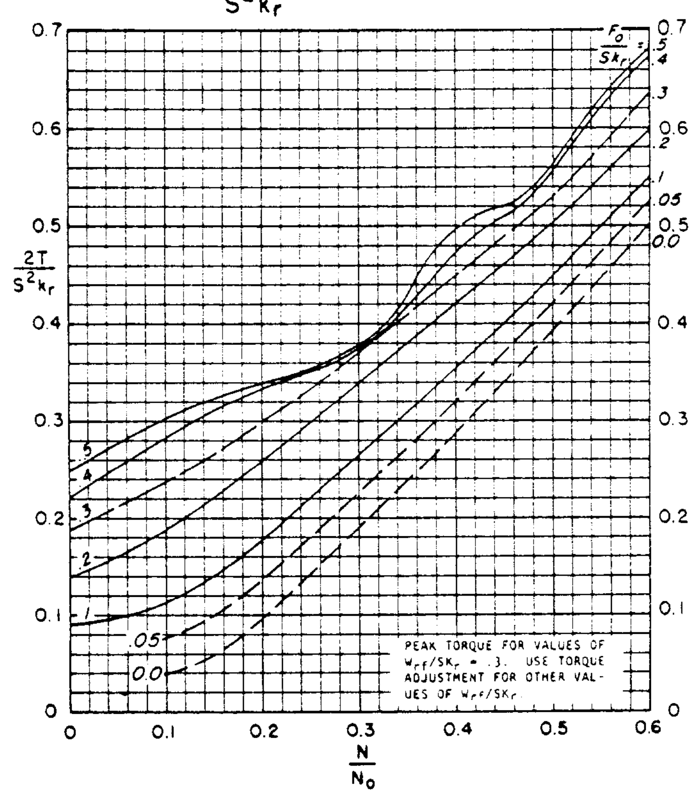


FIGURE 4.4, API RP 11L

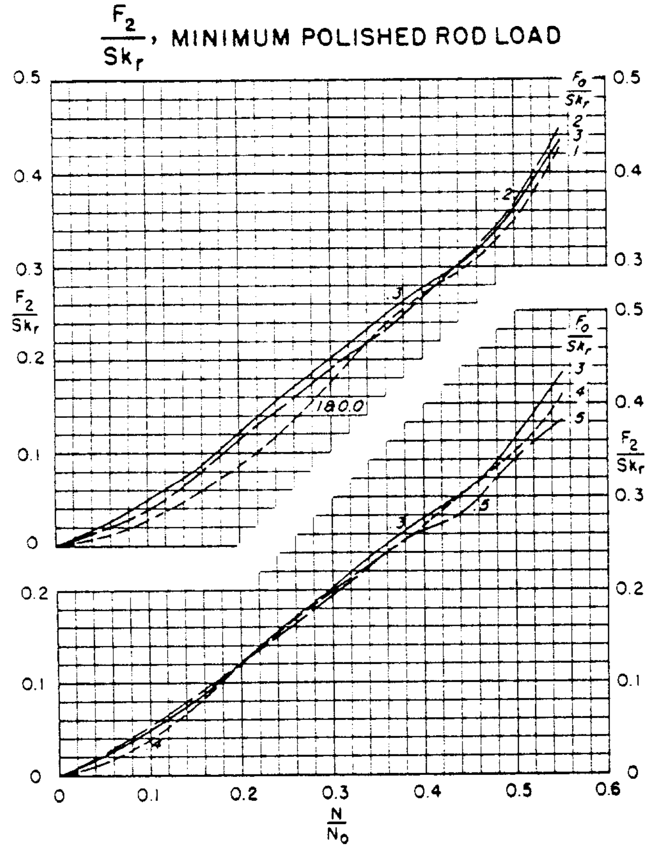


FIGURE 4.3, API RP 11L

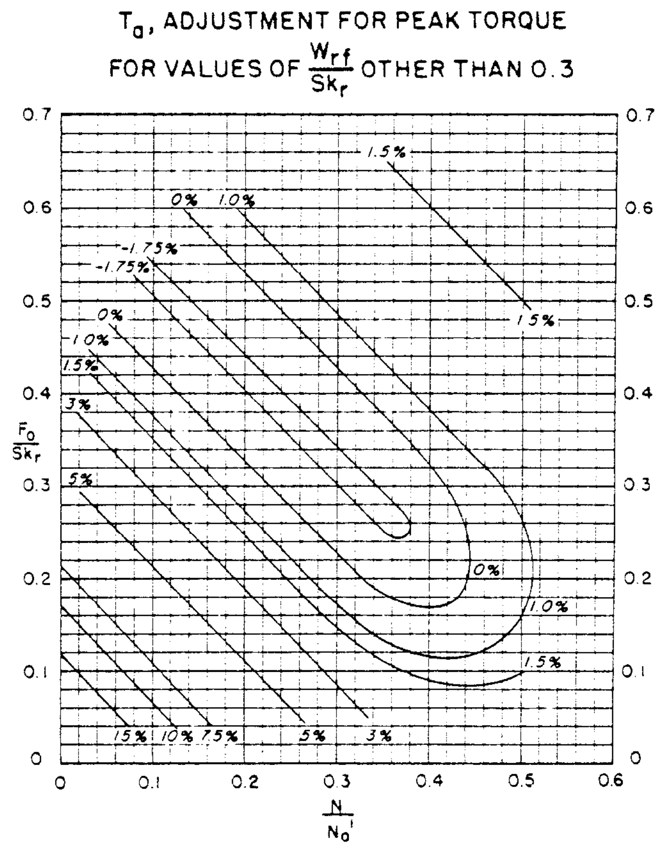


FIGURE 4.6, API RP 11L

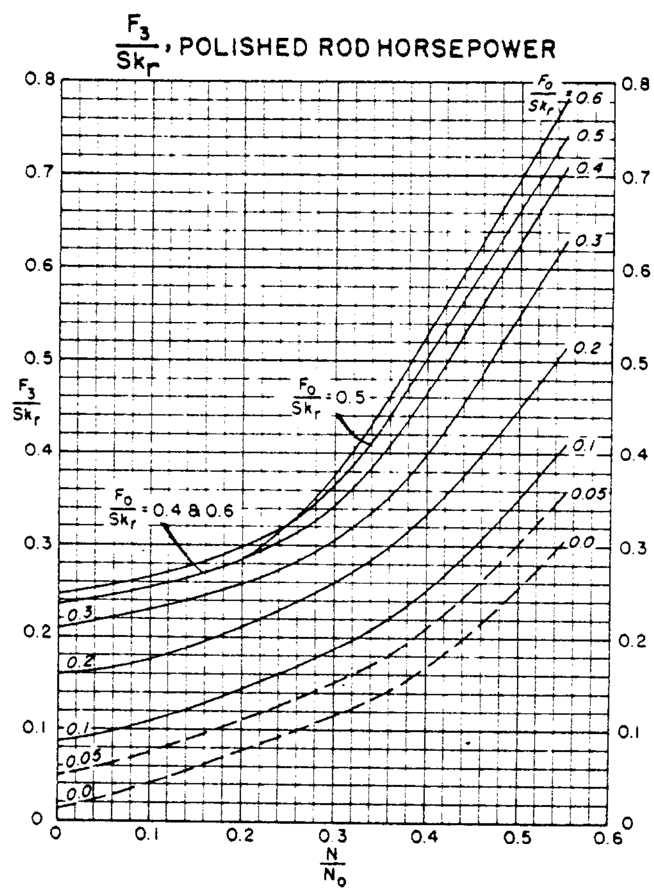


FIGURE 4.5, API RP 11L

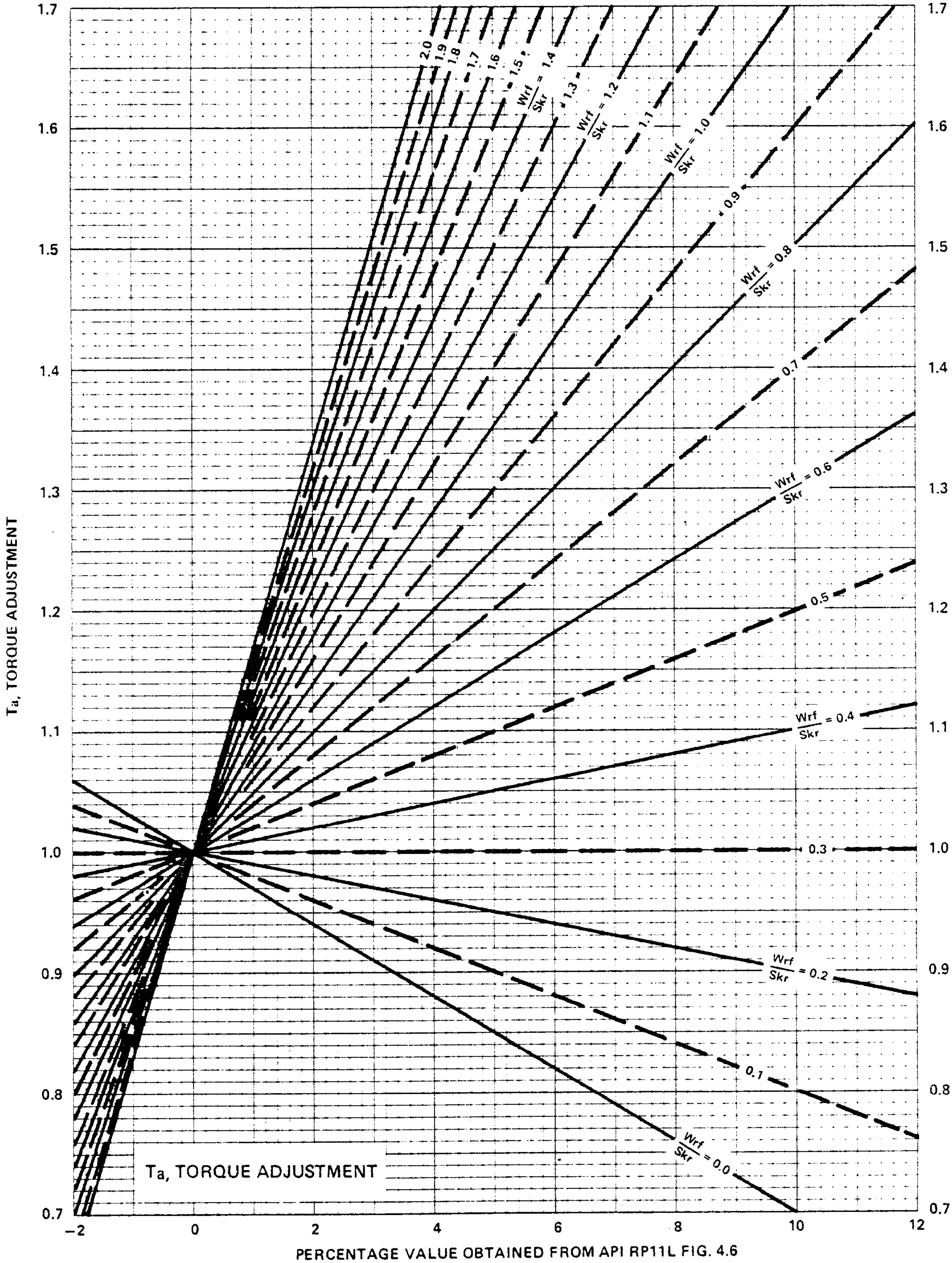


FIGURE 4.6A FOR API RP 11L