ROD PUMPING OVERVIEW

(Figure 1) shows the different parts of a sucker rod pumping system, including (from the top down) its five major components: the prime mover, which provides power to the system; the gear reducer, which reduces the speed of the prime mover to a suitable pumping speed; the pumping unit, which translates the rotating motion of the gear reducer and prime mover into a reciprocating motion; the sucker rod string, which is located inside the production tubing, and which transmits the reciprocating motion of the pumping unit to the subsurface pump; and the subsurface pump.

(Figure 2) is a cross-sectional representation of a subsurface pump at two different stages of the pump cycle. Note the location of the standing valve at the bottom of the tubing, and the traveling valve at the bottom of the sucker rods. Note also the changing position of the plunger.
On the left-hand side of the figure, the plunger is approaching the bottom of its downstroke. The traveling valve is open, and so the standing valve is closed because it is carrying the weight of the fluid above it. At this point in the cycle, the fluid above the standing valve is moving upward through the open traveling valve.

On the right-hand side of the figure, the plunger has reached the bottom of the stroke and is just beginning to move upward. The plunger starts to lift the weight of the fluid above it, and the traveling valve closes. As the plunger continues to move upward, the volume in the working barrel—between the standing valve and traveling valve—increases, while the pressure in the working barrel decreases. As soon as this pressure becomes less than the flowing bottomhole pressure, the standing valve opens and formation fluids flow upward. During each upward movement of the plunger, wellbore fluids are lifted a distance equal to one full stroke length.
When the plunger reaches the top of its stroke, its movement is reversed—the traveling valve opens, the standing valve closes and the cycle repeats its reciprocating movement of the rods and the opening and closing of the two valves. With each stroke, fluid is moved up the tubing toward the surface.

If the produced fluid contains free gas, there are two points to note regarding the pump cycle:

1. *The valves will not necessarily open and close at the exact top and bottom of the stroke.* The point in the upstroke at which the standing valve opens will depend on the spacing (the volume that exists at the bottom of the stroke, between the traveling and standing valves), and on the amount of free gas present in this volume. On the downstroke, the traveling valve remains closed until the pressure below the plunger is greater than the pressure above it. The traveling valve then opens and allows fluid to pass through it into the tubing. The exact point in the downstroke at which the traveling valve opens depends on the free gas volume in the fluid below the valve.

2. *The greater the volume of free gas, the greater the proportion of the stroke that is taken up in gas expansion and compression, without any true pumping action taking place.*

For wells producing a reasonable volume of gas, a gas anchor is normally installed on the tubing below the pump. This device allows the separated gas to be produced up the annulus before it would otherwise enter the pump.

**ADVANTAGES AND DISADVANTAGES OF ROD PUMPING SYSTEMS**

Rod pumping systems can be used to reduce bottomhole pressures to very low levels, and offer great flexibility for low-to-medium production rates. They are relatively simple with respect to design, operation and maintenance, and can be adapted to a wide range of operating conditions. They account for the large majority of artificial lift wells, and are one of the most well-known and generally understood systems in the field. Surface and downhole equipment can easily be refurbished, and tends to have high salvage values.

While they have a very wide range of applications, rod pumping systems are mainly limited to onshore locations due to the weight and space requirements of surface pumping units. Solids production, corrosion and paraffin tendencies, high gas-liquid ratios, wellbore deviation and depth limitations due to sucker rod capabilities have all been seen as problem areas for this lift method, although some of these problems have been alleviated due to improvements in sucker rod metallurgy, the development of long-stroke pumping units and other technical advances.
SURFACE EQUIPMENT

The primary surface equipment components of a rod pumping system are the prime mover, the gear reducer and the pumping unit (Figure 1: Courtesy Lufkin Industries, Inc).

PRIME MOVER

The prime mover, which may be either an internal-combustion engine or an electric motor, provides power to the pumping unit. The choice of prime mover for a particular well depends on the field conditions and type of power available.

INTERNAL-COMBUSTION ENGINES

Internal-combustion engines are most the most commonly used prime movers. They are classified as either slow-speed or high-speed engines.

- Slow-speed engines have one or two cylinders and usually operate at speeds of up to 750 RPM.
- High-speed engines have four or six cylinders; they can operate at speeds ranging from 750 to 2000 RPM, but are usually run at less than 1,400 RPM. Generally, high-speed engines have lower torque than slow-speed engines of comparable horsepower.

Engines can be further classified as two-cycle or four-cycle engines.
Most two-cycle engines are one-or two-cylinder, slow-speed engines. Two-cycle engines operate on natural gas, liquid petroleum (LP) gas, or diesel fuel.

The four-cycle engine is manufactured in either slow- or high-speed versions.

A slow-speed, four-cycle engine has a single horizontal cylinder and uses a large flywheel to provide a constant speed to the pumping unit. Some slow-speed, single-cylinder engines run on diesel or fuel oil, although these fuels are not often used if natural gas is readily available.

Four-cycle, high-speed engines used as prime movers can be fueled by natural gas, LP gas or gasoline.

Five factors determine which type of internal combustion engine to use:

**Available Fuel** - Natural gas is often supplied to the engine from the wellhead or from a field separator. This gas may require processing to remove oil, water and acid gas components such as H₂S or CO₂. Separator gas is considered the best fuel source, although LPG, diesel fuel and light-gravity crude oil have also been used.

**Equipment Life and Cost** - Slow-speed engines, although they may cost more initially, tend to have lower maintenance costs and longer operating lives than high-speed engines.

**Engine Safety Controls** - Because these engines run unattended, they must have reliable safety controls. Most safety controls stop the engine in cases of high water temperature, low oil pressure, overspeed, or pumping unit vibration (in case of sucker rod breaks). The engines are usually stopped by grounding the magneto and shutting off the fuel.

**Horsepower** - Power requirements are dictated by factors such as the size of the pumping unit, the depth of the well and the fluid gravity. API publication *API 7B/11C* describes procedures for the testing and rating of engines.

**Installation** - Slow-speed engines must be set on heavy concrete foundations. Multi-cylinder or vertical engines can be set on much lighter foundations.

**ELECTRIC MOTORS**

Induction electric motors are also used to drive pumping units. Horsepower ratings range from 1 to 200 HP, but most motors on pumping units operate at 10 to 75 HP. These are most often three-phase motors.

In fields where three-phase power is not available, single-phase AC motors may be used to meet power requirements of up to 10 HP on shallow, low-volume “stripper” wells. However, these motors cost more and operate less efficiently than three-phase motors having similar ratings. Direct current (DC) motors may also be used to drive pumping units, but they are not often chosen due to their higher initial and maintenance costs, their inability to use electricity from utilities and the fact that DC voltage cannot be changed by transformers, making transmission and distribution difficult.

**Motor Selection**

Oversized electric motors are often installed to prevent under-powering of pumping equipment. This may reduce motor failures, but it also increases capital costs and power consumption.

When a new pumping unit is installed on a well, motor sizing is based on preliminary estimates of the depth and size of the pump, speed of the pumping unit, and fluid properties. Actual motor requirements are determined by variables such as fluid level, viscosity, well deviation, and quality of electric power, as well as friction in the pump, stuffing box, and pumping unit. As a result, it is difficult to accurately size a motor for a new installation on the first attempt. Sometimes multiple-rated motors are considered for pumping unit drivers because they can be modified to operate at different horsepower settings.

**Motor Controls**

Motors used for pumping units have two types of control devices: one that stops, starts, and controls the motor, and the other that protects the motor. Common control devices include:
• The **Hand-Off-Auto Switch**, which is used to shut off the motor or operate it in automatic or manual mode.
  o In the hand position, the motor runs continuously, but does not override protection controls.
  o In the off position, the motor stops and cannot start, but power is not shut off.
  o In the automatic position, a programmer or time clock controls the motor. Pump-off controls usually function only when the switch is in auto position.

• The **Line Disconnect Switch**, which cuts all electrical power to the motor and is used when maintenance is performed on the pumping unit.

• The **Sequence-Restart Timer**, which delays the simultaneous restarting of several motors. If a power failure occurs in the field, a severe voltage drop can result as many motors connected to the same power source try to simultaneously restart. This condition could potentially keep the motors from restarting or cause the unintended operation of other field safety control devices.

• The **Programmer**, which controls the running times of pumping units. If the pumping unit operates continuously, the pump may not fill completely on the upstroke. This “pumped-off” condition will cause fluid pounding as the pump strikes the top of the fluid column on the downstroke. Such pounding causes shock loading of the sucker rods and pumping unit. Programmers optimize the pumping unit run time and reduce damage to the pumping system.

Devices that are designed to protect oilfield motors include:

• **Motor fuses**, which are located between the motor control and the electric power and limit damage to the motor from electrical or mechanical problems.

• **Air circuit breakers**, which are used instead of fuses to protect the electrical distribution system in case the motor develops an electrical problem.

• **Lightning arrestors**, which protect the system against storm damage caused by lightning.

• **Undervoltage relays, phase loss relays, thermal overload relays, and motor winding temperature sensors**, which are used to stop the motor before any permanent damage is done.

• **Pumping hit vibration switches**, which shut down the pumping unit motor if excessive vibration is detected. Such vibrations tend to occur when sucker rods break.

### Electrical Distribution System

The distribution system that supplies power to oil field electrical systems is made up of 3 main components: the primary and secondary electrical systems, and the grounding system.

The **primary system** supplies electricity to the field at voltages ranging from 4000 to 15000 V. High-voltage primary systems are preferred when electrical power travels a long distance. To protect these systems, it is necessary to install lightning arrestors and static lines that dissipate high static charges associated with electrical storms.

High voltages from the primary system are transformed to lower voltages (generally less than 600 V) through a **secondary system**. The secondary system includes the distribution transformers that convert the primary system voltage to the motor operating voltage and all cables, switches, controls, and other devices that operate at the motor voltage.

Electrical equipment at the well site must be properly **grounded** for personnel safety and to ensure that the electrical devices perform satisfactorily.
If utility-furnished power is not available, field-based generators can provide electrical power to operate pumping unit motors. Distribution equipment for the generator system would be the same as for utility power. The voltage requirements will depend on the size of the producing field.

- In a field with fewer than five wells located close to the generator, the generator voltage may be the same as the motor rated voltage.
- In a field with 5 to 50 wells, the generated voltage should be higher than the motor rated voltage. At each motor, a transformer would be used. This system would be considered a moderately sized system, with a generator having a distribution of 2,300 or 4,160 V.
- In a very large field with more than 50 wells, generated voltage of 2,300 or 4,160 V would be stepped up to 7,200 or 13,800 V for distribution. The higher voltage allows the use of smaller conductors and reduces line voltage drop. At each wellsite, a transformer will drop the distribution voltage to motor rated voltage.


**GEAR REDUCER**

The gear reducer is used to convert the high speed and low torque generated by the prime mover into the low speed and high torque required by the pumping unit. The Lufkin gear reducer shown in Figure 2 is of the double reduction type (*Figure 2: Double reduction gear reducer. Courtesy Lufkin Industries, Inc*). A high-speed gear (the smallest in diameter) is mounted on a shaft, which is connected to a sheave/belt assembly that is driven by the prime mover. Speed reduction occurs between this gear and a larger gear mounted on an intermediate shaft, and between the intermediate gear and a still larger gear mounted on the crankshaft that actually drives the pumping unit. The gears are continually lubricated by an oil reservoir contained within the assembly. When in place at the well, the gear train is mounted in an enclosed box.

**PUMPING UNIT**

The pumping unit changes the rotational motion of the prime mover to a reciprocating vertical motion. The unit is driven by the crankshaft on the gear reducer, and is connected to a polished rod and a sucker rod string, which drives the subsurface pump. Most pumping units employ a counterbalance (usually adjustable weights or pressurized air), which opposes the weight of the sucker rod string.
Pumping units are available in a variety of sizes and configurations. They are classified according to their methods of counterbalance and the ways in which their major components are arranged.

**CLASS I LEVER SYSTEMS**

Class I lever systems include the conventional crank-balanced, beam-balanced, and certain special geometry units. The fulcrum (Samson post bearing) of this type unit is located at the middle of the walking beam, between the well load and the actuating force (Figure 3: Class I lever system: conventional pumping unit. Courtesy Lufkin Industries, Inc).

**Conventional Crank-Balanced Units**

As the cranks on a conventional unit rotate, the pitman side members cause the walking beam to pivot on a center bearing, moving the polished rod. Adjustable counterweights are located on the cranks (Figure 4: Conventional crank balance pumping unit geometry).
This is the most common pumping unit type, because of its relative simplicity of operation, low maintenance requirements and adaptability to a wide range of field applications.

Beam-Balanced Units
On beam-balanced units, the counterweights are positioned at the end of the walking beam (Figure 5: Churchill® beam-balanced pumping unit. Courtesy Lufkin Industries, Inc). This type of unit is susceptible to damage at high pumping speeds, and so its use is limited to smaller sizes and low pumping speeds. Nonetheless, its simple design and dependability make it an attractive option for shallow wells.

Crank Balanced Units with Special Geometry
On some crank balanced units, the gear reducer is moved from directly under the equalizer to a position away from the centerline of the well. This difference from the conventional geometry changes the torque factors and the time intervals on the upstrokes and downstrokes. Units of this type usually have an out-of-phase counterbalance system and require a specific rotation direction.

CLASS III LEVER SYSTEMS
Class III lever systems include air-balanced and the Mark II crank counterbalanced units (Figure 6: Class III lever system: air-balanced unit. Courtesy of Lufkin Industries, Inc. The walking beam hinge point is at the rear of the unit and the actuating force is located between the pivot point and the well.
Air-Balanced Units

Air-balanced units are similar to crank-balanced units in that the rotation of the crank causes the walking beam to pivot and move the polished rod (Figure 7: Air-balanced pumping unit geometry).
A piston and air cylinder partially counterbalances the well load by compressing air in the cylinders. An auxiliary air compressor, controlled by a pressure switch, maintains the system air pressure (Figure 8: Air balance piston/cylinder assembly. Courtesy of Lufkin Industries, Inc.).

The piston/air cylinder assembly allows for more accurate control of the counterbalance than the use of counterweights, and reduces the weight of the unit. This results in lower installation and transportation costs. Air balanced units work to particular advantage on wells that require larger unit sizes or longer pump strokes, where space and weight requirements might preclude using a crank-balanced unit.

**Mark II Units**

On Mark-II units, the cross yoke bearing is located very close to the horsehead (Figure 9: Mark II pumping unit. Courtesy of Lufkin Industries, Inc.). The cranks, which rotate in only one direction, have an angular offset to produce an out-of-phase condition between the torque exerted by the well load and the torque exerted by the counterbalance weights. These features reduce problems associated with torque peaks that are more common with conventional crank balanced units.
LONG-STROKE PUMPING UNITS

For a given depth, pump size, production rate and rod string, a longer pump stroke and slower pumping speed tend to result in reduced power requirements and lower rod stresses than an equivalent system operating with a shorter pump stroke and a higher pumping speed (Tait and Hamilton, 1984). This observation is particularly worth noting in deep, high-volume wells that might normally call for an electrical submersible or hydraulic pump. Long-stroke pumping units (e.g., Weatherford International’s Rotaflex® unit) provide another option. Where conventional pumping units are available with maximum stroke lengths of 240 inches, these units have stroke lengths of up to 366 inches, operating at rates ranging from 1 to 5.2 strokes per minute (Lea, Winkler and Snyder, 2006). Under the right conditions, these units can be used to optimize lift efficiency, increase rod life and minimize power costs.

PUMPING UNIT SELECTION

The choice of pumping unit will be based on well parameters, operating conditions and, of course, cost and availability. A conventional crank-balanced unit might be chosen because field personnel are familiar with it, while the relatively small size, low weight and low inertial and shaking forces of an air-balanced unit would make it a good choice for an offshore site or other confined surface location.

PUMPING UNIT API DESIGNATIONS

Manufacturers design pumping units in standard sizes, and their catalogs feature pumping units that can handle many combinations of torque, polished rod load and stroke length. Pumping units are designated using a 10-character, alphanumeric code established by the API (Figure 10: API pumping unit designations). An example of a pumping unit designation would be C-140D-117-64.
The first character in the API code designates the pumping unit type.

1. C - Conventional
2. A - Air-balanced
3. B - Beam-balanced
4. M - MARK II
5. RM - Reverse Mark
6. LP - Low Profile
7. CM - Portable/Trailer Mount
8. LC - Power Lift

- The next four characters designate the peak torque rating, in thousands of inch-pounds, and the type of gear reducer. In most cases, a double reduction gear reducer is used, indicated by the letter “D”.
- The next three characters define the peak polished rod load rating, in hundreds of pounds.
- The last two characters are the stroke length, in inches.
SUBSURFACE EQUIPMENT

The subsurface components of a rod pumping system include the sucker rod string and the subsurface pump.

SUCCER RODS

The subsurface pump is connected to the pumping unit on the surface by a string of solid sucker rods (Figure 1). These rods come in either 25-foot or 30-foot lengths, and API-standard diameters of 1/2, 5/8, 3/4, 7/8, 1 and 1-1/8 inch.

Pony rods are shorter-length rods used to bring the rod string to the exact length needed in a well. Sucker rods are joined together by threaded connectors or couplings, which are usually about 4 inches long (Figure 2: Courtesy of Tenaris Oilfield Services, 2003 -- http://www.tenaris.com). Some rods are manufactured with pin-and-box couplings, while others are made with pin couplings at both ends and then joined together using rod couplings.

STEEL SUCKER RODS

One-piece steel sucker rods are manufactured from hot-rolled steel. The pin ends of the rod are forged, the rod is heat-treated, the pins are machined, and a finish designed to reduce corrosion is applied to the rod surface.

Three-piece rods are manufactured by attaching threaded metal pin and box connectors to the threaded ends of a machined rod. These sucker rods are sometimes used in shallow wells.

The choice of what size and API grade of steel sucker rods to use in a well is determined by rod stress and well conditions. Rod stress is determined by the production rate, tubing and pump sizes and the characteristics of the pumping unit.


The presence of H₂S, CO₂, salt water or other corrosive agents effectively reduce the maximum allowable stress on a rod string, and require adjustments to rod stress calculations.

FIBERGLASS SUCKER RODS

Fiberglass sucker rods are actually made of fiberglass embedded in a plastic matrix. Steel end fittings with standard API pins are attached using an adhesive. These rods are joined with standard couplings to form rod strings.

The primary reasons for using fiberglass rods is that they lighter and more corrosion-resistant than steel rods. This makes them an attractive option in deeper wells and corrosive environments. There are, however, several disadvantages to using fiberglass rods in place of steel rods. They are more susceptible to fatigue failures resulting from the repeated loading and unloading during the pumping cycle, and are more likely to fail under compressive forces. And they may lose their strength through exposure to high well temperatures or hot-oiling treatments. The resins used during manufacture of the rods are the elements that determine the rod’s resistance to chemicals and temperature; different resins have different physical properties and resistance to chemicals.

API standards for the manufacture and performance of fiberglass sucker rods are detailed in API Specification 11B (1998), which covers dimensional requirements for fiberglass sucker rods, couplings and subcouplings.

CONTINUOUS SUCKER RODS

Unlike conventional sucker rod strings, continuous sucker rods (e.g., COROD®, provided by Weatherford International Ltd) require couplings only at the top and bottom of the rod string - regardless of well depth. This continuous length of steel reduces the weight of the rod string and provides a more uniform contact with the production tubing—a particular advantage in deviated wells.

ROD FAILURES

Steel sucker rods usually break because of failures in the rod body, most often caused by corrosion resulting from exposure to H₂S, CO₂, salt water, or O₂. Corrosion inhibitors have been developed to help reduce these types of rod failures.

Body breaks in steel rods commonly occur near the upset. A bending moment placed on the rod in the area of the upset can weaken the rod. Sucker rods may also fail at the pin if too much torque is applied when the rod string is being made up. Extra care when making up or breaking down and handling sucker rods will reduce these types of failures.
Fiberglass rods usually fail at the end-connector joint where the metal end connector may separate from the rod, although breaks or separations of the rod body also occur. Fiberglass rods are more easily damaged than steel rods; surface scratches and ultraviolet light may weaken them.

API Spec.11BR (1989, reaffirmed October, 1993), covers recommendations on the storage, transportation, running and pulling of sucker rods.

**SUBSURFACE PUMPS**

The main components of a subsurface pump are:

- The *working barrel*, connected to the tubing.
- The *plunger*, connected to the sucker rods.
- The *traveling valve*, which is part of the plunger assembly.
- The *standing valve*, which is located at the bottom of the working barrel.

At the bottom of the pump, connected to the working barrel, there is usually a perforated *gas anchor*, which allows formation fluids to separate before entering the pump. It also directs much of the free gas into the casing-tubing annulus and improves pump efficiency.

(Figure 3) shows the two principal categories of subsurface pumps: the *tubing pump* on the left, and the *rod or insert pump*.

The main difference between these two pump types is in the installation of the working barrel.
- The working barrel of a tubing pump is an integral part of the tubing string. This is advantageous in that it provides for the strongest possible pump construction, and it allows the maximum plunger diameter to be only slightly less than the tubing diameter, thus maximizing the volume of fluid that can be pumped. The disadvantage of the tubing pump is that the entire tubing string must be pulled in order to service the working barrel and other pump hardware.
- The working barrel of a rod pump is run on sucker rods rather than on tubing. Thus, the pump can be serviced by pulling only the rods. However, its plunger diameter has to be smaller than that of a comparable tubing pump, and thus it cannot move as large a fluid volume.

**TUBING PUMPS**

Tubing pumps are classified according to the type of working barrel, the standing valve arrangement and the type of plunger used.

**Working Barrel**

The working barrel may be either a one-piece, tube-type barrel, or a liner contained within an outside jacket (Figure 4: Working barrels).

![Figure 4: Working barrels](image)

The one-piece working barrel, which may be heavy-walled or thin-walled, is made of cold-drawn, seamless steel, cast iron, or corrosion resistant alloys. The working barrel is polished on the inside to allow easier plunger movement.

The widely used *common working barrel* is a heavy-wall version of the one-piece barrel.

The liner-type working barrel consists of a liner surrounded by an outer steel jacket. The liner may consist of a single hardened steel liner or several short sectional liners held in place by clamping collars (Figure 3). The liner-type working barrel has greater precision and lower repair cost, but higher capital cost than the common working barrel.

**Standing Valve**
The second criterion for classifying a tubing pump is whether its standing valve is **fixed** or **removable**.

(Figure 5) shows a tubing pump with a fixed standing valve. In part (a), the standing valve is located at the bottom of the tubing string; in part (b), the plunger is pulling out and the standing valve is still located at the shoe of the tubing. To retrieve the standing valve, the tubing string must be pulled, as shown in part (c).

(Figure 6) shows a tubing pump with a removable standing valve. The standing valve is located at the shoe of the tubing when the pump is operating. To remove the valve (b), the plunger is lowered onto a special fitting on the standing valve, which is then brought to the surface along with the plunger and sucker rods. The barrel and tubing are left in place in the subsurface.
Plunger Type
A third criterion for classifying pumps is the makeup and composition of the plunger. There are two types of plungers: the cup-type (or soft pack) plunger, and the metal plunger.

- The cup-type plunger is the oldest form of seal used in pumping units. Early cups were usually made of leather or rubber-impregnated canvas; however, synthetic materials are now used for flexible plungers. On the upstroke, pressure exerted by the fluid column forces the cup to expand and form a seal between the lip of the cup and the wall of the barrel. On the downstroke, as pressure is equalized on both sides of the cup, it collapses inward to allow the plunger to fall freely. These types of plungers are not generally used below 5,000 ft.
- Metal plungers (Figure 7) are made of cast iron or steel and have either a smooth or a grooved sealing surface. Grooved plungers offer an advantage when the well produces sand. Sand particles can be trapped in the groves, preventing them from abrading the plunger. The metal-to-metal seal for these plungers depends upon an extremely close clearance. This type of plunger usually wears better than the cup-type and is used in deeper wells.

Note that it is possible to have both cup-type and ring-type plungers in a single pump.

Rod Pumps
The two main types of rod pumps are the stationary pump (Figure 8), which has a moving plunger and a stationary working barrel, and the traveling barrel pump (Figure 9), which has a moving working barrel and a stationary plunger.
Another less common type of rod pump is the *three-tube* pump. This pump uses an inner plunger and an outside barrel that telescope down in a concentric manner around a standing barrel, thus forming a long fluid seal between the barrels. Because this pump does not use mechanical seals, it performs well with abrasive fluids in sandy wells.

The *casing pump* (Figure 10) is a special type of rod pump in which a packer, placed at either the top or bottom of the working barrel, provides the fluid pack-off between the working barrel and the casing. No tubing is used. Casing pumps are generally used in large volume, shallow pumping applications.
API PUMP CLASSIFICATIONS

The API classification for subsurface pumps, set out in API Specification 11AX (2001), is based on the specification of three elements of the pump:

1. Whether it is a rod-type or tubing-type pump
2. Whether it has a stationary or traveling working barrel and the type of barrel and plunger installed
3. Whether it has a top or bottom seating assembly (for rod pumps with stationary barrels), or anchor (an anchor is a mechanism that keeps one section of the pump stationary so that the pump can operate properly. It is also referred to as a hold-down).

The API classification uses a three-letter code. Each letter refers to one of the three elements above. The full range of API pump classifications is shown in (Figure 11) (API pump classifications. After API Spec. 11AX):

- Pumps (a) and (b) in this figure are examples of stationary barrel, top hold-down pumps. Top hold-down pumps are recommended for use in sandy wells, because sand particles cannot settle over the seating nipple. This makes the pump easier to remove. These pumps are also
good for wells with low fluid levels, because the standing valve is submerged deeper than it
would be with a bottom hold-down pump. Pump (a) has the API classification RHA.
  o R means that it is a rod-type pump,
  o H signifies that it has a stationary, heavy-wall barrel, and
  o A is used to indicate a top anchor.

The top anchor refers to the fact that the standing valve is held in place from above. If the
stationary, heavy-wall barrel is changed to a stationary, liner barrel, the designation for the
pump changes to RLA. In this case, only the middle letter changes in our pump classification.

- Pump (b) is similar to the type of RHA pump. Again, it is a rod-type pump, but it has a
  stationary, thin-wall barrel and a top anchor. Its designation is RWA. If a soft-packed plunger
  is installed, then the designation changes to RSA.
- A rod-type pump with a stationary, heavy-wall barrel and a bottom anchor is an RHB pump
  (pump (c) in the figure). Note that for a bottom anchor pump, the standing valve is held from
  below and a (B) designation, rather than an (A) designation is used. If the barrel is changed to
  a stationary liner barrel, the designation changes to RLB.
- An RWB pump (pump (d)) has the same configuration, but with stationary, thin-wall barrel. If a
  soft-pack plunger is used in the pump, the designation is RSB.
- A traveling, heavy-wall barrel with a bottom anchor on a rod pump (pump (e)), is designated
  RHT. If the traveling barrel is a liner barrel, the designation is RLT.
- Pump (f) is the same type of pump, but with a thin-wall traveling barrel; its designation is
  RWT. If a soft plunger is added, the designation is RST.
- If, instead of a rod-type pump, a tubing pump with a heavy-wall barrel is used, then the
  designation is TH (pump (g)). (If a liner barrel replaces the heavy-wall barrel, the designation
  is TL.
- A tubing pump with a heavy-wall barrel and a soft-pack plunger is given the designation TP
  (pump (h) in Figure 12). Note that a metal plunger is used in situations where a specific
designation is not given.

NOTE: There is no particular need to try and memorize each of the sixteen designations just covered. It is
sufficient to know that the API classification exists and how to use it. It should be noted that, only three or
four of these designations are generally used.
Standard Designations for a Complete Pump

Building on the general classifications above, the API has developed standard designations for a complete pump (API Specification 11AX, 2001). The designation consists of a 12 character, alphanumeric code (XX-XX-XXX-X-X-X-X-X-X-X-X-X) as shown in Table 1 (after API 11AR, 2000):

<table>
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<th>TABLE 1: API Designation for Subsurface Pumps</th>
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<th>Designation/Description</th>
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<td>· 20 - 2.375 inch O.D.</td>
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<tr>
<td></td>
<td>· 25 - 2.875 inch O.D.</td>
</tr>
<tr>
<td></td>
<td>· 30 - 3.500 inch O.D.</td>
</tr>
<tr>
<td>XXX</td>
<td>Pump bore (basic):</td>
</tr>
<tr>
<td></td>
<td>· 106 - 1.0625 inch</td>
</tr>
<tr>
<td></td>
<td>· 125 - 1.25 inch</td>
</tr>
<tr>
<td></td>
<td>· 150 - 1.50 inch</td>
</tr>
<tr>
<td></td>
<td>· 175 - 1.75 inch</td>
</tr>
<tr>
<td></td>
<td>· 178 - 1.78125 inch</td>
</tr>
<tr>
<td></td>
<td>· 200 - 2.00 inch</td>
</tr>
<tr>
<td></td>
<td>· 225 - 2.25 inch</td>
</tr>
<tr>
<td></td>
<td>· 250 - 2.50 inch</td>
</tr>
<tr>
<td></td>
<td>· 275 - 2.75 inch</td>
</tr>
<tr>
<td>X</td>
<td>Type pump:</td>
</tr>
<tr>
<td></td>
<td>· R - Rod</td>
</tr>
<tr>
<td></td>
<td>· T - Tubing</td>
</tr>
<tr>
<td>X</td>
<td>Type of barrel:</td>
</tr>
<tr>
<td></td>
<td>(For metal plunger pumps)</td>
</tr>
<tr>
<td></td>
<td>· H - Heavy wall</td>
</tr>
<tr>
<td></td>
<td>· L - Liner barrel</td>
</tr>
<tr>
<td></td>
<td>· W - Thin wall</td>
</tr>
<tr>
<td></td>
<td>(For soft-packed plunger pumps)</td>
</tr>
<tr>
<td></td>
<td>· S - Thin wall</td>
</tr>
<tr>
<td></td>
<td>· P - Heavy wall</td>
</tr>
<tr>
<td>X</td>
<td>Location of seating assembly:</td>
</tr>
<tr>
<td></td>
<td>· A - Top</td>
</tr>
<tr>
<td></td>
<td>· B - Bottom</td>
</tr>
<tr>
<td></td>
<td>· T - Bottom, traveling barrel</td>
</tr>
</tbody>
</table>
To illustrate the use of the API designation, we may use the information from Table 1 to describe an API pump designation for a rod-type, stationary pump with a heavy-wall barrel, a metal plunger, and a top anchor using a mechanical-type seating assembly, to be set in 2.375-inch tubing, with a pump-bore size of 1.5 inches. The plunger-length is 4 feet and there are no extensions to be added.

- The first two characters refer to a tubing size code. The 1.9-inch O.D. tubing, for example, is code number 15; a 3 1/2-inch O.D. tubing is code number 30. In this example, with 2-3/8-inch O.D. tubing installed in the pump, so the code number “20” are the first characters of the pump designation.
- The next three characters refer to the code for the pump bore size. There are nine pump bore sizes and each has its own three-number code referring to sizes ranging from 1 1/16 inches through 2 3/4 inches in diameter. In this example, a bore size of 1 1/2 inches, the next three characters will be the number “150”.
- The next four characters refer to a description of the pump.
  - The first character specifies whether the pump is a rod or tubing pump. The example pump is a rod type pump, designated by the letter “R.”
  - The second character refers to the type of barrel and specifies whether it is a heavy-wall, a thin-wall or a liner barrel. The example pump has a stationary, heavy-wall barrel, designated by the letter “H.”
  - The third character refers to the location of the seating assembly, whether top or bottom, and if on the bottom, whether it is a stationary or traveling barrel. The example pump has a top anchor, and this is designated by using the letter “A.”
  - The fourth character in this series refers to the type of seating assembly. There are only two choices: a cup-type assembly, indicated by the letter "C", or a mechanical seating assembly, indicated by an "M." The example pump has a mechanical assembly so the letter "M" is placed in the fourth position. A rod-type pump with a stationary, heavy-wall barrel, a metal plunger, and a top anchor with a mechanical-type seating assembly will have the designation RHAM.
- The next three characters refer to pump length. The first character refers to the barrel length in feet. For this example, it is 8 ft. The second character refers to the nominal plunger length, again in feet. A rule of thumb states that to limit fluid slippage, pump plungers should be:
  - 3 feet long in wells less than 3,000 feet deep
- 3 feet plus 1 foot in length for each 1,000 feet between 3,000 and 6,000 feet of depth
- 6 feet long for wells 6,000 feet and deeper

One exception to this rule is that shorter plungers should be used for pumping viscous oils. For this example, assume a plunger length of 3 feet.

- The final character is the total length of extensions to the barrel, again expressed in feet. Extensions of 6 inches to 4 feet can be added to both ends of a heavy-wall barrel. These extensions are used primarily to prevent scale buildup within the barrel. For the example pump, assume an extension length of 2 feet.

Therefore, for this example, the final API pump designation would be **20-150-RHAM-8-3-2**.

In order to purchase a pump, the only additional information needed would be the type of material that the liner or barrel and plunger are to be made of, the plunger clearance, the valve material, and the length of each extension to be installed.
SUBSURFACE PUMP SELECTION

Selecting a subsurface pump for a beam pumping system is a matter of estimating the pump displacement that corresponds to a desired production rate, and then determining the optimal combination of stroke length, pump speed and plunger diameter for this displacement. Once the pump is sized, we can consider what type of pump is most appropriate for the given set of operating conditions.

PUMP DISPLACEMENT REQUIREMENTS

A subsurface pump displaces a volume defined by its stroke length, pump speed (strokes per minute), plunger diameter and volumetric pump efficiency:

$$PD = 0.1166 \frac{S_p}{B/D} N D^2 E_p$$  \hspace{1cm} (1)

where

- $PD =$ pump displacement at 100 percent volumetric efficiency,
- $S_p =$ effective plunger stroke length, inches
- $N =$ pumping speed, strokes/min
- $D =$ plunger diameter, inches
- $E_p =$ volumetric pump efficiency. This quantity is usually less than 1.0 for the following reasons:

  - Leakage of fluid around the plunger—The volume of fluid that slips down around the plunger during a pump stroke, and thus is not actually displaced, is known as slippage.
  - Foaming of the fluid within the pump—When this occurs, the pump becomes less efficient because it not only displaces fluid but also compresses the gas phase of the foam.
  - Shrinkage of the fluid—The fluid pressure and temperature decrease it is produced to the surface. This causes gas to come out of solution and the volume of the produced liquid to shrink by a factor corresponding to the formation volume factor at the pump depth.

Local operating conditions determine the pump efficiency, which is typically in the range of 70 to 80 percent.

STROKE EFFICIENCY ($E_s$)

The effective stroke length ($S_p$) in Equation 1 is the stroke length at the pump. Because of rod stretch and contraction, acceleration and inertial effects, $S_p$ will be considerably shorter than the polished rod stroke length measured at the surface.

For our initial pump sizing determination, we may define the stroke efficiency ($E_s$):

$$E_s = \frac{S_p}{S}$$

This quantity is typically on the order of 0.75 to 0.85. A more precise determination of $S_p$ is performed as part of the detailed rod pump system design.

ACTUAL PUMP REQUIREMENTS
Taking into account the stroke efficiency, we can modify Equation 1 to determine the pump requirements for a desired surface production rate:

\[
q = 0.1166(D^3)(S)(N)(E_p)(E_s)
\]  

(3)

where:
- \(q\) = surface production rate, B/D
- \(D\) = pump diameter, inches
- \(S\) = stroke length at surface, inches
- \(N\) = pump speed, stroke/minute
- \(E_p\) = pump efficiency, fraction
- \(E_s\) = stroke efficiency, fraction

Example:

The pumping unit currently in place at a well has a stroke length of 86 inches. What minimum plunger diameter is required to attain a production rate of 300 STB/D at a pump speed of 14 strokes per minute, assuming pump and stroke efficiencies of 80 percent?

Solution:

\[
q = 0.1166(D^3)(S)(N)(E_p)(E_s)
\]

\[
D = \sqrt[3]{\frac{q}{0.1166SNE_pE_s}} = \sqrt[3]{\frac{300}{0.1166 \times 36 \times 14 \times 0.8 \times 0.8}} = 1.83 \text{ inch} \Rightarrow 2.00 \text{ inch}
\]

For the conditions of this example, a 2-inch plunger would be more than adequate for a 300 STB/D rate. But many other combinations of \(S\), \(N\) and \(D\) could be used to attain this rate. The challenge in sizing a subsurface pump is to determine which of these combinations is optimal.

**PUMP SIZING**

Pump sizing entails placing certain limits on plunger diameter, stroke length and pump speed. For example, if the plunger diameter is too large, it may impose unnecessarily high stresses on the sucker rods and surface equipment. If it is too small, it will require high pump speeds to achieve the necessary production rate, resulting in higher peak loads on the equipment. Therefore, for a desired production rate, we must find an optimal combination of plunger diameter, stroke length and pump speed.

**PLUNGER SIZE**

We begin the selection procedure by determining the optimal pump plunger size for a desired surface production rate. This is done in Table 1 for well depths of up to 8000 feet, assuming a pump efficiency of 80 percent (the table values are based on an internal report of the Bethlehem Steel Company). In this case, for example, a plunger diameter of 1.25 inches would be recommended for a surface production rate of 100 barrels per day at a pump depth of 5000 ft.
Table 1: Optimal Pump Plunger Diameters (inches)
(After Bethlehem Steel Co. Sucker Rod Handbook (1958) and Brown (1980))

<table>
<thead>
<tr>
<th>Net Lift, ft</th>
<th>100</th>
<th>200</th>
<th>300</th>
<th>400</th>
<th>500</th>
<th>600</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>1.25 - 1.50</td>
<td>1.50 - 1.75</td>
<td>1.75 - 2.00</td>
<td>2.00 - 2.25</td>
<td>2.25 - 2.50</td>
<td>2.50 - 2.75</td>
</tr>
<tr>
<td>3000</td>
<td>1.25 - 1.50</td>
<td>1.50 - 1.75</td>
<td>1.75 - 2.00</td>
<td>2.00 - 2.25</td>
<td>2.25 - 2.50</td>
<td>2.25 - 2.50</td>
</tr>
<tr>
<td>4000</td>
<td>1.25</td>
<td>1.50 - 1.75</td>
<td>1.75 - 2.00</td>
<td>2.00 - 2.25</td>
<td>2.00 - 2.25</td>
<td>2.25</td>
</tr>
<tr>
<td>5000</td>
<td>1.25</td>
<td>1.50 - 1.75</td>
<td>1.75 - 2.00</td>
<td>1.75 - 2.00</td>
<td>2.00 - 2.25</td>
<td>2.25</td>
</tr>
<tr>
<td>6000</td>
<td>1.25</td>
<td>1.25 - 1.50</td>
<td>1.50 - 1.75</td>
<td>1.75</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7000</td>
<td>1.125 - 1.25</td>
<td>1.25 - 1.50</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8000</td>
<td>1.125 - 1.25</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Having established an optimal range of plunger diameters, we can consult the pump manufacturer to select the appropriate pump type and production tubing diameter. Table 2 shows a typical manufacturer’s recommendation.

Table 2: Recommended Tubing Diameters for Various Pump Bores
(After Axelson, Inc., 1980)

<table>
<thead>
<tr>
<th>Pump Type</th>
<th>Tubing size, inches</th>
<th>Pump bore, inches</th>
</tr>
</thead>
<tbody>
<tr>
<td>RH (Rod, Heavy Wall)</td>
<td>1.50</td>
<td>1.0625</td>
</tr>
<tr>
<td></td>
<td>2.00</td>
<td>1.25</td>
</tr>
<tr>
<td></td>
<td>2.50</td>
<td>1.25 to 1.75</td>
</tr>
<tr>
<td></td>
<td>3.00</td>
<td>2.25</td>
</tr>
<tr>
<td>RW (Rod, Thin Wall)</td>
<td>1.25</td>
<td>0.875</td>
</tr>
<tr>
<td></td>
<td>1.50</td>
<td>1.25</td>
</tr>
<tr>
<td></td>
<td>2.00</td>
<td>1.50</td>
</tr>
<tr>
<td></td>
<td>2.50</td>
<td>2.00</td>
</tr>
<tr>
<td></td>
<td>3.00</td>
<td>2.50</td>
</tr>
<tr>
<td>TH (Tubing, Heavy Wall)</td>
<td>2.00</td>
<td>1.75</td>
</tr>
<tr>
<td></td>
<td>3.00</td>
<td>2.25</td>
</tr>
<tr>
<td></td>
<td>3.50</td>
<td>2.75</td>
</tr>
</tbody>
</table>
**STROKE LENGTH AND PUMP SPEED**

The optimal stroke length-pump speed (SN) combination is based on establishing a maximum practical limit below which the rods have sufficient time to free-fall through the fluid on the downstroke. *(Figure 1)* illustrates this limit graphically for the example of a conventional pumping unit (such data are available from the manufacturer for various pumping unit types).

In this figure, the stroke length, in inches, is plotted on the vertical scale, and the maximum practical limit of stroke length and pump speed is indicated by the blue line. During the downstroke, the rods should be allowed to free-fall to avoid excessive stress on the polished rod clamp and hanger bar. Therefore, we must select a stroke length and pump speed that fall below this limit.

Assume, for example, the following conditions:
Desired surface production rate = 400 STB/D,
Pump efficiency = 80 percent
Plunger diameter = 2 inches.
Stroke efficiency = 85 percent.

By rearranging Equation 2 and substituting the specific values above, we obtain a relationship for $SN$:

$$SN = \frac{1262 \text{ SPM}}{(0.8)(0.166)(0.85)(4)}$$

This value is shown in the figure. Any combination of stroke length and pump speed that falls along or to the left of this line will satisfy this specific pump design. The point where this line intersects the maximum practical limit line suggested by the manufacturers is at a stroke length of about 45 inches and a pump speed of about 28 strokes/min. (This pump speed is higher than would generally be seen in the field). As a first approximation in designing the balance of the rod pump system, use a pump design specification of $D = 2$ inches, $S = 45$ inches and $N = 28$ SPM.

Note that the above design procedure is not absolute. A number of wells have operated very efficiently at pump speeds above the designated maximum. Local experience will provide the practical knowledge needed to modify design parameters.
ROD STRING DESIGN

The API has established standard specifications for the fabrication of sucker rods, including pony rods, polished rods, couplings, and sub-couplings (API Spec 11B, 1998). This publication shows that sucker rods are usually made in 25 ft lengths, except in California where they come in 30 ft lengths. The six standard rod diameters are 1/2, 5/8, 3/4, 7/8, 1, and 1 1/8 inch.

MATERIALS

Steel sucker rods normally have an iron composition of greater than 90 percent. Other elements are added to the steel alloys to increase hardness, reduce oxidation, and combat corrosion. For example (Weatherford, 2005):

- **API Grade C** rods are designed for light-to-medium load applications in non-corrosive wells; they have minimum and maximum tensile strengths of 90,000 and 115,000 psi, respectively.
- **API Grade D** rods are designed for heavy loads in non-corrosive or inhibited environments, and have tensile strength limits of 115,000 and 140,000 psi.
- **API Grade K** rods are fabricated from AISI 4623 nickel-molybdenum-alloy steel, and are designed for light-to-medium load applications in corrosive wells.
- **API Grade KD63** rods are fabricated from AISI 4720 nickel-chromium-molybdenum-alloy steel, and are designed for heavy load applications in effectively corrosive wells.

One standard operating practice is to run grade D rods except where severe corrosion or H₂S embrittlement may be a problem. Higher tensile strength rods are available for special applications (e.g., Weatherford S-88 and EL, Norris 97, etc.). Such rods must be handled with care because they may lose their high strength properties if damaged.

Fiberglass sucker rods were introduced in the early 1970's. Initially, their main area of application was in corrosive wells, but it later became clear that fiberglass rods offer additional benefits. Their lighter weight (about 70% less than steel rods) reduces the load on the pumping unit system, allowing for higher production without changing the pumping unit. In addition, when combined in the proper proportions with steel rods, fiberglass rods can produce longer pump strokes than stiffer, all-steel rod strings. At the same time, fiberglass rods are more expensive than steel rods. They also increase the complexity of the installation design, and they require a greater degree of care in the field (Gibbs, 1991).

ROD STRESS CALCULATIONS

It is important to not exceed the maximum allowable rod stress. To calculate the maximum rod stress, we may use the Modified Goodman equation (or, as seen in many publications, the Modified Goodman diagram):

\[
S_A = (0.25 T + 0.5625 S_{min}) \times SF
\]

where:
- \( S_A \) = maximum allowable rod stress, psi
- \( T \) = minimum tensile strength, psi
- \( S_{min} \) = minimum rod stress, psi
- \( SF \) = service factor

The specifications for tensile strength, \( T \), have been calculated for each API rod grade. The minimum rod stress, \( S_{min} \), is measured directly or estimated for the proposed application; it is equal to the minimum polished rod load divided by the cross-sectional area of the rod.

The service factor, \( SF \), depends on the environment in which the rods will be placed (Table 1). \( SF \) will be 1.0 for API grade C and D rods, when used in non-corrosive environments, but less than 1.0, for saltwater and H₂S environments. Note that the service factor for corrosive environments is lower for grade C rods than it is for grade D rods.
Table 1: Service Factors for Sucker Rods (after Brown, 1980)

<table>
<thead>
<tr>
<th>Service</th>
<th>API Grade &quot;C&quot; Rods</th>
<th>API Grade &quot;D&quot; Rods</th>
</tr>
</thead>
<tbody>
<tr>
<td>Non-corrosive</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Salt water</td>
<td>0.65</td>
<td>0.90</td>
</tr>
<tr>
<td>Hydrogen sulfide</td>
<td>0.50</td>
<td>0.70</td>
</tr>
</tbody>
</table>

Example: Use the Modified Goodman equation to calculate the maximum rod stress for a grade C rod operating in a saltwater environment;

- \( S_{\text{min}} = 10,000 \text{ psi} \)
- the minimum tensile strength, \( T = 90,000 \text{ psi} \)
- the service factor, SF, in saltwater for grade C rods is 0.65.

Substituting these values into Equation 3 yields:

\[
S_A = (0.25 \times 90,000 + 0.5625 \times 10,000)(0.65) = 18,281 \text{ psi.}
\]

This calculated maximum rod stress should not be exceeded during pumping operations.

In general, maximum allowable rod stresses under working conditions should not be higher than 30,000 to 40,000 psi. High tensile-strength rods are rated at 40,000 to 50,000 psi for use in non-corrosive environments.

TAPERED ROD STRINGS

The most common design for a rod string longer than about 3,500 feet is a tapered design, using combinations of rods with different diameters, and installed with the largest diameter rod at the top and the smallest diameter rod at the bottom. API design procedures for tapered strings are based on a maximum of four different rod diameters, although commercially available software packages for rod system design can accommodate more sizes.

One fundamental design criterion used for a tapered rod string is that the string should have approximately the same unit stress in the top rod of each section. That is, the rod loads in the top rod of each section divided by their respective cross-sectional areas should be equal. This provides a safe design even if some corrosion pitting occurs.

In order to select the appropriate tapered rod string for a specific well, refer to API Recommended Practice 11L (1988; reaffirmed January 2000).

Data from one of these tables is reproduced in Table 2. The sample information given is for a 5/8"-1/2" tapered rod string.

Table 2: Rod and Pump Data for 5/8-inch - 1/2 inch Tapered Rod String (After API RP 11L)

<table>
<thead>
<tr>
<th>Rod No.</th>
<th>Plunger Diameter (D) inches</th>
<th>Rod Weight (W), lb. per foot</th>
<th>Elastic Constant (E), (10^6) in/lb-ft</th>
<th>Frequency factor (F&lt;sub&gt;e&lt;/sub&gt;)</th>
<th>Rod string, % of each size 5/8 inch</th>
<th>1/2 inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>54</td>
<td>1.06</td>
<td>0.908</td>
<td>1.668</td>
<td>1.138</td>
<td>44.6</td>
<td>55.4</td>
</tr>
<tr>
<td>54</td>
<td>1.25</td>
<td>0.929</td>
<td>1.633</td>
<td>1.140</td>
<td>49.5</td>
<td>50.5</td>
</tr>
<tr>
<td>54</td>
<td>1.50</td>
<td>0.957</td>
<td>1.584</td>
<td>1.137</td>
<td>56.4</td>
<td>43.6</td>
</tr>
<tr>
<td>54</td>
<td>1.75</td>
<td>0.990</td>
<td>1.525</td>
<td>1.122</td>
<td>64.6</td>
<td>35.4</td>
</tr>
</tbody>
</table>
The first column in Table 2 is the API rod number designation, given in a “shorthand” form where “5” indicates 5/8 inch, and “4” indicates 4/8, or 1/2 inch. The second column is the pump plunger diameter. The third, fourth and fifth columns refer to certain rod constants, while the final two columns show the recommended percentages of each diameter rod to be used in a tapered rod string for this specific rod number designation.

For example, for a plunger diameter of 1.06 inches, the right-hand columns show that 44.6% of the string should consist of 5/8-inch rods and 55.4% of 1/2-inch rods. For a 5000-foot well, this would correspond to:

- \(0.446 \times 5000 = 2230\) ft of 5/8-inch rods on top, and
- \(0.554 \times 5000 = 2770\) ft of 1/2-inch rods on bottom.

Because rods generally come in 25 ft lengths, these two numbers would be adjusted to 2225 and 2775 ft, respectively.

Using the data for a 3/4-5/8-inch ("75") tapered string as given in Table 3, a 1.06-inch plunger would require 27% of the rod string to have 7/8-inch rods, 27.4% to have 3/4-inch rods, and 45.6% to have 5/8-inch rods. For the same 5000-foot well as considered above, this would yield a design made up of:

- \(0.27 \times 5000 = 1350\) feet of 7/8-inch rods at the top,
- \(0.274 \times 5000 = 1375\) feet of 3/4-inch rods in the middle section, and finally,
- \(0.456 \times 5000 = 2275\) feet of 5/8-inch rods on the bottom.

### Table 3: Rod and Pump Data for 3/4-inch - 5/8 inch Tapered Rod String
*(After API RP 11L)*

<table>
<thead>
<tr>
<th>Rod No.</th>
<th>Plunger Diameter (D) inches</th>
<th>Rod Weight (W), lb. per foot</th>
<th>Elastic Constant (E), (10^6) in/lb-ft</th>
<th>Frequency factor (F)</th>
<th>Rod string, % of each size</th>
</tr>
</thead>
<tbody>
<tr>
<td>75</td>
<td>1.06</td>
<td>1.566</td>
<td>0.997</td>
<td>1.191</td>
<td>27.0 27.4 45.6</td>
</tr>
<tr>
<td>75</td>
<td>1.25</td>
<td>1.604</td>
<td>0.973</td>
<td>1.193</td>
<td>29.4 29.8 40.8</td>
</tr>
<tr>
<td>75</td>
<td>1.50</td>
<td>1.664</td>
<td>0.935</td>
<td>1.189</td>
<td>33.3 33.3 33.3</td>
</tr>
<tr>
<td>75</td>
<td>1.75</td>
<td>1.732</td>
<td>0.892</td>
<td>1.174</td>
<td>37.8 37.0 25.1</td>
</tr>
<tr>
<td>75</td>
<td>2.00</td>
<td>1.803</td>
<td>0.847</td>
<td>1.151</td>
<td>42.4 41.3 16.3</td>
</tr>
<tr>
<td>75</td>
<td>2.25</td>
<td>1.875</td>
<td>0.801</td>
<td>1.121</td>
<td>46.9 45.8 7.2</td>
</tr>
</tbody>
</table>

Designing a pumping system using the API procedure is an iterative process, including the design of sucker rod strings. First, select a tapered rod string and calculate whether the maximum rod stress exceeds the maximum limits for the class of rods used. If it does, then it is necessary to select other rod number designations until a string is found that will satisfy the design limitation.

Using the API design calculations, one of these tapered strings might meet the design limitations. If not, a heavier or lighter rod string should be designed.
ROD LOADS DURING PUMPING

Once the basic properties of the sucker rods and the design parameters of the rod string are established, the next step in designing the string is to calculate the rod loads during pumping operations. This information is used to size the pumping unit.

Consider the reciprocating motion of the polished rod and the actions of the traveling and standing valves during the pumping cycle. Throughout this cycle, the load on the sucker rod string varies continuously.

- The maximum load occurs shortly after the beginning of the upstroke, when the traveling valve closes. The polished rod must carry the full weight of the fluids, the rods, and the added inertial effects that occur as the motion of the rods is reversed. This is known as the Peak Polished Rod Load, or PPRL.
- The minimum load occurs shortly after the beginning of the downstroke, as the traveling valve opens. At that point, the rods no longer carry the fluid load and the inertial effects are reversed, thereby reducing the total rod load below the weight of the rods in the produced fluids. This is known as the Minimum Polished Rod Load, or MPRL.

To calculate the maximum and minimum rod loads, we begin with the PPRL. This is equal to

\[ PPRL = W - Br + Fo + I \]  \hspace{1cm} (2)

Where:

- \( W \) = weight of rods in air, lb
- \( Br \) = buoyancy of rods in fluids, lb
- \( Fo \) = weight of fluid column, lb
- \( I \) = effects of inertial and acceleration forces, lb.

Consider, for example, the following tapered sucker rod string:

- 1350 ft of 7/8-inch rods
- 1375 ft of 3/4-inch rods
- 2275 ft of 5/8-inch rods

The average density of the hole fluid is 55 lb/cu ft ( \( \gamma = 0.88 \)).

The average unit weight of the rods in air is given as 1.566 lb/ft. Thus, the total weight of the rods in air is simply equal to:

\[ W = (5000 \times 1.566) = 7830 \text{ lb} \]

The second term, the buoyancy of rods in the produced fluids, is equal to the weight of the fluids that the rods displace. It is calculated as follows:

\[ Br = (62.4 \cdot \gamma \left( \frac{W}{488} \right)) \]  \hspace{1cm} (3)

or

\[ Br = 0.128 \gamma W \]

where:

- \( \gamma \) = specific gravity of the produced fluid, dimensionless
- \( W \) = weight of rods in air, lb
- 62.4 = density of water, lb/cu ft
- 488 = density of steel, lb/cu ft.

Using Equation 3,

\[ Br = 0.128 \gamma W = (0.128)(0.88)(7830) = 882 \text{ lb}. \]
The weight of the rods in fluid, $W_{rf}$, is equal to the weight of the rods in air minus the buoyancy effect of the rods, or:

$$W_{rf} = W - B_f = 7830 - 882 = 6948 \text{ lb.}$$

The weight of the fluid column supported by the net plunger area, $F_o$, is equal to the density of the produced fluid multiplied by the net plunger area, multiplied by the height of the static fluid level. The net plunger area is equal to the cross-sectional area of the plunger minus the area of the rods. The suggested API procedure, however, disregards the area of the rods in these calculations.

$$F_o = \rho \cdot A \cdot H \tag{4a}$$

where

- $\rho = \text{density of fluid, lb/cu ft}$
- $A = \text{area of plunger, sq ft}$
- $H = \text{fluid level, ft}$

Substituting $\rho = 62.4 \gamma$, and $A = \pi D^2 / (4 \times 144)$, where $D$ is the plunger diameter in inches:

$$F_o = 0.34 \gamma D^2 H \tag{4b}$$

Where $\gamma = 0.88$, the plunger diameter is 1.06 inches and the static fluid column height is assumed equal to the pump depth of 5000 ft:

$$F_o = (0.34)(0.88)(1.06^2)(5,000) = 1681 \text{ lb.}$$

This is the weight of the fluid column.

Returning to Equation 2 and substituting the calculated values yields a PPRL value of 8629 lb plus the effects of the inertial and acceleration forces, which are still unknown. Due to the complex behavior of the polished rod, predicting the effects of these forces is the most difficult part of the analysis. The rod string stretches and contracts in response to a cyclic motion (not a simple harmonic) and to motions of the crank and pitman that are different for each type of pumping unit. In addition, because of the elasticity of the sucker rod system, stress waves run up and down the rod in response to the various forces that affect the rod.

Several methods may be used to estimate the effect of inertial and acceleration forces in the polished rod loads.

- One method is to make simplifying assumptions about the behavior of the rod and the pump motion; however, this method is not used often today.
- Another method is to calculate rod loads by solving the nonlinear partial differential equations that represent the behavior of the rod string. Gibbs, originally on his own and later with Neely at Shell Oil Co., solved these equations in an effort to analyze pump operation rather than pump design (Gibbs, 1963; and Gibbs and Neely, 1966). Their work uses known values of peak polished rod loads at the surface to calculate the behavior of the pump in the subsurface.
- Yet another method is the API design procedure that uses empirical correlations derived from repeated observations of pump behavior. This research was conducted at the Midwest Research Institute where computer simulations of rod-pumping systems were performed. By simulating a wide range of pumping conditions, the Institute was able to develop correlations for predicting polished rod loads.

Both analytical solution methods and the API procedure are incorporated into many commercially available software programs for rod pump system design.
API SYSTEM DESIGN PROCEDURE

Designing a rod pumping system involves complex calculations of the dynamic relationships between production rates and stresses at various points in the system. For many years, the standard for such design calculations has been the API Recommended Practice for Design Calculations for Sucker Rod Pumping Systems (RP 11L). First published in 1966, updated in 1988 and reaffirmed in January 2000, this recommended practice involves the use of dimensionless variable correlations for optimizing design parameters.

Although API-RP 11L has been supplemented and in some cases supplanted by PC-based programs that are quicker to use and more general in their application, it is discussed here to provide some insight into the considerations involved in rod pump system design.

ASSUMPTIONS AND LIMITATIONS

The following assumptions are incorporated into API RP 11L:

- Applies to conventional pumping unit motion.
- Pumping unit employs medium-slip prime mover.
- Steel rod string—in the case of tapered strings, the rods become smaller with depth.
- Negligible friction at the stuffing box and in the pump.
- Pump completely liquid-filled (no gas interference or fluid pound).
- Anchored tubing (a correction formula is included to approximate effects of unanchored tubing).
- Pumping unit in balance.

DIMENSIONLESS VARIABLES

The experimental work conducted at the Midwest Research Institute (Kansas City, Missouri, 1964) on sucker rod pumping, later adopted by the API, consisted of numerous analog computer simulations over a wide range of pump operating conditions. The results are correlated based on two dimensionless variables: dimensionless pump speed and dimensionless rod stretch.

The dimensionless pump speed variable may have two forms: $N/N_o$ or $N/N'_o$.

- $N/N_o$ is the pump speed in strokes per minute ($N$), divided by the natural frequency of the string ($N_o$).
- $N/N'_o$ is equal to $(N/N_o) / F_c$, where $F_c$ is the frequency factor of the rod string. $F_c$ is equal to 1.0 for non-tapered rod strings, and is greater than 1.0 for tapered rod strings. Table 4.1 of API RP 11L lists $F_c$ values for various tapered rod strings.
- If a rod string is non-tapered, then $N/N_o = N/N'_o$.

These dimensionless variables are calculated as follows:

$$N'_o = \frac{A}{4L}$$

Where $A$ = speed of sound in steel rods (approximately 980,000 ft/min)
$L$ = rod length, ft

Substituting this value of $N_o$ into the dimensionless variable,
and

\[
\frac{N}{N'} = \frac{N}{\left(4E\right)} = \frac{NL}{245000}
\]

Thus, for a given or assumed pump speed, rod string length and frequency factor, we may calculate a value for each of these dimensionless variables.

Before making this calculation, consider the second dimensionless variable, the dimensionless rod stretch, defined as

\[
\frac{F_o}{Sk_r}
\]

where

- \(F_o\) = the static fluid load in pounds
- \(S\) = the polished rod stroke length, in inches
- \(k_r\) = the spring constant of the rod string.

\(k_r\) can be derived from \(E\) (the reciprocal of \(k_r\), multiplied by the rod length, \(L\)), which is also listed in Table 4.1 of API RP 11L. \((\text{Figure 1: Excerpt from Table 4.1 of API RP 11L})\). \(Sk_r\) represents the load in pounds required to stretch the rod string the length of the polished rod.

The dimensionless rod stretch term represents the rod stretch caused by the static fluid load; it is given as a fraction of the polished rod stroke length. \((F_o/Sk_r = \text{rod stretch as a fraction of polished rod stroke length})\) for example, if \(F_o/Sk_r = 1\), then the rod string will stretch an amount equal to the length of the polished rod. The value of this dimensionless variable may be easily calculated for a known or assumed polished rod stroke length.

On a practical note, observations have determined that under-travel and over-travel of the pump plunger can be prevented if \(N/N'\) is less than 0.35, and \(F_o/Sk_r\) is less than 0.50.

From the work performed at the Midwest Research Institute, five correlations were developed—these are shown in API RP 11L as Figures 4.1, 4.2, 4.3, 4.4 and 4.5. These five correlations, based on the two dimensionless variables for their presentation, are used to systematically develop the API design procedure for a pumping system.

The first three correlations affect the rod and downhole pump system—specifically, the plunger stroke length, the peak polished rod load, and the minimum polished rod load. The last two correlations are used to calculate peak torque and polished rod horsepower.
DESIGN EXAMPLE:

To demonstrate the use of these correlations consider a well that is to be pumped at a rate of 100 B/D. The well conditions are:

- Fluid level ($H$) and pump depth ($L$) = 5000 ft
- Tubing diameter = 1.9 inches.
- Tubing is anchored
- A tapered sucker rod string (API 75), consisting of 1350 ft of 7/8-inch rods, 1375 ft of 3/4-inch rods, and 2275 ft of 5/8-inch rods, is calculated to be sufficient for this design. These grade D rods have a maximum allowable rod stress of 34000 psi.
- Plunger diameter = 1.06 in.
- For preliminary design, pump speed ($N$) = 16 strokes/min
- Polished rod stroke length ($S$) = 64 inches.
- Specific gravity ($\gamma$) of the pumped fluid = 0.88.

Substituting these data into the equation for pump displacement, using the polished rod stroke length, $S$, as an estimate of the pump stroke length, and assuming a pump efficiency of 100 percent, we obtain:

$$PD = 0.1166 \cdot S \cdot N \cdot D^2 \cdot E_{p} = (0.1166)(64)(16)(1.06)^2 = 135 \text{ B/D}$$

This quick calculation gives us a pump displacement of about 135 B/D, which should satisfy the well’s IPR-calculated objective of 100 B/D.

PLUNGER STROKE CORRELATION

The first correlation allows us to calculate the plunger stroke length (Figure 2: Plunger stroke correlation, after API RP 11L).
The dimensionless pump speed, \( N/N_0 \), is shown on the horizontal axis.

The bottomhole plunger stroke length divided by the polished rod stroke length \( (S_p/S) \) is plotted on the vertical axis. This ratio reflects the effect of rod stretch on the effective plunger stroke length.

The plot shows a series of curves, each for a different value of the dimensionless rod stretch \( (F_o/Skr) \). Because the values of \( N/N_0 \) and \( F_o/Skr \) are known—or can be calculated from the given data—we may use this correlation to calculate a value for \( S_p/S \).

For the example problem, \( N/N_0 \) is calculated as follows:

\[
N/N_0 = NL / 245,000 = (16 \times 5,000) / 245,000 = 0.326
\]

For the selected tapered rod string, \( F_c \) is equal to 1.191, and so:

\[
N/ N'_0 = N / (N_0 F_c) = 0.326 / 1.191
\]

\[
N/ N'_0 = 0.274
\]

The dimensionless rod stretch is equal to \( F_o/Skr \).

\( F_o \), the weight of the fluid supported by the rods, is

\[
F_o = 0.34 \gamma D^2 H
\]

where \( \gamma \) = specific gravity of fluid, lb/cu ft =
\( D \) = plunger diameter, inches
\( H \) = fluid level, ft

Where \( \gamma \) is 0.88, the plunger diameter is 1.06 inches and the static fluid column height is assumed equal to the pump depth of 5000 ft:

\[
F_o = (0.34)(0.88)(1.06^2)(5,000) = 1681 \text{ lb.}
\]

The polished rod stroke length \( (S) \) is 64 inches, and the reciprocal of \( k_r \) is equal to the elastic constant, \( E_r \), for the example rod string multiplied by its length, \( L \) (i.e., \( 1/k_r = E_r L \)). From API manual \( RP 11L \), we obtain, \( E_r \).

\[
E_r = 0.997 \times 10^{-6} \text{ in/lb ft}
\]

and substitute the value into the relationship for the dimensionless term:

\[
F_o/Skr = (1.681 \times 0.997 \times 10^{-6} \times 5000) / 64
\]

\[
F_o/Skr = 0.13
\]

Note that \( N/N'_0 < 0.35 \), and \( F_o/Skr < 0.50 \). This puts them within the practical limits for preventing under-travel and over-travel of the pump plunger mentioned above.

Referring back to (Figure 2), we see that with known values for the two dimensionless variables, we arrive at a value of \( S_p/S = 0.98 \). This means that if the tubing is anchored, as in this example, the bottomhole plunger stroke length is equal to 98 percent of the polished rod stroke length at the surface. Specifically, for a polished rod stroke length of 64 inches, the bottomhole stroke length will be 62.7 inches.

If the tubing is not anchored, the plunger stroke length corrected to account for the tubing contraction during the upstroke. This term is found by the following formula:

\[
\text{Tubing contraction} = E_t \times F_o \times L
\]

Where \( E_t = \) coefficient of elasticity for the tubing, in/lb ft
\( F_o = \) weight of fluid on rods, lb
\( L = \) tubing length, ft.

For 1.9-inch tubing, \( E_t \) has a value of \( 0.5 \times 10^{-6} \) in/lb ft. With \( F_o \) equal to 1681 lb, and \( L \) equal to 5000 ft, the tubing contraction in unanchored tubing, is:

\[
\text{Tubing Contraction} = (0.5 \times 10^{-6} \times 1681 \times 5000) = 4.2 \text{ in.}
\]
So, in this example, the net plunger stroke length would be \((62.7 - 4.2) = 58.5\) inches.

We can now calculate the subsurface displacement. Assuming that the tubing is anchored, the pump displacement is:

\[
PD = 0.1166S_o N D^2 = (0.1166)(62.7)(16)(1.06)^2 = 131 \text{ B/D.}
\]

This more than meets the example expected inflow rate of 100 B/D. In fact, it allows more downtime per day for maintenance. However, had the pump displacement rate not met the production objective, it would be necessary to revise the assumed pump data and repeat the calculations. Note that to determine the production rate at the surface, we must incorporate the pump efficiency \(E_p\) into the calculation.

**PEAK POLISHED ROD LOAD CORRELATION**

The second API correlation can be used to calculate the peak polished rod load (Figure 3: PPRL correlation, after API RP 11L). The same dimensionless variables, whose values are known, are plotted in the same general locations. In this case, the horizontal axis is simply \(N/N_o\). It is the vertical axis that is changed, and is now equal to \(F_1/Skr\). The terms in the denominator are already defined; their values were calculated earlier. \(F_1\), the only unknown, is referred to as the peak polished rod load factor.

![Figure 3: PPRL correlation](image)

The dimensionless pumping speed, \(N/N_o\), is equal to 0.326. The dimensionless rod stretch, \(F_1/Skr\), is equal to 0.13. With these two known values, we can use (Figure 3) to find a value of 0.37 for \(F_1/Skr\). Because Skr is known, it is possible to calculate \(F_1\). Note that \(F_1\) must be added to the weight of rods in the produced fluids to give a value for the peak polished rod load in pounds. It thus represents two terms: the weight of the fluid carried by the rods and the load caused by the inertial-acceleration forces.

The peak polished load, then, is equal to:
\[ PPRL = W_{rd} + F_1 \]  
\[ \text{Where } PPRL = \text{peak polished load, lb} \]
\[ W_{rd} = \text{weight of the rods in fluid, lb} \]
\[ F_1 = \text{weight of fluids on rods plus inertial-acceleration forces, lb} \]

Or, in terms of the correlation:

\[ PPRL = W_{rd} + \frac{F_1}{Sk_r} \times Sk_r \]  
\[ \text{Where } PPRL = \text{peak polished load, lb} \]
\[ W_{rd} = \text{weight of the rods in fluid, lb} \]
\[ F_1 = \text{weight of fluids on rods plus inertial-acceleration forces, lb} \]
\[ Sk_r = \text{the load in pounds required to stretch the rod string the length of the polished rod.} \]

For this sample problem, we determine the weight of the API 75 rod string in the produced fluid to be 6948 lb. The second term \((F / Sk_r)\) is equal to 0.37. All that is needed to calculate \( PPRL \) is the value of \( Sk_r \).

The spring constant for the rod string \((k_r)\) is equal to the reciprocal of the coefficient of elasticity of the rod string \((E_r)\) multiplied by the length of rods \((L)\):

\[ \frac{1}{k_r} = (E_r)(L) \]
\[ \frac{1}{k_r} = (0.997 \times 10^{-6})(5000). \]

Then, taking the reciprocal, we obtain:

\[ k_r = 200.6. \]

With a polished rod stroke length, \( S \), of 64 inches, \( Sk_r \) is calculated:

\[ Sk_r = (64)(200.6) = 12838 \text{ lb}. \]

With this information the peak polished rod load for the example can be calculated:

\[ PPRL = W_{rd} + ((F / Sk_r) \times Sk_r) \]
\[ PPRL = 6948 + (0.37 \times 12838) = 11698 \text{ lb}. \]

This value represents the best estimate of the rod load, and can be used for design purposes. Under actual operating conditions, it may be different, but not significantly so.
**MINIMUM POLISHED ROD LOAD CORRELATION**

The third API correlation (Figure 4: MPRL correlation. After API RP 11L) involves calculating the minimum polished rod load (MPRL). This load occurs just at the beginning of the downstroke, when there is maximum downward inertial force, and the rods no longer carry the fluid load.

The two dimensionless variables are in the same relative locations, but now the term \( F_2/S_{kr} \) is on the vertical axis. \( F_2 \) is defined as the *minimum polished rod load factor*.

Using the information already obtained, with an \( N/N_0 \) of 0.326, and an \( F_2/S_{kr} \) value of 0.13, the point of intersection on the vertical axis is at 0.21. With the value for \( S_{kr} \) known, we can calculate \( F_2 \).

\( F_2 \) is the weight that must be subtracted from the weight of the rods in the produced fluid to give the MPRL. As such, it represents the inertial and acceleration force at the point of minimum load. Its calculation proceeds directly parallel to that for the peak polished rod load:

\[
MPRL = W_r - F_2 \\
MPRL = W_r - (F_2/S_{kr} \times S_{kr}) \\
MPRL = 6948 - (0.21 \times 12,838) \\
MPRL = 4252 \text{ lb.}
\]

This MPRL of 4252 lb. is used for design purposes; the actual value of \( F_2 \) after installation may be different.

**SELECTION OF COUNTERWEIGHTS**
Counterweights are added to a pumping unit to provide a counterbalance effect at the polished rod. We can appreciate their importance by considering what would happen if this counterbalance was not present. On the upstroke, the prime mover would have to do an enormous amount of work. An excessive torque would be exerted on the gear reducer, requiring a larger prime mover and a more substantial gear reducer. But on the downstroke, with the polished rod load substantially reduced and the force of gravity pulling the rods and plunger downward, the prime mover would have little work to do. In short, it would be practically impossible to design a system to efficiently handle such load variations.

Counterweights serve to reduce these load variations by moving downward during the upstroke (thus helping to lift the rod string) and upward during the downstroke (thus keeping the rod string from falling too quickly). This reduces the size requirements for the prime mover and provides a more even load to the gearbox.

For design purposes, we can estimate of an “ideal” counterbalance that establishes an approximately equal balance between (1) the upstroke and downstroke work done by the prime mover and (2) the net torque exerted on the gear reducer during each half of the pumping cycle. This balance should be the design objective, and is determined by calculating the maximum and the minimum loads on the polished rod during the pumping cycle and then calculating their mean value:

\[
\text{Mean value} = W_{rf} + 0.5 F_o \quad (8)
\]

Where \( W_{rf} \) = weight of rods in fluid, lb.
\( F_o \) = weight of fluid imposed on rods, lb.

The required \textit{counterbalance effect}, \( \text{CBE} \), should be approximately equal to the mean load. For API design purposes, however, the design value of the counterbalance effect for the conventional pumping unit is slightly larger than this amount—in fact, it is 1.06 times larger than the mean load.

\[
\text{CBE} = (1.06) (W_{rf} + 0.5 F_o). \quad (9)
\]

Substituting the data from the sample problem into this equation, we obtain the total required counterbalance effect:

\[
\text{CBE} = (1.06) \left( 6948 + \frac{1681}{2} \right) = 8236 \text{ lb}
\]

This \( \text{CBE} \) is required at the polished rod; it is transmitted there through the counterweights acting on the connecting pumping unit members. To determine the size of counterweights needed, a force balance around the various contributing structural members must be calculated that includes the effect of the unit’s geometry and its contribution to the \( \text{CBE} \). The pumping unit manufacturer provides this information. For a given pumping unit, pump stroke length, and desired \( \text{CBE} \), the manufacturer will specify the types of counterweights to be used and where they should be placed on the unit.

The actual \( \text{CBE} \) needed for a pumping unit is determined by field measurements of the peak and minimum polished rod loads. These measurements will dictate to what position the counterweights must be adjusted or, for an air-balanced unit, by how much the pressure in the cylinder must be changed.

**Peak Torque Correlation**

Consider the torque that is imposed on the crank by the low speed shaft of the gear reducer during the pumping motion. One important system design objective is to be sure that the peak torque does not exceed the manufacturer’s limits.

Remember that torque is equal to the force acting at right angles to a lever arm, multiplied by the length of the arm, and that torque tends to produce rotation at the point of connection. The torque of a pumping unit is provided by the prime mover, and is applied to the crank by the low speed shaft of the gear reducer that, in turn, causes the pump to operate (Figure 5: \textit{Torque generated by pumping unit}). This torque must be sufficient to cause the pump to operate in a continuous manner under normal operating conditions.
The net torque at the gear reducer is equal to the torque caused by the well loads acting on the polished rod minus the torque caused by the counterweights acting along the crank. Because the magnitudes of these two terms change during the pumping cycle, the peak torque is defined as the maximum net torque that occurs during the pumping cycle.

(Figure 6) shows the net torque imposed on a typical conventional beam pumping unit.

The dashed curves show the rod load and counterweight components of the torque during one full cycle,
beginning with the upstroke. Notice that the rod load during the upstroke is positive and the counterweight torque is negative. The maximum and minimum values occur about midway through the upstroke. As one value decreases, the other increases. The values become equal and very close to zero at the top of the upstroke, and reach maximum and minimum points midway through the downstroke. The solid curve shows the net effect of the two components—it reaches a maximum value (peak torque) midway through the up and down portions of the stroke. The gear reducer, then, must be designed to accommodate this peak net torque by an appropriate safety margin. Manufacturers specify the peak torque for each pumping unit. Note that the above representation is for a conventional unit and will be different for other types of units.

The peak torque of a pumping unit is calculated by using two additional API correlations. Peak torque, $PT$, is given by the multiple of four terms:

$$PT = \left(\frac{2T}{S^2k_r}\right)\left(\frac{S}{2}\right)\left(T_a\right)$$  \hspace{1cm} (10)

Where $PT$ = peak torque, lb-in
$T$ = torque, lb-in
$T_a$ = torque adjustment factor, fraction
$S_{kr}$ = load required to stretch rod string the length of the polished rod, lb.
$S$ = polished rod stroke length, in.

Notice that the second and third terms in the example problem are already known. It is the first and the fourth terms that must be obtained from the two new correlations.

The peak torque correlation is shown in (Figure 7) (Peak torque correlation, after API RP 11L). The first term in on the right-hand side of Equation 10 is shown on the vertical axis. The two familiar dimensionless variables ($N/N_0$ and $F_r/Sk_r$) are shown in the same relative positions as they were in earlier correlations.
Starting from \( N/N_0 = 0.326 \) on the horizontal axis, and moving vertically upward to a value of \( F_s/S_{kr} = 0.13 \), we determine that the corresponding value of \( (2T/S_{kr}) \) is 0.32. Thus, for our example design, Equation 10 has the following form:

\[
PT = (0.32) (12838) (64/2) T_a = 131461 \ T_a.
\]

A single unknown, \( T_a \), remains to be determined. Where \( (W_{rf}/S_{kr}) = 0.3 \), then \( T_a = 1.0 \), and no further adjustments to the equation are necessary. Where \( (W_{rf}/S_{kr} \neq 0.3) \), then \( T_a \) differs from 1.0. In order to calculate its value under these conditions, turn to the fourth API correlation shown in (Figure 8) (Peak torque correlation, after API RP 11L, continued).

Here, the familiar dimensionless variables are found on the vertical and horizontal axis. In this case, the horizontal axis is \( N/N_0 \). Within the chart are percentage values ranging from positive values on the outer rim to negative values near the center. For specific values of the dimensionless variables, this chart is used to find the appropriate percentage. The value of this percentage is then used to calculate the torque adjustment constant, \( T_a \), using the following equation:

\[
(11)
\]
Using the specific values for the dimensionless variables that have been used previously (i.e., 0.274 on the horizontal axis and 0.13 on the vertical axis) the percentage correction is found to be equal to about +2.5% (Figure 8, above).

Substituting this value into the equation:

\[ T_a = 1.0 + (\pm X_{%}) \left( \frac{W_{rf}}{Sk_r} \right) - 0.3 \]

\[ = 1.06 \]

When this value is included into the relationship for peak torque, \( PT \):

\[ PT = (131,465)(1.06) = 139,353 \text{ in/lb.} \]

This is the expected peak torque of the example pumping unit. The shaft of the gear reducer should be designed to withstand a maximum torque of at least this value. Later, this design specification will be introduced into the pump unit designation.

The gear reducer, acting through the V-belts, transmits the power and the rotating speed of the prime mover into the power and rotating speed needed by the pumping unit. Normally there are both high-speed and low-speed gears, and a careful selection of the gear reduction system is a necessary part of the manufacturer's overall design of the pumping unit.

**Polished Rod Horsepower Correlation**

The API procedure for calculating the polished rod horsepower, \( PRHP \), for the conventional unit is given by the following relationship:

\[ PRHP = \left( \frac{F_3}{Sk_r} \right) \times Sk_r \times SN \times (2.53 \times 10^{-6}) \]

(12)

where \( F_3 = PRHP \text{ factor, lb and, as defined above} \)
\( Sk_r = \text{load required to stretch rod string the length of the polished rod, lb} \)
\( S = \text{stroke length at surface, inches} \)
\( N = \text{pump speed, stroke/minute} \)

A value for \( F_3 \) may be found by using the fifth API correlation, as shown in (Figure 9) (PRHP correlation, after API RP 11L).
Notice this figure’s similarity to the other API correlations. The two familiar dimensionless variables appear in the same relative positions, and the new variable appears on the vertical axis. This variable is the first term in the calculation for the polished rod horsepower. To find its specific value, enter the horizontal axis the previously determined \( N/N_o \) value of 0.326, and move vertically upward to a \( F_o/Sk_r \) value of 0.13. These correspond to \( F_o/Sk_r = 0.225 \).

Substituting this value plus the known value of other terms into Equation 12 yields:

\[
PRHP = (0.223)(12,838)(64)(16)(2.53 \times 10^{-6}) = 7.48 \text{ HP}
\]

Knowing the \( PRHP \), we can determine the *nameplate horsepower* \( (HP_{np}) \), which is the minimum horsepower required to run the unit:

\[
HP_{np} = \frac{PRHP \cdot CLF}{E_{surf}} \tag{13}
\]

where
- \( PRHP \) = polished rod horsepower,
- \( CLF \) = cyclic load factor
- \( E_{surf} \) = surface efficiency of pumping unit, fraction

The cyclic load factor \( (CLF) \) represents the additional reserve power needed to handle the cyclic beam-pumping load. For low peak torques and uniform torque ranges, the cyclic load factor is low when peak torques are low and torque ranges are uniform. If, on the other hand, the peak torques are high and the ranges are non-uniform, the cyclic load factor is high.

The prime mover normally recommended for beam pumping units is the **NEMA Design D** electric motor. For this unit, the \( CLF \) is about 1.375. For NEMA C electric motors and multi-cylinder engines, the \( CLF \) will...
be about 1.897. These CLF values are valid for conventional and air balanced units, but will differ for MARK II units, as shown in Table 1.

<table>
<thead>
<tr>
<th>Table 1: CLF Values for Various Pumping Unit Types</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>(Courtesy of Lufkin Industries, Lufkin, TX, USA)</strong></td>
</tr>
<tr>
<td><strong>Unit Type</strong></td>
</tr>
<tr>
<td>Conventional and Air Balanced</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Mark II</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

The third term in Equation 13 is the surface efficiency, $E_{surf}$. It is equal to the combined efficiencies of the motor and the moving parts of the pumping unit, including the wire lines, structural bearings, transmission, gears, gearbox bearing, and the V-belt.

The mechanical efficiency is usually about 90%. The motor efficiency depends on the motor, its operating speed and load variation during a revolution of the crank. The motor manufacturer will provide motor efficiency information. For this example, assume that it is 70 percent.

$$E_{surf} = (0.9)(0.7) = 0.63.$$ 

This information may be used to calculate the nameplate horsepower for the example problem. Assume that a NEMA D electric motor, with a CLF of 1.375, is installed.

A number of factors, including polished rod horsepower, torque loading, system voltage, start-up problems and motor characteristics, will influence motor size. Substituting the known values into Equation 13 yields:

$$HP_{np} = \frac{(7.48)(1.375)}{0.63} = 16.3 \text{ HP}$$

The nameplate horsepower for this NEMA D motor is found to be 16.3. Notice that the motor nameplate horsepower is about two times the polished rod horsepower.

Because motors come in discrete sizes, as shown in Table 2, a 20 horsepower unit would be ordered for this example application.

<table>
<thead>
<tr>
<th>Table 2: Recommended Motor Sizes for API Pumping Units</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>(Courtesy of Lufkin Industries, Lufkin, TX, USA)</strong></td>
</tr>
<tr>
<td><strong>Unit Size</strong></td>
</tr>
<tr>
<td>57-109-42</td>
</tr>
<tr>
<td>80-133-54</td>
</tr>
<tr>
<td>114-169-64</td>
</tr>
<tr>
<td>160-200-74</td>
</tr>
<tr>
<td>228-212-86</td>
</tr>
<tr>
<td>320-256-100</td>
</tr>
<tr>
<td>456-304-120</td>
</tr>
<tr>
<td>640-356-144</td>
</tr>
</tbody>
</table>
There are a number of other ways to calculate the horsepower required for the prime mover. Some have been developed by manufacturers, others by individual oil companies. Table 3 shows examples of different nameplate horsepower calculations used by one manufacturer.

<table>
<thead>
<tr>
<th>Table 3: Alternative Nameplate Horsepower Calculations</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Conventional and Air Balanced Units</strong></td>
</tr>
<tr>
<td>Slow speed engines and high-slip electric motors</td>
</tr>
</tbody>
</table>
| \[
\text{HP}_{np} = \frac{\text{Depth} \times \text{PD}}{56000} \\
\] |
| Multi-cylinder engines and normal-slip electric motors |
| \[
\text{HP}_{np} = \frac{\text{Depth} \times \text{PD}}{45000} \\
\] |
| **Mark II Units**                                     |
| Slow speed engines and high-slip electric motors      |
| \[
\text{HP}_{np} = \frac{\text{Depth} \times \text{PD}}{56000} \times 0.8 \\
\] |
| Multi-cylinder engines and normal-slip electric motors |
| \[
\text{HP}_{np} = \frac{\text{Depth} \times \text{PD}}{45000} \\
\] |

**ROD STRESS CALCULATIONS**

The last calculation in the design procedure is made to ensure that the rod stress is not too high. Maximum rod stress is equal to the peak polished rod load divided by the cross-sectional area of the top rod:

\[
\text{Rod Stress} = \frac{\text{PPRL}}{\text{Rod Area}} \quad (14)
\]

In this design example, the rod stress exerted on the 7/8-in top rod is \(11,698 / 0.601\), or 19,464 psi. This value is well below the maximum allowable stress of 34,000 psi that was specified for the Grade D rods.
POLISHED ROD DYNAMOMETERS

Rod-pumped wells must be monitored to ensure the continued efficient and economic operations of a field. After a pumping unit has been installed in the field, dynamometer tests of pumping wells are conducted to determine the system efficiency, and whether adjustments need to be made in stroke length, pump speed or other operating parameters. These tests are designed to answer such well performance questions as (Podio et. al, 2001):

- Is the well pumped off?
- What is the pump intake pressure?
- What is the pump fillage?
- What is the pump displacement?
- Is the standing valve or traveling valve leaking?
- What is the effective pump plunger travel? What is the current pumping speed?
- What is the polished rod horsepower?
- Is the gearbox overloaded?
- Is the unit properly balanced?
- Is the downhole gas separator operating efficiently?

A dynamometer survey measures the load forces acting on a rod string during a complete pumping cycle and records the forces on a chart or computer display. This display is often called a dynamometer card. The card records changes in the rod load versus rod displacement, or changes in the rod load versus pumping time (Figure 1: Example of surface (top) and pump (bottom) dynamometer cards. Courtesy Echometer Company, 2003).

During a pumping cycle, forces acting on the rod string cause changes in the rod load. Measurements of these rod loads reflect the operation of the subsurface pump and the surface unit. Polished rod dynamometers record polished rod loads during the pumping cycle. Most common dynamometers make continuous plots of load measured against the movement of the polished rod. Typical
conventional dynamometers are either mechanical or hydraulic, but electronic dynamometers are also used. These dynamometers are briefly described below.

MECHANICAL DYNAMOMETER
A mechanical dynamometer measures rod loads by measuring the deflection of a steel ring placed between the carrier bar and the polished-rod clamp. A recording of this ring deflection is made on a paper dynamometer card, attached to a rotating drum driven by the vertical movement of the polished rod. The card traces polished-rod loads against rod vertical displacement. Its major disadvantage is that the pumping unit must be stopped before it can be installed on the polished rod.

HYDRAULIC DYNAMOMETER
A hydraulic dynamometer (Figure 2: Courtesy Echometer Company, 2003) can be installed while the pumping unit is running. This type of dynamometer uses a spacer installed on the polished rod between the carrier bar and the polished-rod clamp. As the unit is pumping, two load-sensing hydraulic pistons can be installed between the shoulder of the spacer and the carrier bar. The polished-rod load is transferred to the hydraulic pistons. Changes in polished-rod loads affect the hydraulic pressure. These changes in hydraulic pressure are recorded on a drum chart as changes in rod load versus rod displacement.

ELECTRONIC DYNAMOMETER
The electronic dynamometer uses electronic transducers to measure well loads and rod displacements (Figure 3: Polished rod dynamometer transducer. Courtesy Echometer Company, 2003).
An electronic dynamometer unit’s main components include a load transducer (load cell), a position transducer and the electronics that provide interfacing, signal recording, and processing. The load cell, located between the carrier bar and the polished-rod clamp, uses a strain gauge to measure polished rod loads. The position transducer uses a potentiometer to measure polished-rod displacement. Accelerometers determine rod displacement so that polished-rod load and position can be recorded as a function of time. In this manner, an electronic dynamometer can produce surface dynamometer cards and supply the information needed to construct downhole cards.

Portable electronic dynamometer systems usually include microcomputers that combine real-time data acquisition with easy data storage and retrieval operations, as well as performing on-line analysis of measurements and calculating downhole cards.

**DYNAMOMETER CARD INTERPRETATION**

(Figure 4) shows a dynamometer card shortly after it is recorded at the surface.

A number of measurements must be noted on the card.

- The first is C, the *calibration constant* in pounds per inch of card height. Each inch of height, then, refers to some calibrated load on the polished rod. This is determined in the field.
- D₁ is the *maximum deflection* in inches.
- D₂ is the *minimum deflection* in inches.
- CB is the *counterbalance line*, which is obtained by drawing a line through the dynamometer card at the point that represents the static load on the polished rod when the crank arm is horizontal. This means that the pumping unit is stopped and the load on the polished rod is measured with the crank arm at an angle of 90° or 270° from the 12 o’clock position.
• A₁ is the *lower area* of the card, measured in square inches—that is, it is the area between the lower boundary and the zero line.
• A₂ is the *area within the card* shape, measured in square inches.
• Finally, L is the *length* of the card, in inches, measured along the horizontal axis from one end of the dynamometer shape to the other.
• The left-hand extreme of the card is the bottom of the stroke, and the right-hand extreme is the top of the stroke.

With a dynamometer card for a given rod pump and the definitions above, we can make the diagnostic calculations listed in Table 1:

<table>
<thead>
<tr>
<th>Table 1: Diagnostic Calculations for Beam Pumping System</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Peak polished rod load</strong></td>
</tr>
<tr>
<td><strong>Minimum polished rod load</strong></td>
</tr>
<tr>
<td><strong>Polished rod load range</strong></td>
</tr>
<tr>
<td><strong>Counterbalance effect</strong></td>
</tr>
<tr>
<td><strong>Approximate CBE</strong></td>
</tr>
<tr>
<td><strong>Polished rod horsepower</strong></td>
</tr>
</tbody>
</table>

The final term to be calculated is the *peak torque* that is exerted on the gear box under operating conditions. Normally this calculation is made using a term referred to as the *torque factor*, TF. The torque factor is equal to the torque that must be applied at the crankshaft to offset a one pound weight hanging on the horsehead. Thus, if a 25 ft-pound torque must be applied at the crankshaft to offset a one-pound weight hung on the horsehead, the torque factor is 25. The torque factor will vary with different crank angles.

With a value for TF, the *net torque* calculation, which is a function of the crank angle, can be made by using the following equation:

Net torque = TF (W-B) - M Sine q  \hspace{1cm} (Equation 15)

where: W = well load for a given crank angle, pounds
B = structural imbalance of pump unit, pounds (supplied by manufacturer)
N = torque provided by the counterweights, ft-pounds
TF = torque factor
q = crank angle

Using this equation, we can look at the dynamometer card, insert the crank angle on it for a number of different values of W and TF, and make net torque calculations using the above equation. Note that W is a function of the crank angle and is measured from the dynamometer card. The torque factor, TF, is supplied by the manufacturer for a given unit in 150 angles of the crankshaft.

This net torque calculation should provide a plot similar to that shown in the dashed curve of (Figure 5), showing the *peak torque* at the maximum point on the curve. It suggests that the counterweights should be adjusted in order to even out the net torque curve. After an adjustment of the counterweights, for example, a second curve is obtained (solid curve), showing that the unit is now more balanced.
The downhole condition of the pump and its pumping action can be analyzed with a dynagraph. With the dynamometer, those terms that affect the surface operating condition can be calculated and necessary adjustments to the unit can be made. If the operation of the subsurface pump and the surface unit are satisfactory, is it possible to improve production by changing the pump stroke length, its speed or its surface control system? Generally, a unit that runs intermittently is controlled either by a timer or with a pump-off controller. Careful adjustments in the field are required to maximize production, without damaging the pumping unit. The information from dynamometer tests taken both before and after changes are made in the pumping equipment will help determine whether the adjustments have improved the operating conditions.
ROD LOAD VERSUS DISPLACEMENT

To understand the diagnostic procedures involved in monitoring a beam pumping system, we must first understand the principles of dynamometry. Gibbs and Neely (1966) discuss polished-rod load versus displacement for anchored and unanchored tubing. Typical plots for the unanchored and anchored tubing case are shown in the next series of figures. These plots are developed systematically.

UNANCHORED TUBING

(Figure 1) represents the rod load as a function of displacement for unanchored tubing. Note that the load scale goes as high as 10,000 pounds, and that the pump displacement (in feet) has a coordinate system that runs from a point above the top of the subsurface pump down to the standing valve. When the plunger is at its lowest point, it carries no load; its displacement is 9 feet from the coordinate to the ground coordinate system, but it is stationary with respect to the stretching tubing.

In (Figure 2), as the traveling valve reaches point “C”, the entire fluid load is supported by the rods, until it begins to transfer to the tubing at point “D.”
In (Figure 3), as the traveling valve reaches point "E", the transfer of the load from the rods to the tubing is almost complete, but the tubing continues to stretch somewhat. At point "F", the entire fluid load is now supported by the tubing, causing the tubing to stretch to its greatest length. The plunger now descends with respect to both the ground and tubing coordinate systems. At this point, there is no load on the traveling valve and it continues to descend until it reaches a displacement position, indicated by point "A". The cycle then repeats.
ANCHORED TUBING

If the tubing is anchored, it will not stretch or contract in response to the movement of the pump, and the "ideal" load-displacement diagram will have the appearance shown in (Figure 4) (Ideal load-displacement diagram for a pump with anchored tubing).

![Ideal load-displacement diagram for a pump with anchored tubing](image)

PUMP VALVE DYNAMOMETERS

Efficient operation of a sucker-rod pump depends on the integrity of the standing and traveling valves. If these valves lose their ability to hold a seal, the lifting capacity of the pump is affected. Therefore, the valves should be tested periodically with a dynamometer.

TESTING THE STANDING VALVE

Dynamometer testing of the standing valve requires that pumping unit to be stopped near the bottom of the downstroke. The initial polished-rod load is recorded on the dynamometer chart. Since the fluid load is completely carried by the standing valve, the polished-rod load recorded at the start of the test will measure only the buoyant weight of the rod string.

When the standing valve holds a good seal, the polished-rod load will remain steady, resulting in a line on the dynamometer card that falls on the first measurement. If there is a leak in the valve seat, the pressure immediately below the traveling valve will decrease, then slowly close. As the test proceeds, the fluid load initially carried by the standing valve will move to the traveling valve. This movement increases the load of the polished rod, and will be recorded on the dynamometer chart. The rate of load increase is directly proportional to the severity of the leak in the standing valve and it is not uncommon for a leaking standing valve to show a load increase in about 20 seconds.

This test should be repeated. If the same results are obtained, then the test indicates that the seat of the standing valve is cut. If the valve holds during some of the tests, then a damaged valve ball is indicated.
TESTING THE TRAVELING VALVE

Testing the traveling valve will help to distinguish whether the traveling valve or the barrel-plunger fit is leaking. To start the test, the pumping unit is stopped near the top of the upstroke and the initial polished-rod load on the dynamometer is recorded.

The initial recorded polished-rod load measures the buoyant weight of the rod string and the fluid load acting on the plunger. The standing valve is open so it carries no load. If the seal in the traveling valve is good, and additionally, the pump barrel and plunger fit is good, then no change in polished-rod load should be observed over the test period.

A leak either in the traveling valve or between the pump barrel and plunger will allow fluid to pass through the traveling valve. This leak will slowly force the standing valve to close. Once closed, the fluid load transfers to the standing valve and the tubing, and this load transfer will be observed on the dynamometer card. The initial polished-rod load (rod-string weight plus fluid load) recorded on the card will be greater than the loads measured later in the test. The rate of load decrease is proportional to the leak rate.

This test cannot distinguish between a leaking traveling valve and leaking caused by a worn barrel or plunger. This test should be repeated at different points in the upstroke. Load differences observed at different depths can help to identify the location of the leak.

MEASURING THE CBE

A dynamometer can also measure the counterbalance effect (CBE) that is experienced when a pumping unit is overbalanced, or "weight heavy". This condition occurs when the counterweights move down and lift the polished rod after the motor is stopped and the brake is released. If the pumping unit is underbalanced, or "rod heavy", the polished rod will move down and lift the counterweights after the unit is stopped.

The CBE is measured with the dynamometer in place, carrying the full polished-rod-load. After the motor is stopped, the brake is used to stop the pumping unit with the crank angle at 90° or 270°, so that the torque exerted on the crankshaft counterweights is at a maximum. The vertical force on the polished rod resulting from this torque is the counterbalance effect.

The CBE value measured above is useful for estimating the maximum torque.

Pumping units with non-conventional geometries have counterweights attached to arms that are set at angles offset to the cranks. The maximum counterbalance torque is exerted when the counterweight arms, rather than the cranks, are in a horizontal position. However, the API recommends that CBE be determined with the cranks held horizontal during the measurement.
SYSTEM MONITORING: DYNAGRAPHS

Surface dynamometer cards cannot be used to directly measure the operations of a downhole pump, because the surface measurements record all the static and dynamic forces acting on the rod string. We can make a more accurate evaluation of the pump by placing a dynamometer just above it. Measurements of rod loads recorded immediately above the pump are called dynagraph cards, to distinguish them from surface-recorded dynamometer cards.

To illustrate the use of dynagraph cards, consider the idealized typical profiles shown in the following figures.

- In (Figure 1), a small, gradual curvature on the dynagraph during the upstroke indicates that a small amount of gas, perhaps coming out of solution, is being compressed during the upstroke. As the pump begins its downward stroke, the load is not entirely transmitted to the tubing due to a period of gas compression on the downstroke before the traveling valve opens.

- In (Figure 2), the fall-off in load is sharper during the downstroke than for the gas compression dynagraph of Figure 1. This type of response generally means that the well is pumped-off or almost pumped-off. In this condition, the pump chamber doesn't completely fill with fluid. As the plunger moves down and compresses the gas, the pressure below the traveling valve is not high enough to prevent the plunger from striking the liquid interface in the lower part of the chamber. The shock that results when the plunger hits the fluid interface and releases the load, is called fluid pound. Often, fluid pound can be felt at the surface, indicating that the well is pumped-off. Dynagraph cards recorded on wells with gas compression look similar to wells that experience fluid pound, and it can be difficult to distinguish between the two.
• (Figure 3) is a dynagraph profile indicating low pump efficiency. The pump is “gas locked” and has no fluid in its chamber. The pump is simply causing gas to compress and then expand.

![Dynagraph Profile](image)

• (Figure 4) indicates a leaking traveling valve. Movement during the upstroke is required before a full load is taken on and, because the traveling valve is leaking, the rods begin to lose load before the pump has reached the top of the upstroke.

![Dynagraph Profile](image)

• (Figure 5) indicates a leaking standing valve. The leak causes premature loading of the rods before the beginning of the upstroke, and also delays the unloading of the rods during the downstroke, thus the traveling valve closes during the downstroke, causing it to bear some of the load.

![Dynagraph Profile](image)
If (Figure 4) and (Figure 5) were combined, the results would look like (Figure 6). This dynagraph indicates a worn-out pump that is leaking at the traveling valve, at the standing valve, and probably has excessive plunger slippage, too. This pump should be replaced.

SYNTHETIC DYNAGRAPHS

The dynagraph is a valuable diagnostic tool for analyzing pump performance. Unfortunately, because it is located at the pump depth, this analysis may be difficult to obtain. Alternative procedures include:

- Using dynamometer cards of load displacements measured at the surface to interpret what is happening in the subsurface, and
- Using surface measurements and procedures (Gibbs, 1963 and Gibbs and Neely, 1966), to develop mathematically derived dynagraphs called synthetic dynagraphs (often referred to as pump cards).

In the latter case, Gibbs and Neely made it possible to measure loads and displacements at the surface and then, to calculate the required synthetic dynagraph by solving the appropriate non-linear partial differential equations.

Obtaining a synthetic dynagraph is fairly straightforward. The polished rod load versus time (Figure 7) and the polished rod displacement versus time (Figure 8) during a pumping cycle are measured.

(After Gibbs and Neely, Copyright 1966 SPE-AIME)
These two graphs are then combined to give a polished rod load versus polished rod displacement profile. Using the data collected at the surface and solving the appropriate partial differential equations, the measured surface information can be transformed to yield the load-displacement profiles at any point along the rod-pump system. Gibbs and Neely showed the load-displacement diagram at the two points along a tapered rod string where the rod sizes changed (Figure 9).

The first such chart is at a depth of 1,475 feet. The transformation of the dynamometer card to this depth shifts the position of the curve to lower loads and shorter displacements as expected.

- The second transformation shows a load-displacement at a depth of 3,125 feet.
- The third transformation is made for the pump depth of 8,525 feet. This, in essence, is the synthetic dynagraph.

In (Figure 10), the scale of the dynagraph has been expanded to show that the fluid load is 3,200 pounds. The buoyant force on the rods is given by the magnitude of the negative load on the dynagraph. Here, it is 1,400 pounds.
The displacement dimensions show that the gross pump stroke is 7.1 feet, but that the net liquid stroke is 4.6 feet. This means that the pumping efficiency, based on the relative displacements, is approximately 65% (4.6 feet / 7.1 feet) and that some gas compression is occurring on the downstroke. It also shows that the tubing anchor is working well to hold the tubing during the upstroke. With knowledge of the pump diameter and the pump speed, the displacement at the pump is calculated to be 200 B/D. The surface production rate was measured at 184 B/D and so no serious tubing leaks are evident.

With the dynamometer cards calculated at the top of each tapered rod section the load on the top rod of each section can be measured; dividing that load by the rod cross-sectional area gives an estimate of the respective rod stress. Doing this yields a peak rod stress of about 30,000 psi, a value that is below rod limitations. The dynamometer card is used for the balance of the analysis.

When synthetic dynagraphs are not used, an alternative is to use the dynamometer card to interpret both the surface and subsurface operation of the pumping system. As a guide to this type of interpretation API Bulletin 11L2 (1969; reaffirmed Sept. 1999) provides an extensive set of typical dynamometer cards, categorized according to the dimensionless variables used in the pump design procedures.