New Developments In Rod Pumping

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Over 400,000 oil wells are now produced by rod drawn pumps. These vary in depths from "post holes" of less than a hundred feet to some of the deepest wells drilled, where the setting depth of the pump may be as much as 12,000 feet. The volume output of these pumps is just as varied as their setting depth. It may range from less than a barrel a day in small stripper wells to more than 2,000 barrels a day in wells where large bore pumps and high operating speeds are employed.

In addition to the wide variations in setting depth and volume requirements, oil wells vary widely in the character of the fluid they produce. Prime considerations are abrasion, and corrosion, and the amount of lubrication the well fluid is able to supply to the wearing parts of the pump.

New developments have naturally been directed toward a pump design that will extend both the depth and volume limits of rod drawn pumps. They have also been directed toward longer pump life through the use of corrosion and abrasion resistant alloys.

Progress has been rapid. Metallurgical advances have made available new materials that provide an excellent combination of abrasion and corrosion resistance in barrels of composite or full barrel construction. Several companies now offer premium quality liners fabricated by centrifugally casting a special alloy inside a steel supporting sleeve. One company shrinks a steel sleeve over a thin cylinder of stainless steel. These liners are then retained inside a steel jacket.

One company has developed a unique method of permanently fixing the liners in the jacket, since instead of shrinking the jacket onto the liners, the liners are caused to grow inside of the jacket until an interference fit results. The permanent growth in the liners is brought about by application of sub-zero temperatures to the assembled barrel. These low temperatures alter the nickel-iron crystalline structure of the liners sufficiently that a permanent increase in diameter of several thousandths of an inch is effected.

The ceramic manufacturers are producing a material especially processed for this application. It has excellent abrasion and corrosion resistance, and properly supported in the barrel, adequate mechanical properties. The high cost of finishing the ceramic liner has limited its application, but in wells where corrosion and abrasion are extreme, its high initial cost can be justified.

One development in downwell pumps that is not new, but is just now gaining wide acceptance is the use of tungsten carbide seats and balls in the valve units in pumps. Actually, the carbide used is more properly called a "cemented carbide" or a "scintered carbide."

The cemented or scintered carbide is a man made metal. Pure tungsten carbide is an extremely hard crystalline solid, only slightly softer than a diamond. Unfortunately, it has very low mechanical strength and is rarely used in its pure form. The science of powder metallurgy has, however, provided a method of producing a material which utilizes the extreme hardness of the tungsten carbide, but still exhibits good mechanical properties. Briefly, this is done by scintering an intimate mixture of pulverized tungsten carbide and cobalt. The cobalt becomes the "cement" and the tungsten carbide the "aggregate" hence the term cemented carbide. The material used in seats and balls is approximately 88 percent tungsten carbide and 12 percent cobalt binder.

To better understand how this super hard, man created metal solves valve problems in downwell pumps, let's analyze the three primary causes of failures and see how the carbide can combat each of these causes. The first is a cut or fluid washed ball and seat. This failure occurs most frequently in pumping abrasive fluid. Since the carbide ball is capable of crushing or pulverising grains of sand on the carbide seating surface without deforming or wearing the surface, the initial flow, which can cause the fluid cut is never formed.

The second is a battered seat, beaten down and deformed from heavy ball action. This is observed most frequently in the corrosion resistant type of valves where the seat is necessarily somewhat softer than a stainless seat. The carbide seat cannot deform under ball pound.

The third common cause of valve failure is corrosive attack. Naturally, a corroded pit in a seating surface can start a fluid wash as readily as a dent pounded into it by a grain of sand, the seat and ball material must have adequate corrosion resistance to prevent pit formation. It is a happy coincidence that cemented tungsten carbide, in addition to its extreme resistance to abrasion, possesses a superb resistance to corrosion. Therefore, a cemented carbide ball and seat can operate successfully in a heavy running well pumping corrosive and abrasive fluid.

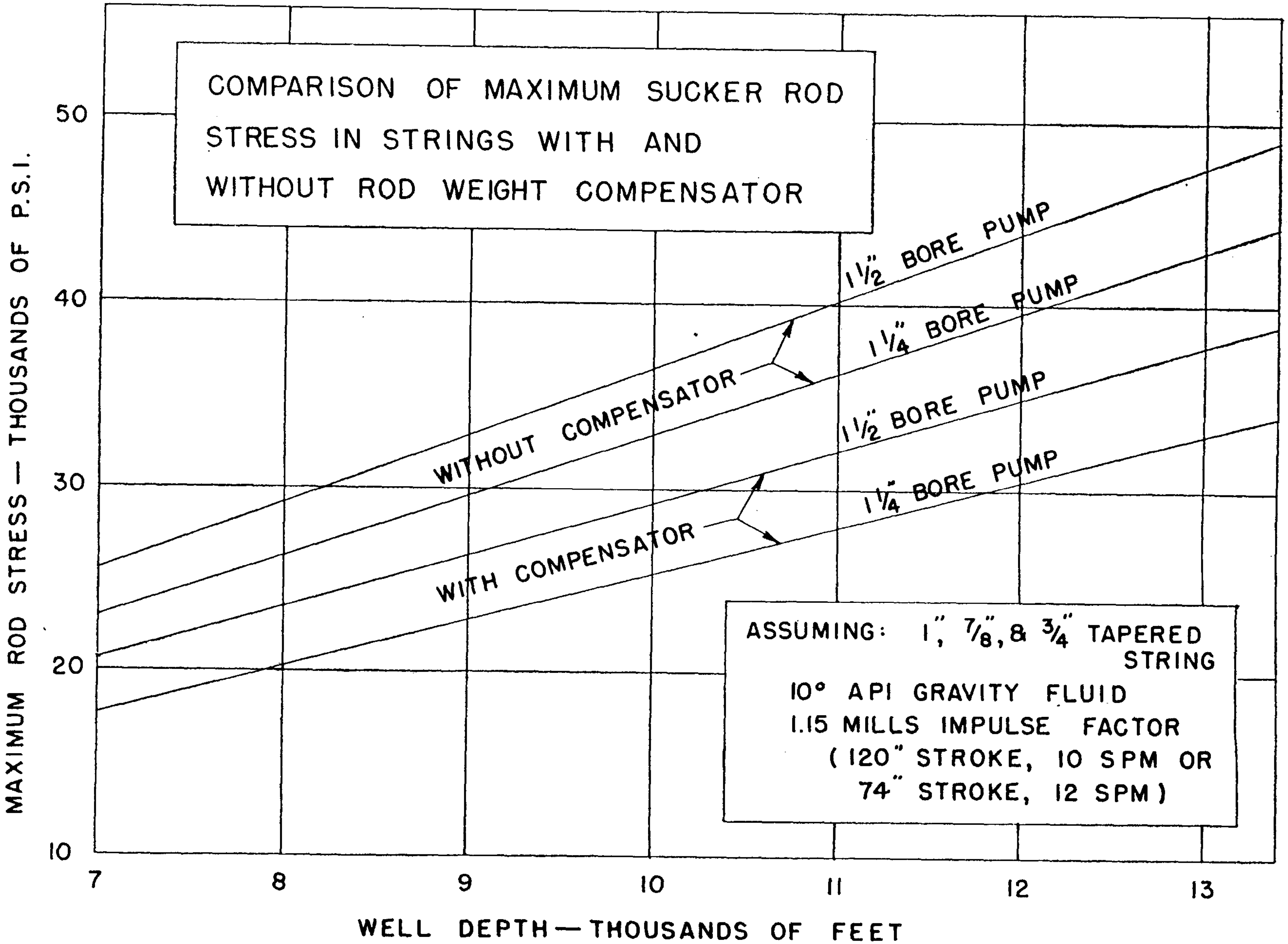
The above information points strongly toward carbide valves in pumps, and the question logically arises, "Why not use them in all pumps?" Actually, the factor that limits the use of tungsten carbide is its high cost. A carbide valve unit costs approximately ten times what a comparable stainless steel unit costs. Therefore, when carbile valves are specified, the production man must be in a position to justify this increased expense. Obviously, if he is having no trouble with his present valves, he cannot justify the additional expense of carbide valves. No overall statistics are available, but prior to the introduction of carbide, most production men agreed that twenty to forty percent of pump failures were caused by cut valves. Use of tungsten carbide balls and seats will almost eliminate these failures. This decreases the number of pulling jobs and consequently the pulling cost. It will also permit the pump to operate, in most cases, until the full wear life has been received from the barrel and plunger, reducing the total pump repair expense charged against the lease.

Progress in rod pumping has not been confined to the pumps. The use of longer stroke surface units is a very apparent trend. Ten foot stroke beam units, almost unknown ten years ago, are quite common today. Hydraulic pumping jacks have provided stroke lengths up to thirty feet for wells where load reversals and losses due to rod stretch must be held to a minimum.

An interesting new development is the hollow sucker rod. These are offered by a number of manufacturers with standard 7/8" API sucker rod connections. These connections are either flash welded or forged onto steel tubing which is 5/8" inside diameter by 1-1/8" outside diameter. Hollow rods use standard 7/8" rod couplings and may be run into and removed from the well with standard 7/8" rod tools in the same fashion as regular sucker rods.

There may be a number of useful applications for hollow rods, however, at present they are used mainly to combat paraffin accumulations. In this application, they replace the portion of the rod string where paraffin normally accumulates and are connected to the lower part of the rod string by a perforated coupling. By the use of a hollow polished rod, hot oil can be pumped through the hollow sucker rods effectively

DIAGRAM I


COMPARISON OF MAXIMUM SUCKER ROD STRESS IN STRINGS WITH AND WITHOUT ROD WEIGHT COMPENSATOR
MAXIMUM ROD STRESS — THOUSANDS OF P.S.I.
10
20
30
40
50
7
8
9
10
11
12
13
WELL DEPTH — THOUSANDS OF FEET
1 1/2" BORE PUMP
1 1/4" BORE PUMP
WITHOUT COMPENSATOR
1 1/2" BORE PUMP
1 1/4" BORE PUMP
WITH COMPENSATOR
ASSUMING: 1", 7/8", & 3/4" TAPERED STRING
10° API GRAVITY FLUID
1.15 MILLS IMPULSE FACTOR
(120" STROKE, 10 SPM OR
74" STROKE, 12 SPM)

SCHEMATIC DRAWING OF OILMASTER ROD WEIGHT COMPENSATOR

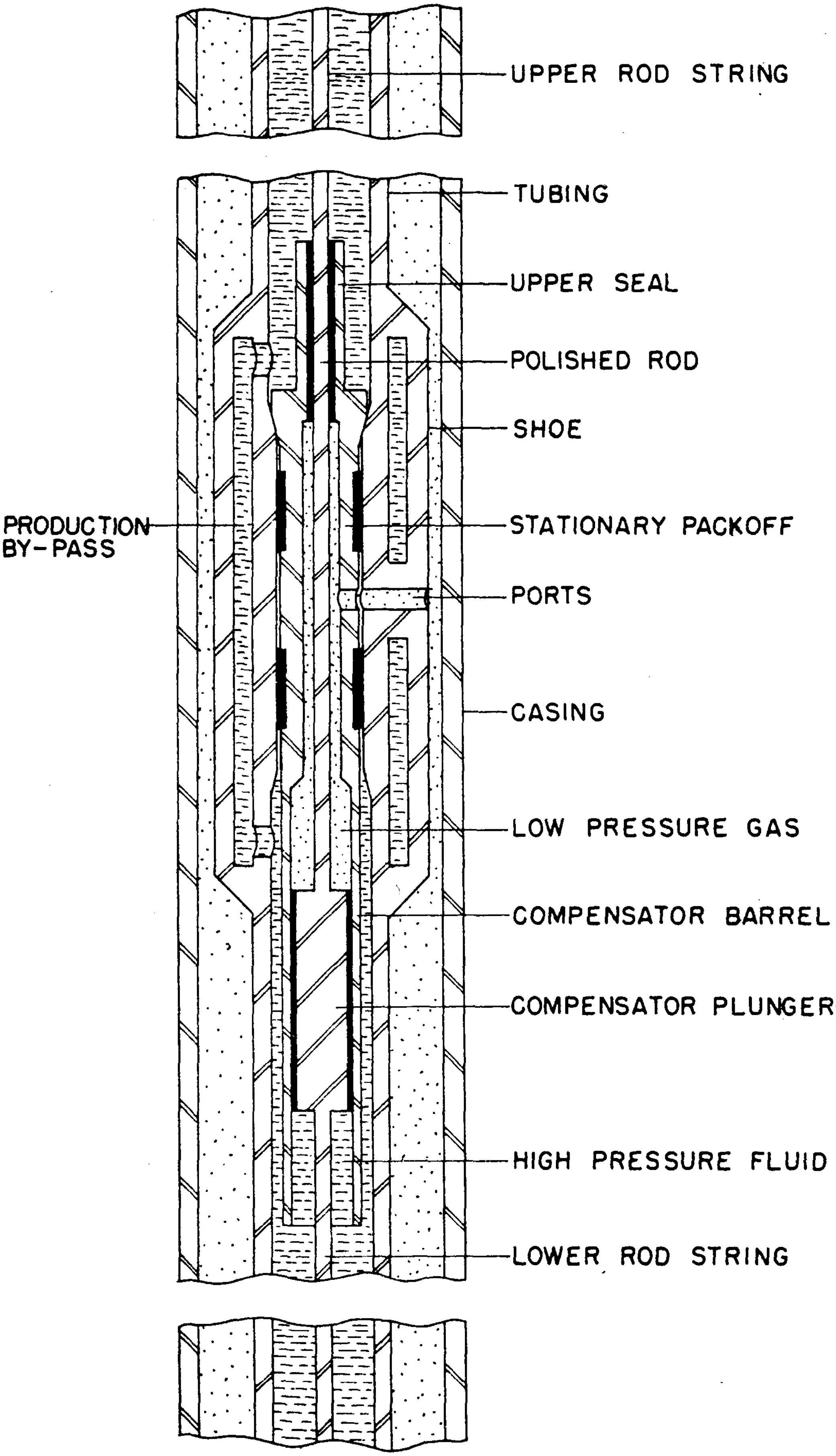


DIAGRAM II

melting the paraffin from the annular space between the hollow rods and the tubing.

Another advance that should ease the burden on an operator who is seating a pump in a tubing string not equipped with a shoe is a new type of insert pump anchor that sets by vertical motion without rotating the rods. This anchor is actually a small hookwall packer-anchor designed to set anywhere in the tubing string. A dowel pin on the slip assembly works in slots milled in the anchor body. There is a long setting slot and a slightly shorter running slot interconnected at the bottom by a diagonal transfer slot. The running slot permits a normal upward movement of the rods of two feet, but any movement past this will rotate the pin into the setting slot, and a subsequent downward movement of the rods will set the slips and then the packer.

A second V shaped slot is provided to permit the dowel pin to be returned to the running slot if it is desirable for any reason to move and reset the anchor. Thus, this anuchor operates like any "J" slot anchor, but transfers of the pin from one slot to the other is done by vertical movements, easily accomplished with the hoist, instead of rotational movement requiring use of wrenches and entailing the hazard of unscrewing a joint.

There is one development we would like to describe to you in some detail, not because it has a particularly wide application, but because it does extend the limits of rod pumping. This is a device called the Rod Weight Compensator.

Actually, it is a hydraulic counterbalance. However, instead of locating it at the surface where the pumping unit has its own counterbalance, the Compensator is located at about half the depth of the well. By setting it here, it balances out most of the weight of the lower part of the rod string, and relieves the upper part of the string of this dead load.

Let's take an example to show what the Compensator will do; then we will describe how it works. Since the Compensator is specifically designed for deep wells, let us assume a 10,000 foot well. A properly tapered string of 1", 7/8", and 3/4" rods in this well would have a dry weight of about 20,000 pounds, or submerged in oil would weigh about 18,000 pounds. The total polished rod load cannot exceed 23,600 pounds without exceeding 30,000 pounds per square inch stress on the top 1" rod. Therefore, if it is desirable to operate at a stress lower than 30,000 psi, the entire fluid and impulse load would have to be less than the difference between 23,600 pound maximum allowable load and the 18,000 pound wet weight of rods. This would leave 5,600 pounds which would just about permit running a 1-1/4" bore pump at five ten-foot strokes per minute without exceeding the allowable stress.

In the same well, a Rod Weight Compensator could be set at a depth of 5,900 feet. Here, it would have a counterbalance effect of 5,000 pounds. Therefore, all of the rods above the Compensator would be relieved of the dead weight of 5,000 pounds of rods on both the upstroke and downstroke. The top 1" rod would be carrying a dead weight of 13,000 pounds instead of 18,000 pounds. Therefore, the total fluid and impulse load could be increased from 5,600 pounds to 10,600 pounds before the 30,000 psi maximum stress was exceeded. This 10,600 pounds would permit both a larger bore pump and an increased impulse factor. For example, a 1-1/2" bore pump could be operated at ten ten-foot strokes per minute without exceeding the allowable stress. The combination of the larger bore pump and the higher operating speed would permit approximately three times the fluid production.

The Compensator accomplishes this down the hole counterbalancing hydraulically. In the 10,000 foot well in the example, the hydrostatic pressure of 30 degrees gravity oil in the tubing string at a depth of 5,900 feet would be about 2240 psi. Let us assume the casing pressure is 40 psi, leaving an effective differential pressure of 2200 psi.

The Compensator is arranged so that this difference in pressure is applied across a 2" bore pump plunger. The high pressure fluid in the tubing applies a powerful upward force on the bottom of this plunger while the low pressure gas in the casing applies relatively little back pressure on the top plunger. Since the net area of the plunger is 2.26 square inches, this differential pressure of 2200 psi applies a lifting force of 5,000 pounds on the sucker rod string. Since the wet weight of rods below the Compensator is more than 6,000 pounds, the rod string still operates in tension, but at a greatly decreased load.

Now let's get into the design of the Compensator to see how the low pressure area is maintained above the plunger. First there is a shoe that is inserted in the tubing string at the depth at which the Compensator is installed. This shoe provides liberal by-pass areas so that fluid can flow around the Compensator without obstruction, but it also contains ports, which admit low pressure casing gas to the top of the Compensator plunger.

The balance of the Compensator is contained in the insert assembly which is run in and removed from the well with the sucker rod string. This insert assembly contains the Compensator barrel and plunger, the stationary packoff, and the upper seal. The Compensator barrel and plunger are the standard barrel and plunger assembly used in a 2" bore insert pump.

The upper seal consists of a long, chrome plated polished rod, accurately ground and fitted to a four foot, 1-1/16" bore pump barrel. The upper end of this rod connects to the bottom of the upper rod string, while the lower end passes through the stationary packoff and connects to the top of the Compensator plunger. The lower end of the Compensator plunger connects to the top of the lower rod string. The polished rod and plunger move with the rod string, while the balance of the assembly remains stationary in the shoe.

The stationary packoff is very much like a cup holddown for a pump except that it is designed to pack off both above and below the low pressure ports. The top of the packoff contains the standard "no-go" ring which furnishes the shoulder for seating the entire assembly.

Reference to the diagram will show how the Compensator operates. The rod pump at the bottom of the tubing string discharges fluid in the normal fashion. This fluid passes around the Compensator without restriction. However, the high pressure due to the hydrostatic head of fluid in the tubing is applied to the bottom of the Compensator plunger. The low pressure admitted through the ports, but isolated from the high pressure by the packoff and the upper seal acts on the top of the plunger. The difference between these two pressures is applying a lifting or counterbalancing force on the sucker rod string.

The question logically arises, "How much extra work is it to run a Compensator in a well." In an installation already in place, it will be necessary to pull tubing to the point where the Compensator shoe is inserted. Then the downwell pump is run in and proper spacing between the Compensator and pump is obtained by use of pony rods. The mandrel of an on and off tool or a rod hook on attachment is fastened above the Compensator, and the balance of the tubing is run in and landed. The upper half of the hook on is run in with the upper rod string, and this engages the mandrel when contact is made. Proper spacing between the Compensator and the pump need be established only once, and on a subsequent pulling job, the hook on attachment may be removed.

How does the reduction in rod weight accomplished by the Compensator pay off? Principally in two ways. First, by reducing rod weight it permits an increased fluid load and thereby an increased production without increasing rod stress. Or second, by reducing rod weight, it permits an equal fluid load but an increased life expectancy on the sucker rod string.