

April 6, 1937.

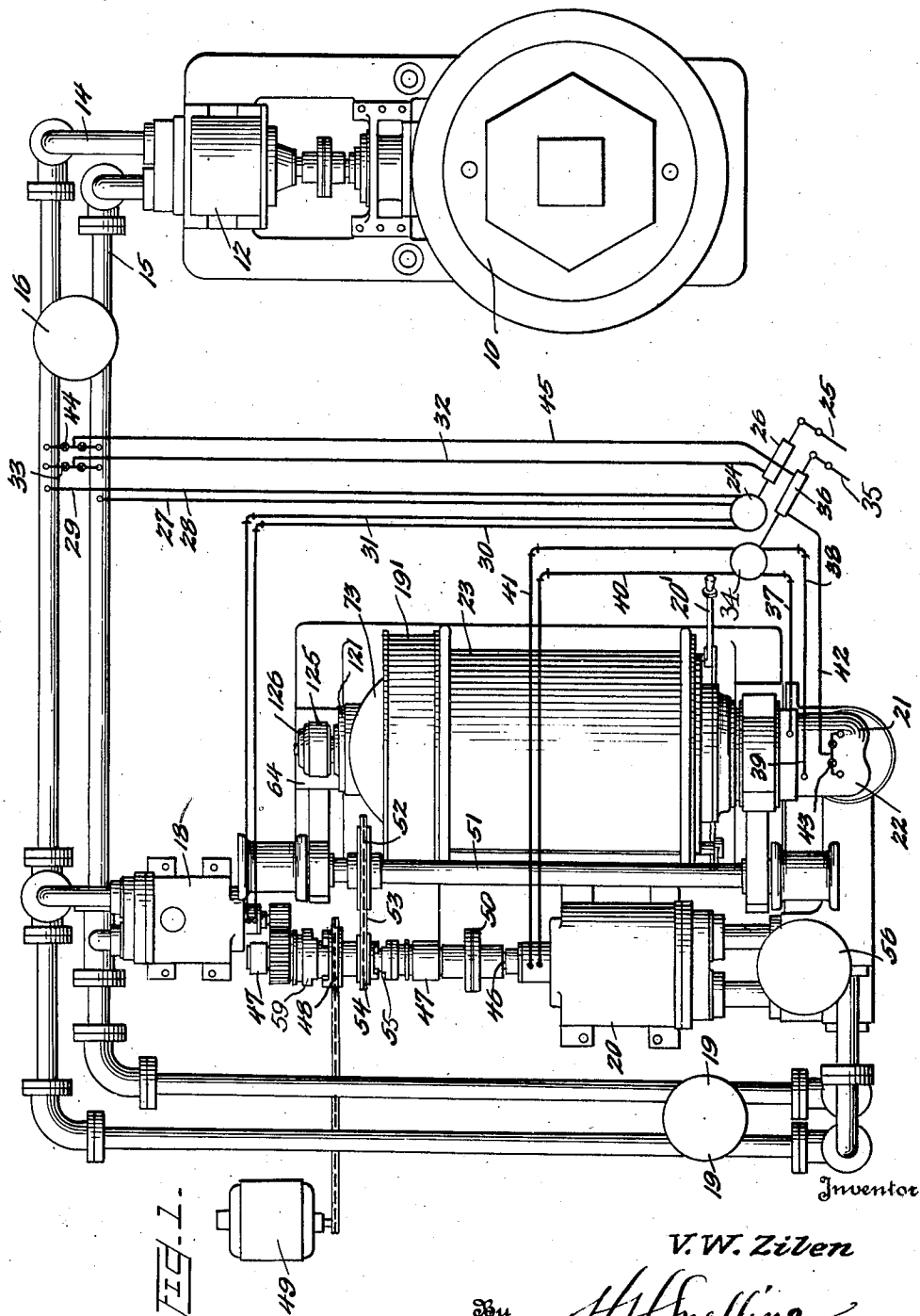
V. W. ZILEN

2,076,138

ROTARY DRILLING MECHANISM

Filed June 16, 1931

5 Sheets-Sheet 1



Inventor

V. W. Zilen

H. H. Snelling

Attorney

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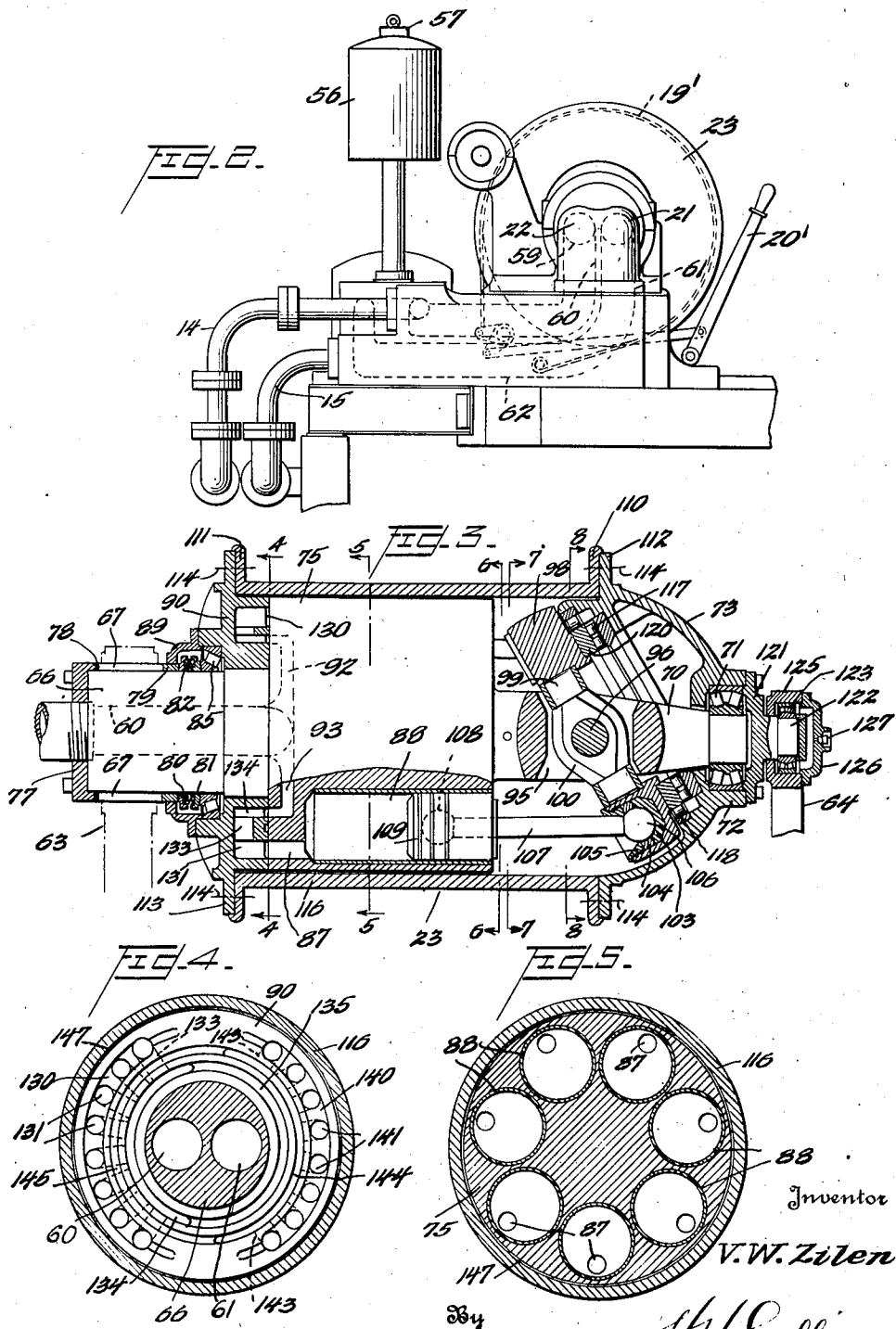
V. W. ZILEN

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ROTARY DRILLING MECHANISM

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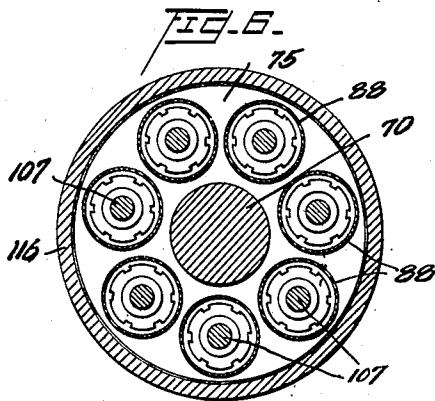
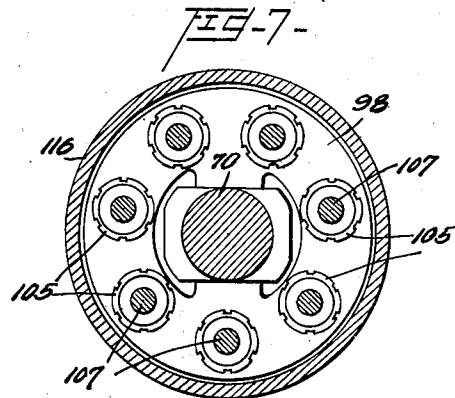
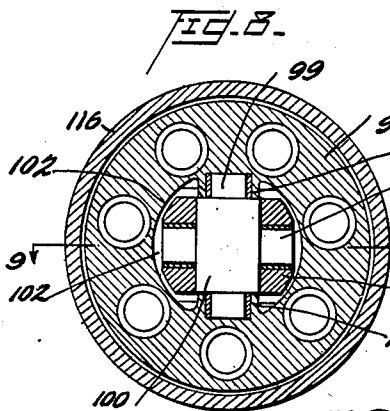
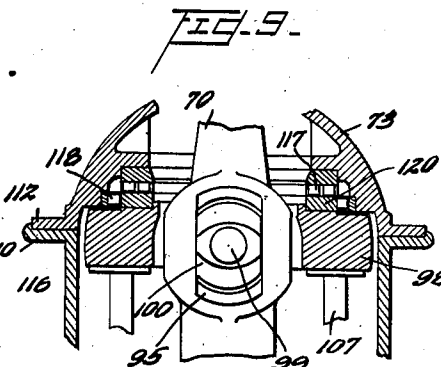
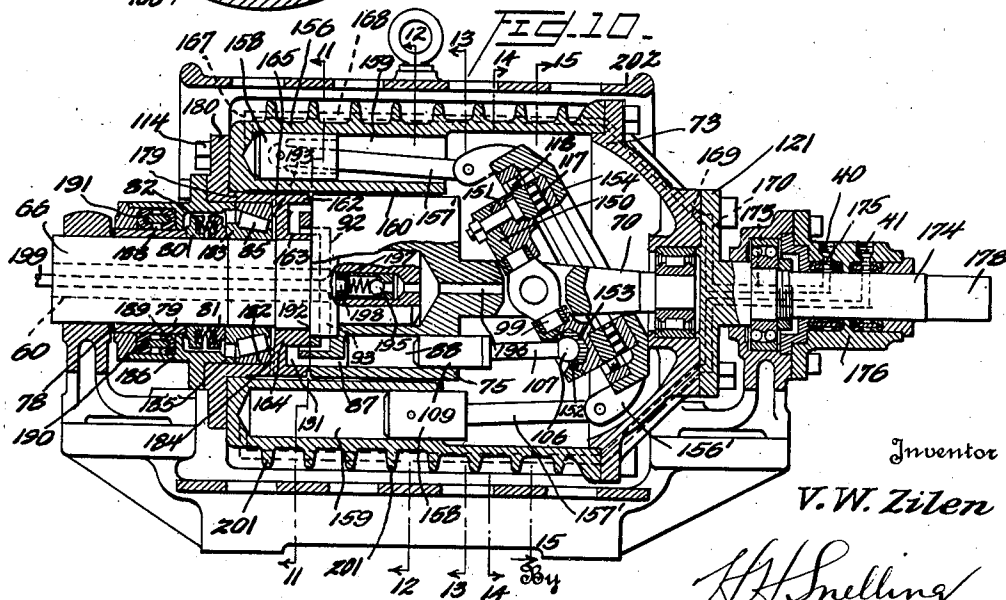
V. W. ZILEN

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ROTARY DRILLING MECHANISM

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5 Sheets-Sheet 3



Inventor

V. W. Zilen

H. H. Snelling
Attorney

April 6, 1937.

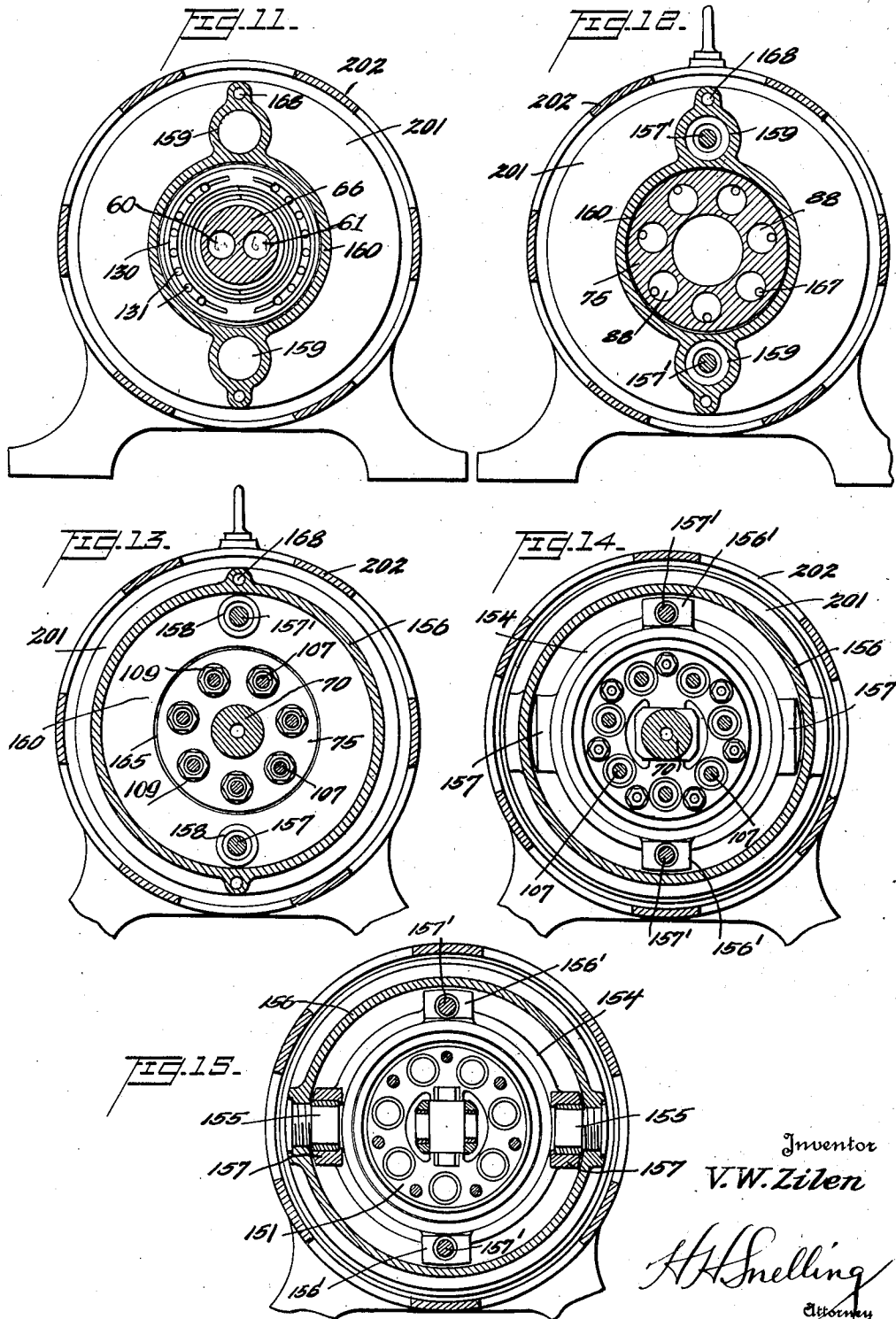
V. W. ZILEN

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ROTARY DRILLING MECHANISM.

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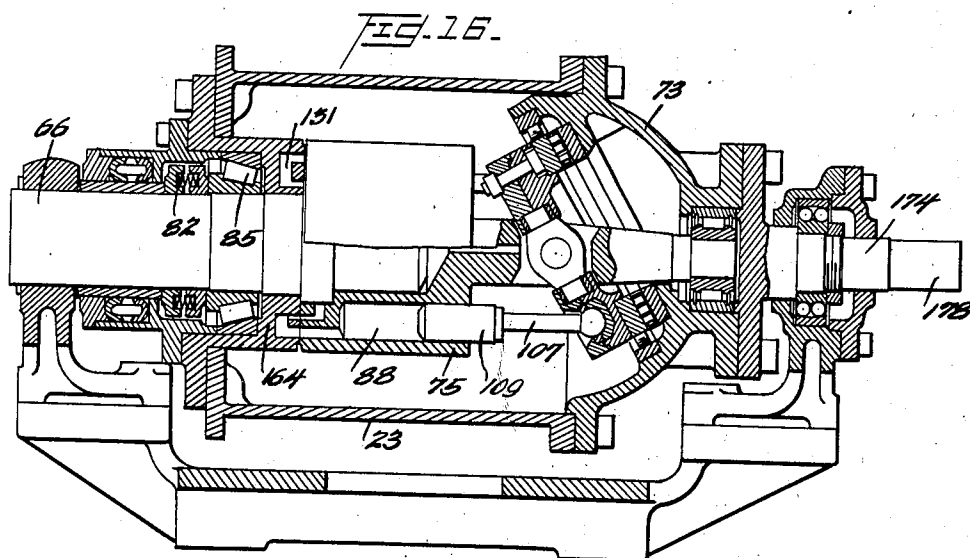
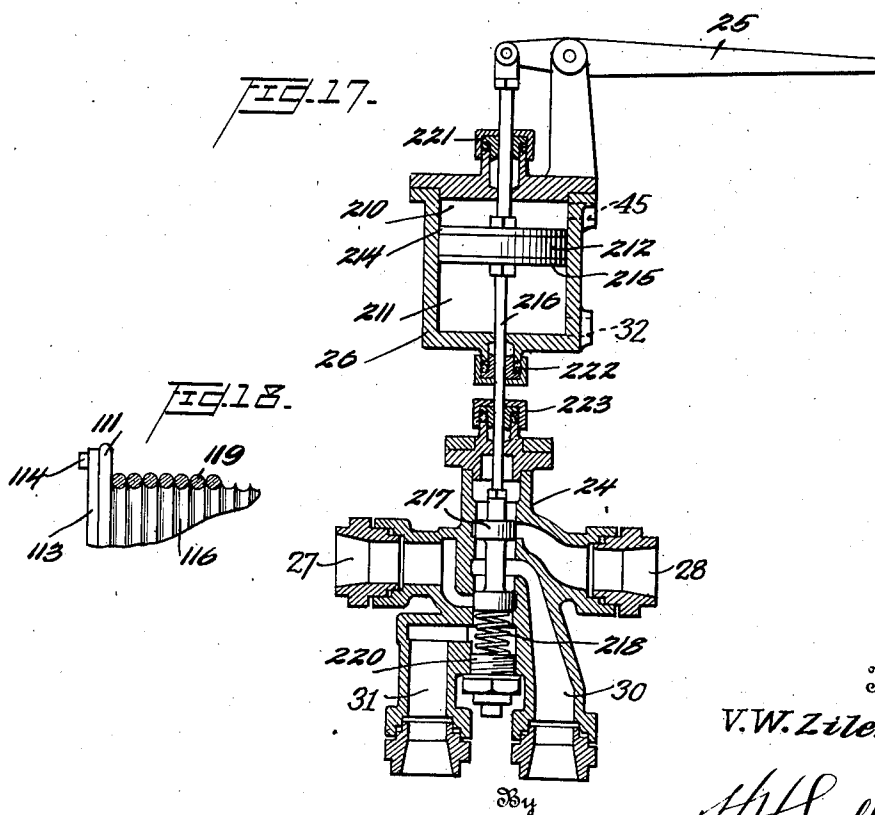
V. W. ZILEN

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ROTARY DRILLING MECHANISM

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5 Sheets-Sheet 5



Inventor

V. W. Zilen,

Attorney

UNITED STATES PATENT OFFICE

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ROTARY DRILLING MECHANISM

Victor W. Zilen, Buffalo, N. Y.

Application June 16, 1931, Serial No. 544,862

20 Claims. (Cl. 254—186)

This invention relates to drilling and has for its principal object the provision of apparatus for rotating a drill pipe while suspending the drill stem by hoisting means connected with the rotating means, both being hydraulic, whereby the changes in resistance encountered by the drill bit are reflected by changes in pressure in the connecting pipe lines with an instant raising or lowering of the drill stem by slight angular movement of the drum of the hoisting means.

An important object of the present invention is the provision of a hydraulic variable speed transmission embodying two separate units, a variable delivery fluid pump of the tilting box type and a variable speed hydraulic motor, in one and preferably in both of which the outer casing or shell which heretofore according to usual practice has been stationary, is a revolving drum mounted on a substantially stationary multiple cylinder barrel and shaft thru which stationary members the propelling fluid is delivered to the cylinders thru a distributing plate attached to and revolving with the drum.

Another important object is to eliminate the pressure between the rubbing surfaces of the piston and cylinder walls due to centrifugal action which is considerable at high speeds in rotary barrel multiple cylinder pumps and motors. I accomplish this by making the barrel stationary thus eliminating all but the reciprocating motion in the pistons and in order to get the necessary relative turning motion between the barrel and shell I allow the latter to turn.

A further object is to eliminate all gearing between the motor and the work by providing the revolving shell with various means so that it can be used directly as a hoisting drum, traction wheel, winch capstan etc.

A still further object is to provide means for more rapidly cooling the propelling fluid as it passes thru the pump which I accomplish by the provision on the revolving drum shell of the pump as well as on a portion of the drum of the motor of heat radiating fins which increase the transfer of heat thru the shell to the cooling air. The transmission of power thru any medium is found to be attended with a loss of energy by friction in the form of heat and the efficiency of transmission is often limited by the speed of dissipation of this heat. In the old stationary casing pumps and motors of the type with which this application is concerned, it is apparent that the rate of dissipation of heat as compared to mine is very much lower since there is no relative movement in the older devices between the cooling air

and the heat radiating surfaces except of course the small movement due to convection. Where the radiating surface as in this type of hydraulic machine is so large as compared to the fluid contacting surface it is quite important that the radiating surface be given movement, and also the provision of fins on a moving surface is of much greater relative importance than the provision of fins on an equal stationary surface. In order to still further increase the cooling efficiency the fins may be provided with propeller or fan-like portions which break up the atmosphere that tends to revolve with the drum and also to increase the movement of the cooling air.

A further object of the invention is the provision of simple and efficient means for holding the revolving valve plate against the stationary barrel of the hoisting motor and at the same time preventing excess pressure between these two parts, the first means being an adjustable nest of springs and the second means being the provision of balancing grooves in the face of the revolving valve plate thru which grooves the live oil circulates under full pressure.

A still further and important object of the present invention is the method of coupling together two hydraulic pumps with two hydraulic motors so that both of the pumps can operate one of the motors when required, where under normal operation one pump operates one of the motors and in the piping between the pumps excess or diminished pressure actuates mechanism which controls the angle of the tilting box in the second pump which in turn causes a relatively small movement of the larger motor. The mechanism just recited also lends itself readily to hand operation since in ordinary drilling the stem is not raised but is lowered as in the judgment of the operator seems proper.

Other objects of the invention have to do with the specific design of the various parts all of which contribute to the provision of a power transmitting system which obviates the use of brakes and permits an almost instant change of direction of rotation of the motor unit without the slightest shock and which permits in case of drilling the building of the draw-works into a self-contained piece which can be positioned in the derrick as a unit so that in moving the unit from place to place there are no loose parts to be fitted to the derrick such as posts and head boards.

In previous devices for control of the pressure upon the drill bit by cooperation between the rotating and hoisting mechanisms the method of control has been thru changes in the strength

of the electric current or by purely mechanical means. The present invention contemplates the elimination of all electrical equipment between the rotating and hoisting means and substituting
5 for the mechanical means of inter-relating the driving and hoisting means of pipe lines conveying the oil of an air-tight closed system from a plurality of pumps to a plurality of means, the latter including a hydraulic motor for rotating
10 the drill stem and a larger motor for raising and lowering the drill stem.

The principal advantages gained by the use of the hydraulic draw-works of the present invention are ease of operation, assurance of a straight
15 hole, elimination of danger to the workmen because of abrupt stops and step-by-step gradations which are here eliminated, greatly decreased cost of inspection and repairs, and capacity to run for long periods of time without attention while preserving a definite and absolute
20 control of the torque on the rotary and of pressure at the end of the bit, eliminating respectively the many fishing jobs which are required by reason of the twisting off of sections of the drill
25 pipe and the variations from vertical in the drilling of the hole where inefficient control of pressure at the end of the bit occurs.

In the drawings:

Figure 1 is a plan view of the lay-out, being
30 partly diagrammatic to show the control system.

Figure 2 is a side elevation.

Figure 3 is a vertical axial section thru the hydraulic motor which is the drum.

Figures 4 to 8 are sections taken on lines 4-4,
35 5-5, 6-6, 7-7, and 8-8, respectively, of Figure 3.

Figure 9 is a fragmentary section taken on line 9-9 of Figure 8.

Figure 10 is a vertical axial section thru the
40 variable delivery pump.

Figures 11 to 15 are sections on lines 11-11, 12-12, 13-13, 14-14, and 15-15, respectively, of Figure 10.

Figure 16 is a vertical axial section of a modification of the motor.

Figure 17 is a vertical axial section of the control valve.

Figure 18 is a detail.

In order to give a comprehensive idea of the
50 invention I will first describe its general application in a well drilling control system such as diagrammatically illustrated in Figure 1. In this figure the rotary 10 is shown as driven by hydraulic drill motor 12 which receives its power by
55 supply and return pipes 14 and 15 via the four-way valve 16 either from variable delivery drill pump 18, a second four-way valve 19 being closed, or from variable delivery pump 20, both valves
60 16 and 19 being open, these pumps 18 and 20 being arranged so that these may be driven independently or by a common prime mover as will be more fully described hereinafter. The hydraulic hoist motor 23 is connected to the supply
65 pump 20 by pipes 21 and 22 which are also connected to mains 14 and 15 thru valve 19.

The drill motor 12 is controlled by means of a four-way valve 24 moved either manually by handle 25 or automatically by pressure chamber 26. The valve 24 communicates on one side with
70 two pipes 27 and 28 between the mains 14 and 15 at point 29 and on the other side with pipes 30 and 31 to the control cylinder of pump 18; one end of the pressure cylinder 26 of the valve 24
75 being connected by pipe 32 to the mains 14 and 15

at shunt 33, the other end being open to the atmosphere 45 (see Figure 17).

The hoist motor 23 is controlled substantially in the same manner as motor 12, that is, by valve 34 which, like valve 24, is connected on one side to mains 21 and 22 by pipes 37 and 38 at point 39 and on the other side to the control end of pump 20 by pipes 40 and 41, but unlike valve 24, which has an automatic control that is responsive only to the feed pressure of the motor 12, the valve 34 has a control that is not only responsive to the feed pressure of hoist motor 23 but is also responsive to the feed pressure of motor 12 as well. This is a differential control and is accomplished by connecting one end of the pressure cylinder 36 to the mains 21 and 22 with pipe 42 at shunt 43 and connecting the other end of cylinder 36 to mains 14 and 15 at shunt 44 by means of pipe 45. The shunts 33, 43 and 44 are each provided with two valves in order that pressure may be maintained in the control valve cylinders 25 and 36 regardless of the direction of flow in the mains.

The main shaft 46, mounted in bearings 47, is driven in any preferred manner as, for example, by an electric motor mounted directly on the shaft but as I prefer it and as illustrated this shaft receives power thru a sprocket wheel 48 connected by a chain to a source of motive power 49 which may be gas, electric, steam, or water driven. The hydraulic pump 20 is directly secured to the main shaft 46 by the coupling 50 and follows standard practice, the angle of the tilting box being governed by the position of valve 34 as will be more fully set forth hereinafter.

The cathead shaft 51 is mounted in bearings and has at either end the usual cathead drum. This shaft receives its motion thru sprocket wheel 52 connected by a chain 53 with sprocket wheel 54 which may be caused to rotate with the main shaft 46 by the clutch 55. It will be noted that this chain 53 is the only chain used in the mechanism except for the optional chain to the source of power which is only used in case the motive power is not directly applied to the shaft.

The pump 20 discharges oil under pressure to the hydraulic motor 23 which is the hoisting drum, the discharge pipe being 22 and the corresponding return pipe being 21. It is extremely important that the oil within the system shall fill all of the space except that occupied by the metal and it is equally important that the oil shall be free of air or other gas that is compressible. To this end I follow established practice and position an oil reservoir 56 above the oil level in the pump and the motor and while I shield this reservoir from dust and dirt by means of the cap 57 I maintain communication with the outside air in order to preserve atmospheric pressure within the receptacle 56.

Considering particularly Figure 2, the discharge pipe 22 from the pump leads thru a passage 59 to the vertical pipe or passage 60 at the left of the center of the drum while the return pipe 21 communicates with the right hand vertical pipe 61 thru the horizontal pipe 62. When the valve 34, and therefore the tilting box of the pump 20, is in neutral position no oil is pumped thru pipe 22 and the hydraulic motor 23 is therefore stationary since no oil is flowing and consequently no movement of the drum is possible. A slight movement of the valve 34 changes the angle of the tilting box of the pump 20 and causes a very slight amount of oil to be pumped

by each of the pump pistons giving a slight but constant torque on the drum as will be described later. The pump 20 may be of standard manufacture but I prefer to use one of the type developed by me and to be described farther on in the specification.

Without describing at this time the detailed construction of the hydraulic pump 20 and motor 23 which forms the hoisting mechanism of the device I will now describe the general operation. Drilling. Automatic control: First open valve 16 and close valve 19. Next start pump 20 and adjust the control valve 34 so that motor 23 will exert enough torque thru the hoisting drum to support most of the weight of the drill stem allowing the balance, a certain predetermined weight, to rest on the drill bit. The relative amount of weight taken by the drill bit and by the hoisting drum is governed by standard and well known practice and permits extremely fine changes in actual pressure by the cutting end of the drill stem on the material encountered. The third step is to open and adjust control valve 24 so that pump 18 starts and drives motor 12 and thus rotary 10 at any desired speed.

The pressure in main 14, proportional to the load on motor 12, is reflected thru the pipe 45 to one end of the pressure cylinder 36 of valve 34 which tends to cause the latter to allow an increased flow of fluid to the control end of pump 20 and thus increase the torque of hoist motor 23. But this reflected pressure in cylinder 36 is opposed and balanced by pressure in the opposite end reflected from main 22 so that the motor 23 still exerts the predetermined pull or lift on the drill stem. However, if the drill bit should strike a hard strata or should become wedged and thus increase the load on motor 12 to a greater extent than determined by the original setting of the valves 24 and 34 a back pressure will build up in line 14 and from there will be transmitted to both control cylinders 26 and 36. The increased pressure in 26 will unbalance the forces as determined by the original setting and cause valve 24 to allow pump 18 to increase its delivery of power to motor 12 to take care of the increased load. The increased pressure in 36 via pipe 45 will overbalance the opposing pressure in the cylinder due to the line pressure from main 22 transferred to the cylinder by pipe 42. This overbalance will move the valve 34 in a direction to cause pump 20 to increase the power delivered to motor 23 which in turn will lift the drill stem but only an amount necessary to relieve the overload on motor 12. Of course as soon as this occurs the valve 34 is automatically turned again to its original setting since the two pressures delivered by pipes 42 and 45 will have returned to their original canceling values and the predetermined rate of drilling will be resumed until further obstacles are encountered by the drill bit. There will be an opposite effect on the control valves 24 and 34 if the bit strikes a soft strata since this takes the load off motor 12 and increases the load on motor 23. No load on the former means low pressure in main 14 and thus in pipes 32 and 45. Low pressure in 32 causes valve 24 to turn in a way to cut down the supply from pump 18 allowing motor 12 to slow up while the same decrease in line 45 causes valve 34 to cut down the delivery from pump 20 which allows the drill to be lowered by hoist motor 23. The action of the control of pumps 18 and 20 as well as the structure of valves 24 and 34 will be explained in detail later on.

Drilling. Hand Control: First open valve 16 and close valve 19 as in the automatic control. Next start pump 18 and operate valve 24 by hand lever 25 until motor 12 is driven at the desired speed. Thirdly, start pump 20 and by means of hand lever 35 cause the pump to deliver enough power to motor 23 to suspend some of the weight of the drill stem allowing the rest of the weight to force the drill bit into its work. If desired the pump 20 and motor 23 may be stopped entirely and the drill may be fed downward by means of the hand brake 19' and handle 20'. It will be seen that this operation is very similar to the automatic control and differs therefrom only in that the valves 24 and 34 are no longer responsive to pressure since pressure lines 32, 42, and 45 are shut off which permits the valves 24 and 34 to be controlled entirely by means of hand levers 25 and 35. The sensitiveness of hand control will, of course, depend on the ability of the operator to respond to the reactions of the drill motor 12 and in most cases the automatic control is much better. But for drilling thru soft strata hand control is preferable.

It will be apparent that valve 24 may be left under automatic control while valve 34 is operated by hand or vice versa.

Hoisting: Since the drill stem is lifted entirely by means of motor 23 it is necessary that motor 12 should be shut down which is done by closing valve 16 and setting pump 18 on neutral or stopping the pump entirely by throwing out clutch 59 on main shaft 46. Then by manipulating handle 35 the valve 34 may be turned so as to supply enough pressure to the control end of pump 20 to cause the latter to deliver the amount of oil necessary to drive the motor 23 at the desired hoisting speed. In raising long heavy drill strings it may be necessary to use both pumps in which case valve 19 is opened and pump 18 is adjusted to deliver power in parallel with pump 20.

Referring now to Figure 3 which shows a vertical section generally thru the axis of the hoisting drum. The two frame pedestals 63 and 64 support the device but it must be remembered that the large portion 66 of the shaft of this motor does not revolve in the bearing of 63 but on the contrary the shaft 66 is held stationary by a plurality of keys 67 which are purposely made not too tight fitting. The bearing in the pedestal 64 on the other hand is a rotatable one and the part supported is the right hand end of the revolving shell which latter in turn supports the stationary torque shaft 70 in self-adjusting roller bearings 71 housed in the hub 72 of the rotating angle box 73. The central stationary piece, which is substantially if not actually integral and consists of the ported shaft 66, the solid torque shaft 70 and the cylinder barrel 75, is supported at the left in the pedestal bearing 63 to which it is keyed, and at the right by the bearings 71. It is very important that this stationary unit shall maintain the left hand plane face of the cylinder barrel precisely parallel to the engaged face of the rotating portion of the motor, this being true of all devices of this kind.

Considering first the parts that have no rotation,—the larger shaft portion 66 is closed at the left as seen in Figure 3 by a pipe flange 77 tapped to receive the two oil pipes 60 and 61 which are merely horizontal continuations of the previously mentioned vertical pipes 60 and 61 and consequently bear the same numerals. Surrounding the large shaft 66 and in engagement with the pipe flange 77 are shims 78, the number of which will be

varied to secure the desired pressure adjustment between the cylinder barrel and the revolving valve plate, this adjustment being arranged thru the keys which, as before remarked, may have movement axially of the shaft, and by a ring 79 which engages the outer case 80. The adjustment may be obtained by means of a plurality of adjusting screws (not shown) spaced around the flange 77 and acting either on the shims 78 or directly on the keys 67 and thus providing ready means for moving the keys inward against the ring 79 to keep the desired pressure on the case 80 between which and the similar inner case 81 are a plurality of helical springs equi-spaced about the shaft 66. These springs, which will vary in size and strength with the different sizes of motor, are arranged with their axes parallel to the axis of the shaft 66. The spring nest 82 is not needed when the drum is being revolved but it is very important when the device is not under load as these springs, whose total strength is very material, preserve always the needed firm bearing between the valve plate and the cylinder barrel.

Inward of the spring nest and its two cases are the roller bearings 85 upon which revolves the valve plate 80 to which is secured the annular closure 89 which has an oil tight engagement with the ring 79. Instead of the packing shown in Figure 3 I may use an arrangement similar to that illustrated in Figure 10 which will be described farther on in connection with the pump unit, therefore in Figure 3 in several places I have purposely omitted the packing for the sake of clearness of illustration but it will be understood that the device is made as oil tight as possible at every joint.

The barrel is a relatively short cylinder of solid metal having within it a plurality of piston receiving cylinders 88 each having one or more ports 87 somewhat radially outward from its axis. Near the left hand end, as seen in the figure, there are ports 92 and 93 opening in fan shape from the passages 60 and 61 in the solid metal of the shaft 66 and the barrel 75. Since these passages are mere continuations of the pipes 60 and 61 they have been given the same numeral for the sake of clearness so that the numeral 60 may be considered the entrance passage for the oil from the pump 20 via the horizontal pipe section 22. It will be noted that the radius of the arc of the segmental port 92 is considerably greater than the radius of the similar arc of the segment-shaped port 93 so that these two ports may communicate with what corresponds to the elongated ports of the typical valve plate which is here varied materially in order to secure greatly increased strength and to avoid the noticeable hoop tension in the walls of the usual valve plate.

At the right hand end of the barrel 75 is the torque shaft 70 which ends just beyond the margin of the inner race of the roller bearings 71. Between the bearing 71 and the point of junction with the integral barrel the torque shaft has a central opening 95 in which is mounted the trunnion block 100 having horizontal trunnions 96 on which turns the socket ring 98 which in addition to its oscillation with the block about the axis of the two trunnions 96 has an oscillation relative to the block 100 at right angles thereto, about the axis of the two trunnions 99 arranged at right angles to the axis of the trunnions 96 both axes lying in a single plane, here positioned at an angle from the vertical. I preferably provide bushings 76 between the shaft and the trunnions 96 and

between the socket ring and the trunnions 99. Two trunnions are integral with the trunnion block 100 and two other trunnions are formed at the ends of a pin driven thru the trunnion block so that the non-rotating socket ring has movement about the stationary torque shaft as would a disk which is rockingly mounted at its center upon a point so that the point of dip runs around the circumference but the disk has no angular motion about the point. Such motion is common in this art but the present device differs from other motors in that in this case it is the torque shaft which is stationary and it is the angle box which rotates.

The swash-plate or socket ring 98 may be a two-piece ring as shown in Figure 10 and as described later on in connection with the pump but in the modification shown in Figure 3 is a one-piece ring having hubs 101 to surround the trunnions 99 and being cut away as at 102 to clear the enlarged yoke section of the torque shaft 70. In this ring are a plurality, in the present case seven, nearly equi-spaced sockets each receiving a bushing 103 cooperating with an intermediate bushing 104 and a threaded bushing 105 to receive snugly the spherical end 106 of the connecting rod 107, the other spherical end 108 of which is received in the piston 109 in quite similar manner, again following usual practice, my desire being that the fit of the piston in the cylinder 88 shall be so perfect that I require no rings or packing. The pistons and cylinders are ground and lapped to a smooth working fit and I prefer to follow the approved scheme of cutting narrow and shallow grooves around each of the pistons, preferably having at least three such grooves in each piston so as to trap dirt and also break the fluid leakage lines.

The principal rotating parts are the valve plate 90, the drum itself, and the angle box 73, these three parts being substantially integral but actually formed as three separate units which are securely but not permanently secured together since it is desirous of having convenient access to the interior of the drum for repairs and adjustments. The spooling drum is a hollow cylinder having right angular flanges 110 and 111 to which are secured respectively the flange 112 of the angle box 73 and the flange 113 of the valve plate, the line of bolts being diagrammatically represented by the numerals 114. The outer cylindrical surface of the drum may be and preferably is grooved as at 116 to fit the size of rope 119 (see Fig. 18). It is desired to use rope usually somewhat in excess of an inch in diameter and normally perhaps slightly less than half a mile in length.

The angle box is somewhat hemispherical between its flange 112 and its hub 72. The angle of this box corresponds exactly to the angle of the socket ring, that is, for example, about 20° or 30° from vertical. The angle box which revolves about the axis of the shaft 66—70 is provided with thrust and roller bearings 117 and 118 in suitable races one member 120 being common to both. The hub 72 of the angle box 73 has secured to it at the right hand end in Figure 3 a drum closure 121 integral with an outwardly extending shaft 122 supported by outboard roller bearings 123 in an outboard bearing box 125 which in turn is closed by an outboard bearing closure 126 having a grease inlet valve 127. The outer race of the bearing 123 is made floating to allow for the necessary but extremely slight axial and angular movements of the rotating parts.

The valve plate 90 is shown in cross section in Figure 3 and in elevation in Figure 4. This valve plate differs from usual practice in certain important ways, first the usual elongated port which pierces the ordinary valve plate is here reduced to a shallow groove 130 communicating with a series of spaced ports 131, the axes of which are parallel to the axis of the cylinder barrel. These ports 131 each communicate thru radially disposed passages 133 with the inner port 134 which in turn communicates with the port 93 in the cylinder barrel. On the other semi-circular half of the valve plate the groove 140 corresponds with 130, the ports 141 are similar in size and position with the ports 131 but the radial ports 143 are considerably shorter than the corresponding ports 133 since the former lead to the port 144 which communicates with the port 92 in the barrel.

An important feature of the invention is the continuance of the ports 134 and 144 as shallow grooves 135 and 145 respectively, these two grooves together insuring that the oil under pressure shall at all times completely encircle the contacting faces of the valve plate and cylinder barrel thereby relieving these two faces of the huge pressure exerted at times by the reaction to the thrust of the connecting rods 107. Space 147 provides a clearance between the valve plate and the casing. The small round ports are used for strength, otherwise the port wall would have to be very heavy because of the large radius of the port walls from the center.

The operation of the drum is believed to be quite apparent, the action being as follows: Upon increase in pressure in the pipe 14 and therefore in the control pipe 45 connected thereto, the control valve 34 of the pump 20 is operated in a manner described below, thereby causing a slight additional amount of oil to discharge from the pump 20 thru pipe 22. This fluid continues thru the passage 60 in the larger shaft portion 66 of the stationary member of the drum, passes upwardly in Figure 3 and to the right as in Figures 4 and 5, passing into the segmental shaped port 92 of the cylinder barrel and from thence into the aligned port 144 at the right of Figure 4 and in the upper half of Figure 3, from here passing thru the radial passages 143 to the series of ports 141 and passing from here into the aligned ports 87 leading to a plurality of the cylinders 88, causing these pistons 109 to move to the right in Figure 3 which movement causes the upper end of the socket ring to move to the right, this in turn causing a rotation of the angle box so as to increase the tension on the hoisting drum rope.

While this action is taking place the lower portion of the socket ring 98 is moving to the left as seen in Figure 3 and this forces oil thru the lower ports 87 in the cylinder barrel to the aligned port 131 in the valve plate, the oil from here passing radially inward thru the ports 133, then thru the port 134 in the valve plate to the fan or segment-shaped port 93 in the cylinder barrel, this port being constantly in communication with the passage 61 which leads to the oil return pipe 62, returning the oil thru the pipe 21 to the pump 20. This is the normal drilling operation but when it is desired to raise the entire drilling stem, the valve 19 is opened and the valve 16 closed. In this way the rotary drum 23 is driven by the combined action of the pumps 20 and 18, my preference being that the pump 20 shall be of say twice the power of the pump 18 and that

the drum 23 for full power shall require the combined horse power of the pumps 20 and 18, whereas the needs of the motor 12 shall always be satisfied with the delivery of fluid from the pump 18. When raising the stem the stop valves in the auxiliary lines 32, 42 and 45 are closed when the valve 16 is closed and the tilting box control valves 24 and 34 are operated by hand to give the required speed to the drum.

The improvements pointed out in the above description of my hydraulic motor are also present in my variable-delivery pump as will be apparent on consideration of Figure 10 which is a central sectional view of the new pump. As in the motor the pump is provided with a stationary cylinder barrel 75 which may be integral with the shaft 66 as in the motor but for large capacity machines is preferably made separate and provided with a large central bore for fitting over the reduced end of shaft 66 to which it is keyed. The barrel has a plurality of cylinder bores 88 parallel to the axis of the stationary shafts 66 and 70 and communicating by cylinder ports 87, valve plate ports 130, 131 and barrel ports 92 and 93 with the passages 60 and 61 of the shaft 66. Each cylinder bore 88 carries a piston 110 fitted with a connecting rod 107 having spherical ends 106, working in a socket ring of two pieces 150—151 clamped about the ends 106 and mounted to wobble on the main stationary shaft 70 by means of a universal joint as in the motor.

Between the two parts 150—151 of the socket ring are clamped the bushings fitting the trunnions 99 and there are also clamped the split bushings 152 against the bushings 153 which provide sockets for the ends 106. The part 151 has the same thrust and side bearing structure as the socket ring described above in connection with Figure 3 and therefore cooperates in the same manner with the roller bearings 117 and 118 mounted in the tilting box 154.

However, while in the motor of Figure 3 the angle box had a fixed angle with respect to the axis of the drum with which it turned in the case of the pump as shown in Figure 10, the angle is not fixed but is made adjustable by mounting the tilting box 154 to turn on diametrically positioned trunnions 155 which are studs screwed into the casing 156 and provided with sockets 157 so as to be replaced and adjusted from the outside of the casing (see Figure 15). On the two ends of a diameter perpendicular to that passing thru the trunnions 155 are provided lugs 156' which are preferably integral with the tilting box 154 and to which are connected, either by ball and socket joint or otherwise, the connecting rods 157' socketed at their opposite ends in the control pistons 158 mounted in cylinders 159. As shown in the drawings the cylinders 159 may be integral with the revolving casing 156 as well as integral with the auxiliary casing 160 which fits over the portion 162 of the inward extension 163 of the valve plate 164; the auxiliary casing, however, is slightly spaced at 165 from the remainder of the extension 163 as well as from the barrel 75 most of which it telescopes. A connection with the pressure lines is provided for each cylinder 159 one of which may be traced from the cylinder end at 167 along the casing at 168 down the drum head 73 where at 169 it connects with the radial passage 170 in the flange or drum closure 121 which passage joins the passage 173 in the driving shaft 174 which in turn is connected to the annular passage 175 in the stationary stuffing box 176 where a threaded bore 75

40 provides means for connection to a line 40 leading to one of the four way control valves 24 or 34. The other pressure connection may be traced from the lower cylinder 159 in a similar manner to bore 41 in the stuffing box 176.

It will readily be seen that the positions of the control pistons 158 determine the angle of the tilting box 154 and that by varying the relative pressures on the control pistons the stroke of the pump pistons may be positively adjusted. Therefore by means of one of the above mentioned four-way control valves, I connect the control cylinders 159 to the previously described pressure mains 14 and 15 (or 21 and 22) and then by moving the valve and admitting oil under pressure to the cylinders I can determine the delivery of the pump, the casing 156 of which is driven at a constant speed by means of a gear or pulley connected on the end 178 of the driving shaft 174. Instead of connecting the control cylinders to the pressure mains I may connect them to a separate source of control pressure thus making the control independent of the pressure in the mains which may be desirable for starting the pumps from neutral when the pressure in the mains is too low to move the pistons 158 but after starting the pump the pressure lines may be again connected to the mains if desired.

Referring to the left hand side of Figure 10, it will be seen that the annular valve plate 164, surrounding shaft 66 and engaging the vertical surface of cylinder barrel 75, is integrally supported on the end of the cylinder-like projection 179 of the drum closure 180 which is bolted to the drum or casing 156 by means of bolts 114. Within the cylindrical projection 179 are the roller bearings 85 held in inner and outer annular races 182 and 183, respectively. The outer race 182 is held against a ring-like projection 184 on the rear of the valve plate by the cylindrical projection 185 of the annular closure 186 bolted to the drum closure 180. Within and spaced from the projection 185 and engaging the stationary shaft 66 is a nest of springs 82 held between two rings 80 and 81, the latter of which engages the race 183. As fully described in connection with Figure 3 the tension of the springs is adjusted by moving the collar 79 to the right either by shims 78 between flange 77 and keys 67 or by screws projecting thru flange 77 into engagement with the keys 67. The closure 186 also carries a stuffing box 188 within which is the leather U-packing 189, filler ring 190 and packing gland 191 all of which make an oil-tight closure for the casing. If any oil should leak by the inner annular lip 192 of the valve plate and find its way to the ball races it would not force its way out thru the stuffing box 188 because passages 193 are provided for draining it into the casing by way of the above mentioned space 165 between the casing 160 and the plate and barrel. As I intend to have all space filled with oil it will be seen that additional leakage into the casing 156 will raise the pressure, therefore two ball checks 195 are provided for relieving this pressure. Two checks are provided (one in each main 14—15 or 21—22) in order that there will always be a low pressure main connected to the inside of the casing, for example, by way of passage 196 regardless of the direction of flow. As will be seen in Figure 10 I locate these check valves at the end of the bores 14—15 and by providing the set screws 197 with kerfs or sockets 198 I am able to set or to adjust the valves from the outside by means of a long

shanked socket wrench 199 which I insert thru a hole in the manifold 77 which hole, of course, is ordinarily plugged.

In order that the heat from the transmitting fluid may be quickly dissipated I provide radiating fins 201 on the revolving casing 156 and for protection of the workmen enclose the whole in a perforated casing 202.

Referring to Figure 16 wherein is shown a modification of the motor illustrated in Figure 3 it will be seen that most of the parts are identical with my improved pump described in the preceding paragraphs from which it differs essentially only in replacement of the tilting box 154 by the stationary angle box 73 of my other modification and, of course, the absence of the control cylinders and pistons as well as the elongated stuffing box 176. The drum is substantially the same as that of Figure 3 but in order to provide additional means for taking power from the motor I use an extended shaft 174 having means at 178 for attaching a pulley, gear, coupling or any other power transmitting device.

In some types of work it may be desirable to build the gear or pulley directly on the drum, also in traction work the drum may constitute the hub of the driving wheel.

Altho most any one of several types of pressure responsive valves 24—34 may be used in my system diagrammatically shown in Figure 1, I prefer the one shown in detail in Figure 17 which comprises two main parts, a four-way valve 24 and a pressure cylinder 26, the latter being divided into two expansible chambers 210 and 211 by the reciprocating piston 212 provided with the usual air-tight or fluid-tight gaskets 214 and 215. The compartment 210 is provided with a port 32 for connection to pressure line 32 while compartment 211 has a port 45 which may be either connected to a pressure line 45 as is cylinder 36 of Figure 1 or may be left open to the atmosphere as is cylinder 26 of Figure 1. The piston 212 is carried on rod 216 connected at one end to the hand lever 25 and at the other end to the piston 217 of the four-way valve 24.

The piston 217 as well as the piston 212 are biased to an upward position partly by the hand lever 25 but mostly by the spring 218 on which the piston 217 rests. The tension of this spring 218 therefore provides a way for adjusting and adapting the pressure cylinder to operate with systems of different pressures and with different connections, for example, if port 45 is to be left open to the atmosphere as is the case with cylinder 26 of Figure 1 the spring tension should be made large to balance the pressure in line 32 but where both ports are connected to pressure lines as are 42 and 45 of cylinder 36 the spring tension need not be so large. A ready means for regulating the spring tension is provided by the plug 220 in the base of the four-way valve. The usual stuffing boxes are provided at 221, 222 and 223 so that no oil will leak out around the piston rod.

The operation of the valve will be apparent when considered with the description of the operation of the system as a whole given near the beginning of this specification.

Certain details and instructions have purposely been omitted from this specification but will be apparent to those skilled in the art and to whom the specification is addressed, for example, in all such mechanisms it is necessary to provide relief valves to keep the oil pressure from rising above that which is necessary to overcome nor-

mal resistance in hoisting and consequently such valves are to be added to the present device being such, for example, at 1100#, the valves being merely spring pressed check valves preferably of the type where the pressure can be altered. Upon excess pressure the oil escapes from the high pressure port passage into the drum space and from here passes into the low pressure port thru a suitable check valve.

10 What I claim is:

1. In combination, a hydraulic pump of the type having a tilting box, a change of angle of which causes flow of fluid from the pump in chosen direction and amount, a hydraulic motor cooperating therewith having an angle box characterized by the barrel of the hydraulic motor being stationary while the shell is fixed to and revolves with the angle box, said shell forming a hoisting drum, and a fixed shaft secured to the barrel and serving as a support for the barrel and the shell, said shaft having bores therein for conducting operating fluid to and from the motor.

2. A brakeless hoisting drum for drilling operations, consisting of a hydraulic pump motor unit in which the motor and the pump are of the swash plate type, said pump and motor each having a stationary piston barrel, a revolving shell enclosing said pump, a revolving shell enclosing said motor, an angle box carried by each shell, means on the pump shell for tilting the angle box carried thereby, a control rod for controlling said means to vary the stroke of the pump pistons, the angle of the socket ring of the motor being constant, the pistons of the motor acting as a drum brake in response to changes in position of the control rod of the pump.

3. In a hydraulic motor hoisting drum, a stationary shaft consisting of an end section having a pair of fluid passages therethru, a barrel substantially integral with said shaft, and having ports communicating with said passages, and a torque shaft, a socket ring oscillating about a trunnion block carried by the torque shaft, pistons reciprocating within the barrel, connections between the pistons and the socket ring, and a revolving unit mounted to turn about said stationary shaft and consisting of a valve plate, a drum and an angle box.

4. In a hydraulic motor hoisting drum, a valve plate, a cylinder barrel, communicating ports in said barrel and valve plate characterized by the ends of each of the ports being continued annularly about one of the contacting faces as a shallow groove, thru which the fluid under pressure may circulate to reduce the friction between the valve plate and the barrel, a valve plate port consisting of an arcuate shallow groove, a plurality of axial bores extending from said shallow groove, a port radially inward from said bores and in communication therewith.

5. A valve plate having on each half an arcuate groove, having in one-half an arcuate port of a chosen radius and having on the other half an arcuate port of different radius, and radially disposed channels establishing communication between said grooves and one only of said ports at each side.

6. A hydraulic motor, the outer shell of which is a hoisting drum, a cylinder barrel, a socket ring having oscillating movement with respect to the cylinder barrel, an angle box fixedly secured to and forming one end of said drum, a valve plate secured to and constituting the other

end of said drum, and means for holding the cylinder barrel against rotation, said means comprising a drum supporting bracket and a shaft secured to the bracket and to the barrel.

7. In a hoisting drum revolving motor, a cylinder barrel, a shaft substantially integral with the cylinder barrel, a valve plate mounted to revolve about the shaft, distributing ports in the rotating valve plate and fluid passages thru the shaft and the cylinder barrel for delivering fluid from a pump to the distributor ports in the valve plate and from thence to the cylinders of the cylinder barrel.

8. A constant stroke swash plate type hydraulic motor in which the trunnion block has a fixed angle of oscillation and the outside shell of the motor revolves while the cylinder barrel is held stationary and an angle box fixedly secured to said shell and cooperating with the trunnion block to maintain said fixed angle.

9. In a hoisting drum, a stationary torque shaft, an angle box having a hub rotating about the axis of the torque shaft, a roller bearing between the end of the torque shaft and the hub, a shaft extending from the angle box, a stationary frame, and a floating roller bearing supporting the end of the angle box shaft in said bearing.

10. A shaft, a valve plate, a cylinder barrel substantially integral with said shaft, a roller bearing between said valve plate and said shaft, a frame, a key engaging said frame and said shaft to prevent rotation of the shaft, a spring nest to resiliently hold the valve plate against the cylinder barrel, and means including said key for altering the pressure exerted by said spring nest.

11. The device of claim 10 in which the spring nest consists in a plurality of helical springs placed annularly about the shaft with their axes parallel to that of the shaft.

12. A valve plate having two spaced ports, one on either side of a diameter to establish communication with the cylinders of a cylinder barrel, said ports being characterized by the provision of spaced bars communicating with a shallow groove in the face of the valve plate and communicating thru separated passages with the intake and exit ports, whereby the strength of the valve plate is greatly increased over that given by a single opening port.

13. In a well drilling system, a hydraulic pump of the swash plate type, a stationary cylinder barrel, a swash plate carried by said barrel, a revoluble shell coaxial with and enclosing said barrel and plate, an angle box pivotally supported by said shell, a pair of cylinders and pistons therein carried by said shell, said pistons being connected to diametrically opposite points on said angle box whereby the relative positions of said pistons control the angle of said box, a pipe connection between one of said pair of cylinders and the pressure side of said pump and a second connection between the other of said pair and the return side of said pump and means in said pipe connection for regulating the pressure between the pump and the cylinders whereby upon the rise of pressure to a predetermined limit the pistons are moved to change the angle of the angle box to thereby reduce the stroke of said pump.

14. In a well drilling system, a hoisting drum comprising a pair of spaced aligned supports, a stationary shaft carried by one of said supports, a drum between said supports having a journal at one end for turning on said shaft, a drum head closing the opposite end of said drum, a shaft

secured to said head and journaled in the other of said supports, a cylinder barrel secured to said stationary shaft and located in the opposite end of said drum, and having a projection journaled in said drum head, a swash plate trunnioned in said shaft, and an angle box carried by said drum.

15. The device of claim 14 in which said stationary shaft is provided with passageways for carrying fluid to and from the cylinder barrel.

16. The device of claim 14 in which said stationary shaft has a central bore extending thru said barrel and to said swash plate, a bore on each side of said central bore acting as conduits for transferring fluid to and from said cylinder barrel, said barrel having a semi-annular groove in one face communicating with one of said conduits, and a second semi-annular groove of the same radius communicating with the other conduit, an annular groove surrounding said semi-annular grooves and communicating with the cylinders in said barrel, a valve plate having ports for cooperating with said grooves to control the flow of liquid between the barrel cylinders and the conduits, and a hub on said stationary shaft between its support and said plate.

17. In a well drilling system, a hydraulic machine of the swash plate type, a pair of spaced supports, a stationary shaft in one of said supports, a stub shaft aligned with said stationary shaft and journaled in the other of said supports, a hub journaled on the free end of said stationary shaft and secured to said stub shaft, a cylinder barrel secured to said stationary shaft, a second hub journaled on said stationary shaft between its support and the cylinder barrel, a drum mounted on said hubs and forming therewith a fluid-tight enclosure for said barrel, an angle box secured to said drum, a swash plate trunnioned in said stationary shaft, pistons reciprocally mounted in said barrel and connected with said plate, said stationary shaft having conduits therein for transmitting work fluid to and from said barrel and a valve plate secured to said drum for controlling the flow of fluid thru said conduits.

18. In a hydraulic motor hoisting unit of the swash plate type, a stationary shaft fixedly supported on one end, a winding drum journaled on said shaft, a cylinder barrel fixed to said shaft intermediate the ends of the drum, a swash plate pivoted to said shaft for cooperating with said barrel, an angle box forming one end of the drum and cooperating with said swash plate, and a valve plate forming the other end of the drum, said shaft having bores from the fixed end for conducting operating fluid to and from the cylinder barrel.

19. In combination with a winding drum, a hydraulic motor of the swash plate type including a shaft fixedly supported on one end, and a cylinder barrel fixed to said shaft, said shaft having bores therein from said one end to said barrel, one end of said drum consisting of a valve plate for controlling the flow of fluid between said bores and the cylinders in said barrel, said winding drum forming the enclosing shell for said motor.

20. In a well drilling system, a hydraulic machine of the swash plate type adapted to operate as a pump when mechanically driven or as a motor when hydraulically driven and comprising a support, a shaft fixedly secured to said support, a swash plate and a cylinder barrel secured to said shaft, a cylindrical housing journaled at one end on said shaft and enclosing said barrel and said swash plate, an angle box pivoted in said housing for cooperating with the swash plate, a stub shaft secured to the other end of said housing for rotatably supporting the same, said other end of the housing being journaled on the inside for receiving the free end of said fixed shaft, said fixed shaft having bores from the support end for conducting operating fluid to and from said cylinder barrel, and pressure responsive means carried by the housing for controlling the angle of said box, said stub shaft having a bore therein for conducting actuating fluid to said pressure responsive means.

VICTOR W. ZILEN.