

April 7, 1953

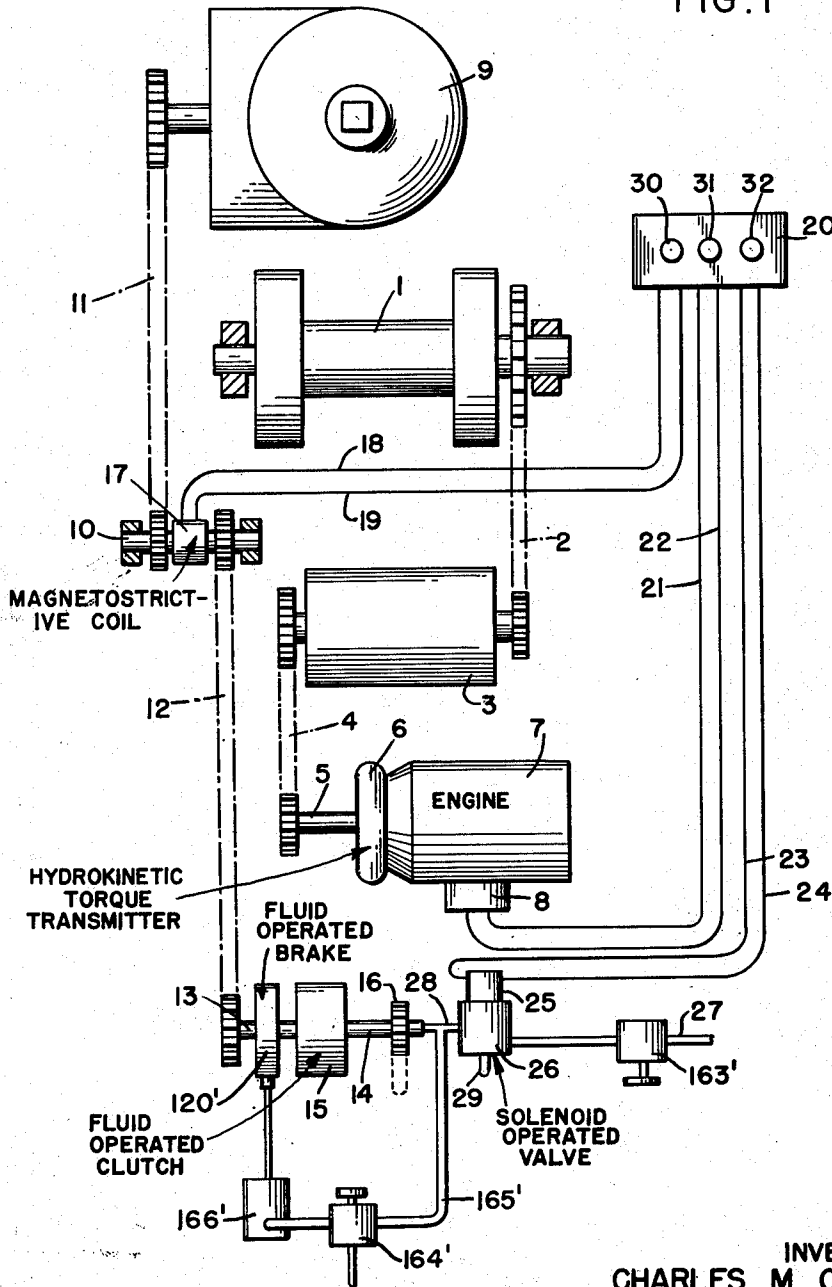
C. M. O'LEARY
AUTOMATIC DRILLING MACHINE

2,634,099

Filed Oct. 10, 1946

5 Sheets-Sheet 1

FIG. 1



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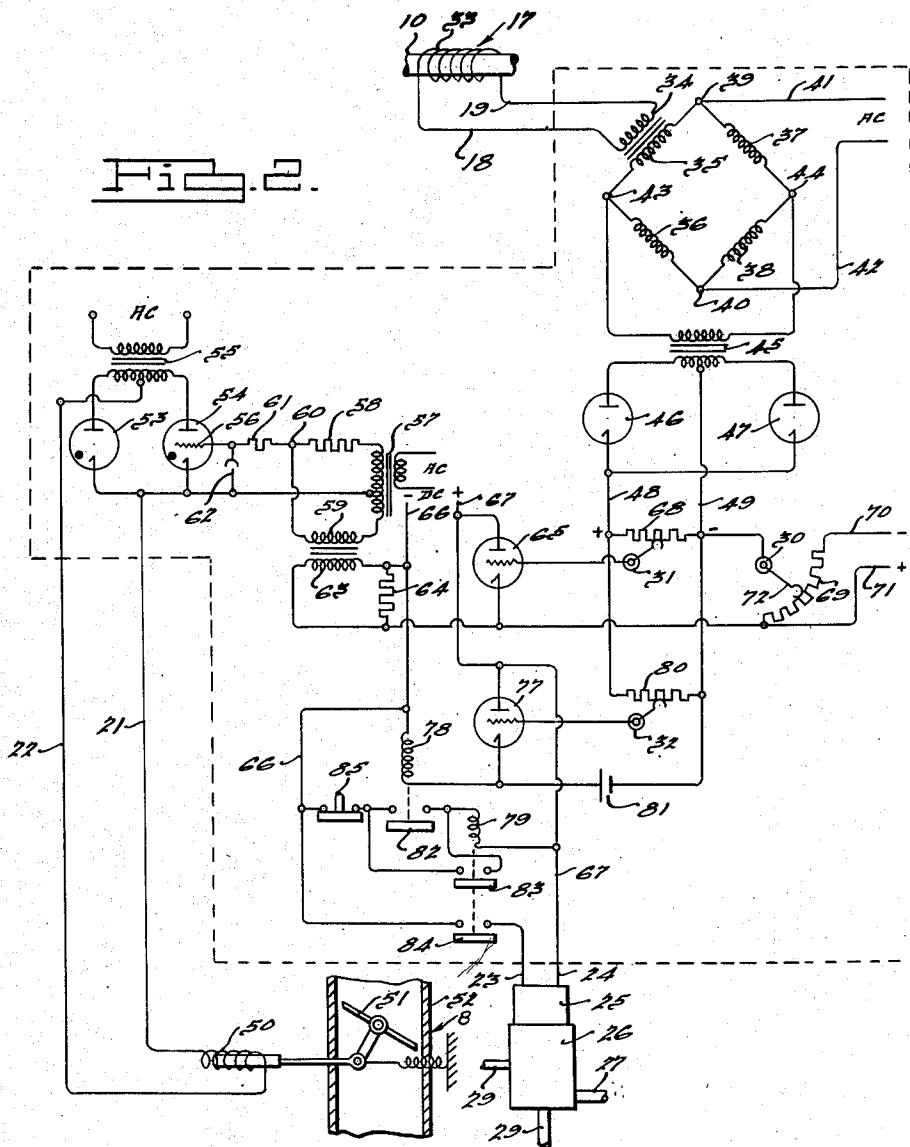
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AUTOMATIC DRILLING MACHINE

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5 Sheets-Sheet 2



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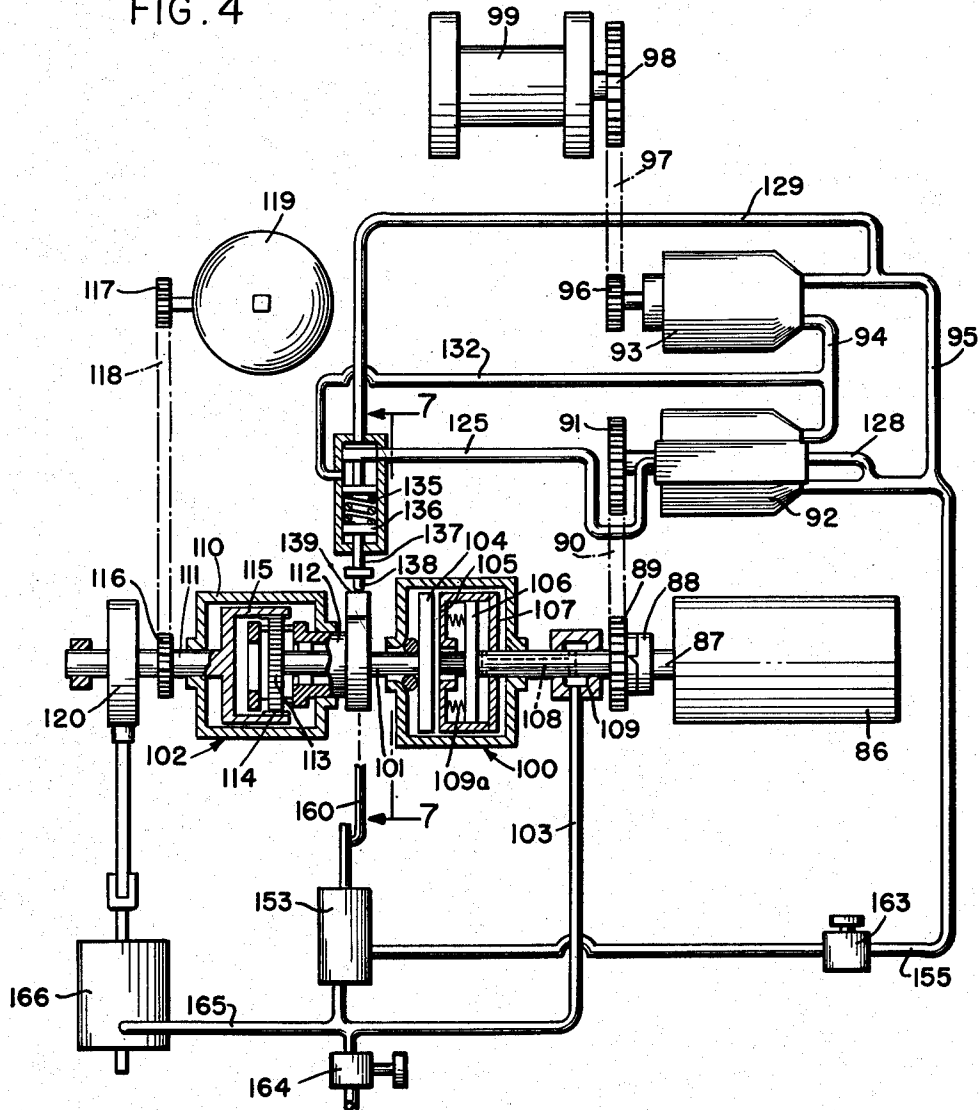
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5 Sheets-Sheet 4

FIG. 4



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AUTOMATIC DRILLING MACHINE

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5 Sheets-Sheet 5

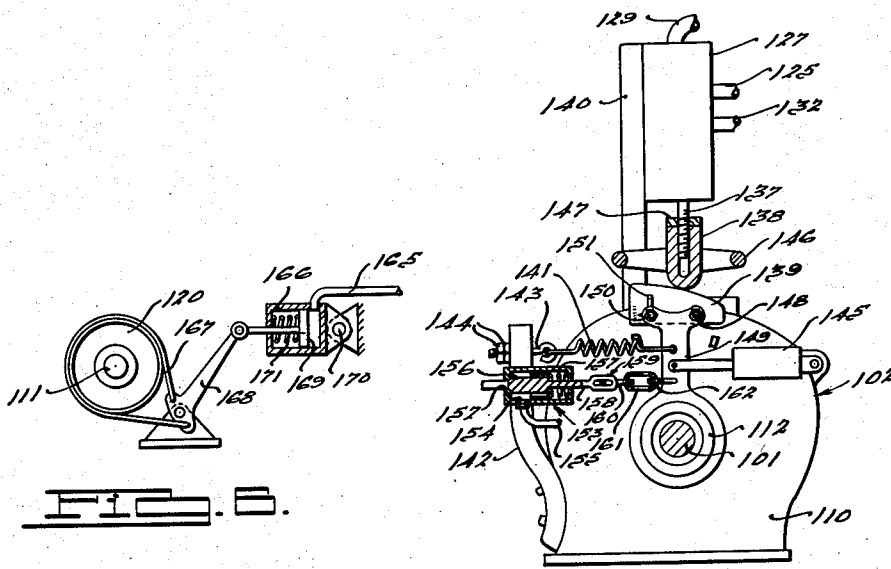


FIG. 1.

FIG. 2.

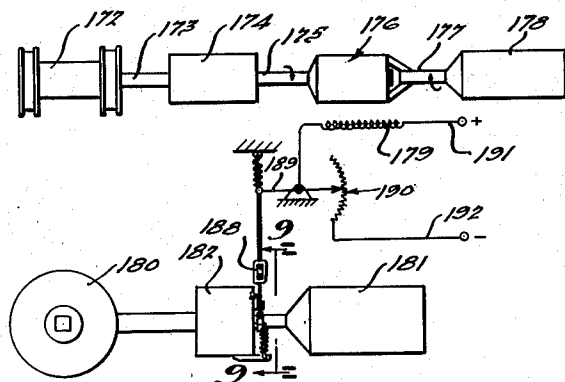


FIG. 3.

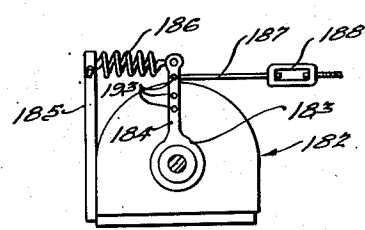


FIG. 4.

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UNITED STATES PATENT OFFICE

2,634,099

AUTOMATIC DRILLING MACHINE

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Application October 10, 1946, Serial No. 702,539

8 Claims. (Cl. 255—19)

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The present invention relates to rotary well drilling apparatus and constitutes an improvement on applicant's copending application, Serial No. 602,620, filed June 30, 1945, now Patent No. 2,589,118, on Automatic Well Drilling Mechanism. This application is based in part on applicant's copending application, Serial No. 628,793, now Patent No. 2,582,217, filed November 15, 1945, on Hydraulic Drilling Mechanism.

The foregoing application, Serial No. 602,620, discloses an apparatus for controlling the weight of the drill bit on the formation in such a manner that the drill bit pressure varies in accordance with the torque required to rotate the drill stem, thus preventing the development of an excessive drill stem twisting force. This result is achieved by varying the drill stem supporting force exerted by the hoisting drum. An important feature of the mechanism resides in the fact that the percentage rate of change of the drill stem supporting force is less than the corresponding percentage rate of change of the drill stem rotating torque. This result is achieved by transmitting to the hoisting drum a drill stem supporting torque from one source which remains at a constant value, independent of drum speed, and augmenting the drill stem supporting torque by auxiliary means which transmit a torque proportional to the drill stem rotating torque.

It is the general object of the present invention to accomplish the objects of the foregoing application, Serial No. 602,620, by improved mechanisms.

Another object of the present invention is to provide improved means for adjustably controlling the torque delivered by a variable torque transmitting device in accordance with the torque delivered by a separate mechanism, said means being capable of varying the torque transmitted by the variable torque transmitting device at a lower percentage magnitude of change than that of the torque transmitted through said separate mechanism.

Another object of the invention is to provide means for stopping rotation of the drill stem when the drill stem rotating torque becomes excessive in order to prevent twist-offs.

Another object of the invention is to provide means effective to prevent rotation of the upper end of the drill stem in either direction after the drill stem rotating torque has reached the predetermined value.

Other objects and advantages of the invention will become apparent from the following specification, the accompanying drawings and the appended claims.

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In the drawings:

Figure 1 is a diagrammatic view of the preferred form of the present invention;

Figure 2 is a diagrammatic view of the electrical circuit employed in the form of the invention illustrated in Figure 1;

Figure 3 is a diagram illustrating the performance characteristics of the invention;

Figure 4 is a diagrammatic view of a modified form of the invention;

Figure 5 is an enlarged view, partly in section, of a portion of the mechanism shown in Figure 4;

Figure 6 is a fragmentary view, partly in section, of a brake mechanism employed in the form of invention shown in Figure 4;

Figure 7 is a fragmentary section taken on the line 7—7 of Figure 4;

Figure 8 is a diagrammatic view of a third form of the invention; and

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

In accordance with all forms of the present invention, a suitable source of power, such as an internal combustion engine, is connected to the hoisting drum of a well drilling machine by a means which, in connection with the source of power, is capable of transmitting to the drum a torque that may be varied and maintained at any desired value, independently of the variations in the unwinding speed of the drum incident to the progress of the well drilling operation. A means responsive to the torque required to rotate the drill stem is provided for controlling the torque delivered to the drum by the variable torque transmitting device. This last means is adjustable not only to vary the general level or magnitude of the drill stem supporting torque, but also to vary the percentage rate of change in the drill stem supporting torque induced by a given percentage change in the drill stem rotating torque. Each of these adjustments may be made independently of the other. As a result, the percentage rate of change in the drill stem supporting torque may be substantially less than the corresponding percentage rate of change in the drill stem rotating torque, as is desired, and the mean drum torque may be readily adjusted to accommodate changes in the weight of the drill stem and/or the pressure of the drill bit on the bottom of the hole.

In the preferred form of invention illustrated in Figures 1 and 2, the variable torque transmitting device comprises an internal combustion engine, the output of which is connected through a hydrokinetic torque converter to the drum. Since with such an arrangement the output speed

of the torque converter is negligible during the drilling operation, the engine and torque converter will deliver a constant torque to the drum for any given engine throttle setting and that torque may be varied by adjusting the throttle position. Such adjustments in the throttle position are automatically made by an electrical torque responsive device associated with the rotary table drive. It will be noted that the value of the transmitted torque is varied by changing the speed of the source of power, and, consequently, the difference between the input and output speeds of the torque converter. Any suitable torque transmitting device capable of transmitting a torque which varies in accordance with the difference between the input and output speeds may be used in place of the hydrokinetic torque converter. Examples of such devices include a hydrokinetic coupler and an electrical eddy-current coupler or clutch; however, a hydrokinetic torque converter is preferred because of its torque multiplication characteristics.

As shown diagrammatically in Figure 1, a hoisting drum 1 is connected by means of a chain 2 and suitable sprockets to the output of a change-speed transmission, indicated diagrammatically at 3. The input of the transmission 3 is connected by means of a chain 4 to the output shaft 5 of a hydrokinetic torque converter, indicated diagrammatically at 6. The input side of the torque converter is connected to and driven by an internal combustion engine 7 having an adjustable fuel feeding device, indicated diagrammatically at 8. The adjustable fuel feeding device 8 may be the usual intake manifold throttle valve employed in spark ignition engines or, in the case of diesel engines, may include the fuel injector control mechanism.

The rotary table 9, which may be of conventional construction and is effective to rotate the drill stem during drilling operations, is driven from a suitable countershaft 10 by means of a chain 11 and suitable sprockets. The countershaft, in turn, is driven by a chain 12 and suitable sprockets from a shaft 13, which may be connected to or disconnected from an aligned shaft 14 by means of a pneumatic clutch 15. The shaft 14 may be driven in any suitable manner by means of a sprocket 16 either from a separate source of power or from the previously mentioned engine 7, as desired. A pneumatically-operated brake 120' is also mounted on shaft 13.

Associated with the countershaft 10 is a suitable induction coil, indicated diagrammatically at 17 and connected by means of a pair of wires 18 and 19 to an electrical control box 20 which contains a major portion of an electrical circuit hereinafter described. A pair of wires 21 and 22 extends from the control box 20 to the adjustable fuel feeding device 8 of the engine 7 for the purpose of controlling the rate at which fuel is fed to the engine, as hereinafter more fully described. A third pair of wires 23 and 24 extends from the control box 20 to the solenoid 25 of a three-way air valve 26 which controls the admission of air under pressure to the clutch 15. The valve 26 is connected to an air pressure supply line 27 and is connected by means of a line 28, which communicates with a central bore in shaft 14, with the operating mechanism of the clutch 15. The valve normally connects the line 28 to an exhaust discharge line or port 29; however, when the solenoid 25 is energized, the exhaust discharge outlet is blocked and the pressure supply line 27 is connected to the line 28. The clutch 15 is of the normally en-

gaged type and is disengaged when air is supplied thereto by means of valve 26. The details of construction of the clutch 15 form no part of the present invention and any suitable form of clutch may be employed. One suitable type is illustrated diagrammatically in greater detail in Figure 4, hereinafter described.

The control box 20 is provided with three control knobs 30, 31 and 32. As hereinafter set forth in greater detail, the control knob 30 adjusts the magnitude of the drill stem supporting torque, the control knob 31 adjusts the relative percentage magnitude of change of the drill stem supporting torque with respect to the drill stem rotating torque, and the control knob 32 adjusts the drill stem rotating torque at which the clutch 15 is automatically disengaged.

As shown diagrammatically in Figure 2, the torque responsive device 17 constitutes the pickup of a magneto-strictive torque responsive electrical circuit. While any suitable form of pickup may be employed for this purpose, the form illustrated comprises an induction coil 33 which surrounds a portion of the countershaft 10. The coil 33 is connected through lines 18 and 19 to a coil 34 which is inductively coupled to a corresponding coil 35 which forms one leg of a usual Wheatstone bridge, the other legs thereof being provided with impedances 36, 37 and 38. The input terminals 39 and 40 of the bridge may be connected to any suitable source of alternating current by lines 41 and 42.

The output terminals 43 and 44 of the bridge are illustrated as being directly connected to the terminals of the primary winding of a transformer 45, although it will be understood that, if desired, an amplifier may be interposed between the bridge and this transformer. The output of transformer 45 is rectified, through a pair of usual rectifiers 46 and 47, and the rectified voltage is impressed across lines 48 and 49.

An adjustable portion of the voltage appearing between lines 48 and 49, which, as will be understood, is proportional to the torque being transmitted through countershaft 10, is utilized to control the current flow in the previously identified lines 21 and 22, which lines supply a solenoid 50 which controls the position of the throttle valve 51 in the intake manifold 52 of the engine 7. More particularly, solenoid 50 is supplied with direct current through a conventional rectifier comprising rectifiers 53 and 54, which are supplied with alternating current from transformer 55. As shown, rectifier 53 is a diode, whereas rectifier 54 is of the grid controlled type. It will be appreciated that by varying the phase of the alternating current potential impressed between the grid 56 of rectifier 54 and its cathode with respect to that of the alternating current potential impressed between its anode and its cathode, the magnitude of current output of the rectifier 53-54 may be varied between relatively wide limits. Of course, if desired, both of rectifiers 53 and 54 may be triodes and the phases of both their grids may be simultaneously shifted with respect to their cathodes.

Any suitable means may be utilized to variably control the just-mentioned phase relations. The arrangement illustrated is conventional and comprises a transformer 57 having a secondary winding provided with a center tap which is directly connected to the cathode of rectifier 54. The terminals of the secondary winding of the transformer 57 are connected together by an impedance network comprising a resistor 58 having a

variable impedance 59 of the saturable reactor type. The junction 60 of elements 58 and 59 is connected to the grid 56 of rectifier 54 through a resistor 61 and, preferably and as illustrated, a filter condenser 62 is directly connected between the grid and cathode of the rectifier 54. It will be understood that by varying the impedance of element 59, the phase of the potential of terminal 60, and, consequently, of that of the grid 56, may be varied from an in-phase relation to the cathode potential to any desired degree of out-phase relation to the cathode potential, thereby varying the output of the rectifier, as previously mentioned. The relative polarities of transformers 55 and 57 are such that the excitation of solenoid 50 varies inversely with respect to the impedance of winding 59.

Changes in the impedance of element 59 are accomplished by variably exciting the direct current winding 63, which is connected across a resistor 64. Resistor 64 is directly connected in the anode circuit of a usual grid controlled vacuum tube 65, which is supplied with anode current from any suitable direct current source represented by conductors 66 and 67. The grid of vacuum tube 65 is variably connected, by the previously mentioned knob 31, to a potentiometer resistor 68, which, in turn, is directly connected across the previously mentioned lines 48 and 49. The cathode of vacuum tube 65 is connected to one terminal of a potentiometer resistor 69, which is supplied from a suitable direct current source represented by lines 70 and 71. The previously mentioned knob 30 controls a movable tap 72, which is also connected to one terminal of the resistor 68. The effective portion of potentiometer 69 thus serves to provide a bias voltage for the grid of vacuum tube 65, which normally maintains this grid at a desired negative potential with respect to its cathode. As a result, increases in the voltage across resistor 68 increase the current flow through tube 65 and winding 63 and thereby decrease the impedance of coil 59. This in turn increases the current through coil 50. Decreases in the voltage across resistor 68 have the reverse effect.

It will be appreciated that the above described system will be so adjusted that for a given normal table torque imposed upon the countershaft 10, the voltage imposed across the lines 21 and 22, and, therefore, effective upon the solenoid 50, will maintain the throttle valve 51 in a position which will result in the imposition of a torque on the hoisting drum of the desired magnitude. More particularly, this normal torque of the countershaft 10, required to rotate the drill stem, establishes an impedance for the winding 33 which produces a normal amount of unbalance of the previously identified Wheatstone bridge and causes the rectifier 46-47 to impress corresponding voltage across lines 48 and 49. A portion of this voltage, determined by the setting of knob 31, opposed by an adjustable bias, determined by the setting of knob 30, establishes a normal bias for the vacuum tube 65, which normal bias produces a normal current flow through the saturating winding 63 associated with the variable impedance 59. This normal current flow establishes a corresponding phase relation between the grid and cathode voltages of rectifier 54 and enables the rectifier 53-54 to pass the corresponding value of current to the solenoid 50.

The total drill stem supporting torque exerted upon the drum for any given table torque applied through the countershaft 10 may be adjusted by

rotation of knob 30. This results from the fact that adjustments of the knob 30 adjust the grid bias of tube 65 and, therefore, the current flow through the tube. This, in turn, adjusts the position of the throttle valve 51. The effect of adjustments of the control knob 30 is best illustrated diagrammatically in Figure 3, in which variations in table torque are shown plotted against variations in drum torque. If the system were so adjusted that throttle 51 was fully closed when the table torque transmitted through countershaft 10 was zero, the apparatus would have a performance curve corresponding to that indicated by the line 73. This, however, is undesirable since it results in excessive variations in the drum torque for normal table torque fluctuations. By adjusting the control knob 30, the position of the performance curve may be shifted from that indicated by the numeral 73 to such curves as those indicated by the numerals 74 and 75, in which a substantial drum torque is imposed even though no table torque exists. However, since the magnitude of change in drum torque for a given change in table torque is unchanged by these adjustments while the total drum torque is increased, the percentage change is altered somewhat.

It will be noted that the slopes of the lines 73, 74, 75, etc., are determined by the setting of the knob 31, which serves to determine the magnitude of change in drum torque for a given change in table torque. Thus, in order to reduce or increase the actual magnitude of change in drum torque for a given change in table torque, it is only necessary to adjust the position of the knob 31. For example, if it is desired to obtain a performance curve corresponding to that indicated by the numeral 76, it is only necessary to adjust knob 31 in a direction to reduce the amount of the total potential across lines 48 and 49 which is applied to the grid of tube 65. This increases the slope of the curve and the lateral position of the curve may be varied by adjusting knob 30 to produce any desired mean drum torque.

It will be understood that the electrical control circuit connecting the pickup coil 17 with the engine throttle is, in itself, merely illustrative of one of many possible forms of measuring and amplifying circuits which may be utilized to control the throttle position in accordance with variations in the torque transmitted through countershaft 10. Accordingly, other measuring and controlling circuits known to those skilled in the art may be employed in place of the type illustrated. In addition, it will be apparent that any form of electrical torque transmitting device may be substituted for the coil 17. Thus, for example, the coil 17 may be replaced by the familiar strain gauge type of pickup which may be associated, in any desired manner, either with the rotating shaft 10 or, in accordance with conventional practice, with any member which is subject to a strain proportional to the table torque.

In some cases, for one reason or another, during drilling operations the drill stem becomes wedged by falling boulders or its rotation otherwise becomes blocked, with the result that the continued application of a twisting force to the upper end of the stem will cause a twist-off or breakage of the stem. In order to avoid this difficulty, there is incorporated in the preferred form of the present invention automatic means for disengaging the table drive when the torque required to rotate the drill stem reaches a pre-

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determined maximum value. This means is controlled by the same magnetostrictive pickup coil 17 which controls the position of the throttle valve 51.

Thus, as best shown in Figure 2, the solenoid 25 of control valve 26 is adapted to be energized when current flows through the lines 23 and 24, which are extensions of the previously mentioned lines 66 and 67. The flow of current through the extensions of lines 66 and 67 is controlled by a vacuum tube 77 and a pair of relays 73 and 79. The coil of relay 78 is connected in the anode circuit of the tube 77, which is of the usual grid controlled type and which is supplied with current from the lines 66 and 67. The grid of vacuum tube 77 is variably connected by the previously mentioned knob 32 to a potentiometer resistor 80, which, in turn, is directly connected across the previously mentioned lines 48 and 49. The cathode of vacuum tube 77 is connected to a suitable source of direct current potential, indicated diagrammatically at 81, the opposite side of which is connected to one terminal of potentiometer 80. This provides a bias voltage for the grid of the vacuum tube 77, which normally maintains this grid at a desired negative potential with respect to its cathode.

The amount of grid bias supplied by the source 81 and the position of the knob 32 which controls the potentiometer 80 are so selected that when the torque transmitted through the countershaft 10 reaches a predetermined maximum value, the current flowing through the anode circuit of the tube 77 will be just sufficient to close the contact 82 of the previously mentioned relay coil 78, this relay being of the type which does not operate until a predetermined current flows through the coil. Closure of the contact 82 energizes relay coil 79 and thereby closes the contacts 83 and 84 associated therewith. Contact 83 is positioned in a holding circuit for the relay coil 79; and contact 84 closes the circuit to the valve operating solenoid lines 23 and 24, thus energizing the solenoid 25 and shifting the valve 26 to the position in which it disengages clutch 15. This will immediately reduce the torque imposed upon the drill stem, but the resulting reduction in torque will not de-energize the solenoid 25 by reason of the fact that the holding circuit for the relay coil 79 remains closed even after the contact 82 is opened. Consequently, the drive to the rotary table 9 remains disconnected until the circuit through the relay coil 79 is broken by opening the normally closed manual switch 85 in the circuit of the coil 79. The torque at which the clutch 15 is disengaged by the above described apparatus may be adjusted at will by means of the control knob 32 of the potentiometer 80, which controls the amount of potential across lines 48 and 49 which is imposed upon the grid of the tube 77.

The detailed construction of the clutch 15 of Figure 1 forms no part of the present invention and, therefore, it may be of any desired or conventional type. Thus, it may, for example, be similar to the hydraulically operated clutch illustrated and described hereinafter in connection with the form of invention shown in Figure 4.

During deep rotary well drilling operations, the drill stem rotating torque winds up the long drill stem to a substantial degree. Consequently, if when the table torque reaches a maximum desired value the table is disconnected from the

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drive engine by clutch 15, the drill stem will untwist rapidly, thereby driving the table in reverse direction at high speed. Since this is undesirable, a brake 120' is provided for preventing such reverse rotation of the table driving mechanism. This is accomplished by connecting the operating cylinder 166' of the pneumatically-operated brake 120' to a pressure line 28. This brake mechanism may be of the type illustrated in greater detail in Figures 4 and 6. The brake may be released gradually to permit a slow unwinding rotation of the drill stem by closing a manual shut-off valve 163' in line 27 and just cracking open a bleed-off valve 164' in line 165' to the brake cylinder.

In Figures 4 through 7 is illustrated an alternative form of the invention, in which the variable torque transmitting mechanism comprises an infinitely variable positive displacement hydraulic transmission having means for maintaining the torque transmitted thereby at a substantially constant value so long as the table torque remains constant. A torque responsive means is associated with the rotary table drive for modifying the torque transmitted to the hoisting drum in such a manner as to produce a characteristic curve of the type indicated by the lines 74, 75 and 76 in Figure 3.

Referring to Figure 4, the apparatus includes an engine 86, the shaft 87 of which is connected by means of a manually operated clutch 88 to a sprocket 89. The sprocket 89 is connected by means of a chain 90 to a sprocket 91 on the shaft of a positive variable displacement pump 92. The pump 92 is connected to a positive displacement pump 93 by a low pressure line 94 and a high pressure line 95. The output shaft of the pump 93 is connected by a sprocket 96 and a chain 97 to a sprocket 98 on the drill stem hoisting drum 99. Any suitable torque multiplying device may be incorporated in the connections between the sprocket 96 and the drum 99, as desired.

The shaft 87 of the engine 86 is also connected by means of a hydraulically operated clutch 100 to the input shaft 101 of a differential mechanism, indicated generally at 102. The clutch 100 is normally engaged but may be disengaged by the application of fluid under pressure in pipe 103. Thus, as shown more or less diagrammatically, the clutch 100 includes a plate 104 fixed to the shaft 101 and a plate 105 which is splined to the shaft 87. Plate 105 has a rearwardly directed annular flange forming a cylinder which receives a piston 106 fixed to the shaft 87. The right-hand end of the cylinder is closed by means of a plate 107. Operating fluid is admitted to the space between the piston 106 and the right-hand end of the cylinder by means of a passageway through the shaft 87, indicated in dotted lines at 108. This passageway communicates with the interior of an annular chamber in a stationary member 109, which surrounds the shaft 87 and is connected to the previously mentioned pipe 103. Springs 109a normally hold the clutch engaged, but, on application of fluid under pressure to the pipe 103, the clutch plate 105 is retracted to the right and thereby disconnects the drive connections between the shafts 87 and 101.

The differential mechanism 102 may be of any suitable type but, as illustrated, comprises a housing 110 which supports and journals an output shaft 111 and a tubular torque reaction shaft 112. Shaft 101 passes through the tubular shaft

112 and is journaled for independent rotation therein. The inner end of the shaft 101 carries a spur gear 113 which meshes with a plurality of planet gears 114 journaled on the tubular shaft 112 and meshing with a ring gear 115. As a result of this construction, torque may be transmitted through the differential from shaft 101 to shaft 111, it being understood, however, that shaft 111 will rotate in the opposite direction to that of shaft 101 so long as the tubular reaction shaft 112 is substantially stationary. In addition, it will be appreciated that there will be transmitted to the tubular shaft 112 a reaction force proportional to the torque transmitted to shaft 111. This reaction force, as hereinafter pointed out in greater detail, is utilized to control the torque transmitted by the hydraulic transmission 92-93.

The output shaft 111 is connected in any suitable manner, as by means of sprockets 116 and 117 and a connecting chain 118, to a conventional rotary table 119 adapted to rotate the drill stem during the drilling operation. Any suitable speed ratio change devices or reverse gears may be incorporated in the drive between the shaft 111 and the rotary table, as desired. Mounted on the outer end of shaft 111 is a brake drum 120 which, as hereinafter pointed out, is adapted to hold the shaft 111 and, therefore, the rotary table against rotation under certain circumstances.

In accordance with conventional practice, either one of the positive displacement pumps 92 and 93 may be provided with means for varying its displacement from zero to a flow of any desired magnitude in either direction. However, in the preferred form of apparatus illustrated diagrammatically in the drawings, only the pump 92 is of the variable displacement type, the pump 93 being a constant positive displacement pump.

While any desired type of positive variable displacement pump may be utilized, one satisfactory type of pump is the wobble plate type pump. Both of the pumps 92 and 93 may be of the same type, but only pump 92 need have means to adjust the angle of the wobble plate to vary its displacement. Since the detailed construction of the hydraulic pumps 92 and 93 may be conventional and forms no part of the present invention, no further illustration and description are necessary. It is sufficient to note merely that when the wobble plate of pump 92, indicated fragmentarily at 121 in Figure 5, lies in a plane at right angles to the axes of the pump cylinders, the displacement of the pump is zero and, consequently, rotation of the pump shaft by the sprocket 91 will not result in the flow of any liquid through the pump. When the angle of the wobble plate 121 is shifted in one direction from its zero displacement, flow will occur in one direction; while a shift of the wobble plate in the opposite direction from the zero displacement position will result in flow through the pump in the opposite direction.

The two pumps 92 and 93 are so connected hydraulically that the shaft of pump 93 will rotate in one direction or the other, depending upon the direction of flow of the liquid through the pumps 92 and 93, and will remain stationary when no flow occurs through the pumps. As a result, during the drilling operation the weight of the drill stem supported by the hoisting drum 99 in the usual manner will be transmitted through sprockets 96 and 98 to the shaft of pump

93 and develop a back pressure at the high pressure side of pump 93 which is proportional to that portion of the weight of the drill stem which is supported by the hoisting drum. This pressure is transmitted to the pump 92, and connections are provided for adjusting the angle of the wobble plate 121 in pump 92 to maintain the pressure developed by pump 93 at a substantially constant value, regardless of the direction of rotation or the speed of rotation of the hoisting drum. Therefore, except as hereinafter pointed out, the hydraulic transmission always tends to support a fixed weight, regardless of the rate of progress of the drilling operation.

The means for controlling the pressure developed in the high pressure line 95, which connects the pumps 92 and 93 in order to maintain that pressure at a substantially constant value, is best shown in Figure 5. As there shown, the wobble plate 121 of pump 92 is connected to a differential double-acting piston 122 by means of a ball and socket joint, indicated generally at 123. The piston 122 is slidable in a cylinder 124, the large end of which is connected by a conduit 125 to a port 126 in a pressure control valve 127. The small end of the cylinder is connected by a conduit 128 to the high pressure line 95, as best shown in Figure 4. A line 129 also connects the high pressure line 95 to a port 130 of the control valve 127. The pressure control valve is supplied with a third port 131, which is connected by means of a line 132 to the low pressure line 94.

Valve 127 includes a spool having a pair of lands 133 and 134. A coil spring 135 normally urges the valve toward the left, as viewed in Figure 5. Land 133 is adapted to control communication between conduit 125 and the conduits 129 and 132. Since the pressure existing in the high pressure line 95 is in communication with the left-hand side of the spool, the spool will assume the position in which that pressure balances the force exerted by spring 135. When that position corresponds to the position illustrated in Figure 5, port 126 is closed by land 133. Consequently, the right-hand end of cylinder 124 is in communication with the high pressure line 95 and the fluid at the left-hand end of the cylinder 124 is trapped. This locks the wobble plate 121 in whatever position it assumes at the time the spool of control valve 127 reaches the position illustrated in the drawing. In the event that an additional torque is imposed upon the pump 93 by the hoisting drum 99, the pressure in line 95 will tend to increase, thereby shifting the spool of valve 127 to the right, as viewed in Figure 5, against the action of the spring 135. This permits fluid to flow from the line 129 through the valve and line 125 to the left-hand end of cylinder 124, thereby shifting the wobble plate to a position in which it permits a greater flow of liquid through the pump 92 in a direction from the line 95 to the line 94. It will be apparent that this increased flow through pump 92 (which is then acting as a motor, driving pump 93) increases the unwinding speed of the drum 99 and, therefore, causes the drill stem to apply greater pressure to the bottom of the hole. This reduces the amount of weight of the drill stem which must be supported by the hoisting drum 99 and, consequently, reduces the pressure in line 95. As soon as the pressure in line 95 reduces to its original value, the spool of valve 127 will return to the position illustrated and thereby hold the wobble plate 121 in its

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new position of adjustment. Accordingly, the valve 127 serves to maintain the pressure in line 95 at a constant value and, therefore, the torque exerted by the pump 93 on the drum remains at a constant value. This value may be adjusted by adjusting the tension of spring 135; and means are provided for so varying the tension of spring 135 as to vary the pressure in line 95 and, consequently, the hoisting drum torque to produce the desired changes in the supported weight of the drill stem previously described.

Thus, as shown in Figures 4 and 5, the rear end of the spring 135 seats against a plunger 136, the position of which is controlled by a sliding stem 137. An adjustable cap nut 138 is threaded on the end of stem 137 and bears against a cam 139 which, as best shown in Figures 4 and 7, is fixed to the tubular torque reaction shaft 112 of the differential mechanism 102. It will be noted, as best shown in Figure 7, that the control valve 127 is mounted on a bracket 140 secured to the casing 110 of the differential mechanism. If the input shaft 101 of the differential is rotated clockwise, as viewed in Figure 7, as is normal, the torque reaction exerted on the tubular shaft 112 and, therefore, on the cam 139 tends to swing the cam in a clockwise direction. Rotation of the cam is resisted by means of a relatively heavy coil spring 141, one end of which is connected to the cam and the other to a stationary bracket 142 by means of an eye-bolt 143 and a pair of adjusting nuts 144. In order to dampen the movements of the cam 139, the cam is connected to the casing of the differential by means of a double-acting dashpot 145.

The operation of the mechanism so far described is as follows. As previously indicated, the control valve 127 normally operates to maintain the supported weight of the drill stem at a constant value independent of the speed of rotation of the hoisting drum. That value is determined for any given drill stem rotating torque transmitted through the differential 102 by the adjustment of the nut 138 on the valve plunger 137. To facilitate such adjustment, the cap nut 138 is preferably made as the hub of a hand wheel 146, as best shown in Figure 7. Any suitable means, such as a lock nut 147, may be employed to lock the hand wheel in any desired adjusted position. It will be understood that this adjustment will be changed from time to time as the weight of the drill stem increases during the drilling operation. So long as the torque required to rotate the drill stem remains constant, the cam 139 will remain stationary and, consequently, the torque on the drill stem will remain constant. In the event of an increase in the torque required to rotate the drill stem, the torque reaction exerted on the cam 139 will tend to rotate the cam in a clockwise direction, as viewed in Figure 7, thereby compressing the spring 135 and causing the control valve 127 to balance the system at a higher pressure in the line 95 and, therefore, with a greater weight supported by the hoisting drum 99 and less weight on the bottom of the hole. Reductions in the drill stem rotating force will have the reverse effect. In order to produce the desired result, the mechanism is so designed that the percentage magnitude of change in the drill stem supporting torque on the hoisting drum 99 is substantially less than the percentage magnitude of change in the drill stem rotating torque which caused that change in the drum torque; thus producing a characteristic curve of the general type indicated by the

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lines 74, 75 and 76 in Figure 3. Thus, for example, fifty per cent change in the table torque may effect only five per cent or less change in the drum torque.

In order to permit an adjustment in the rate of change in the drum torque for a given change in the table torque, a spring having a different characteristic may be substituted either for the spring 135 or spring 141, or, as is preferred, the slope of the cam 139 is adjusted to change the travel of rod 137 for a given angular movement of the cam. This is accomplished by pivoting the cam near one end by means of a bolt 148 on an arm 149 fixed to shaft 112. A second bolt 150 extends through an opening in arm 149 and a slot 151 in the cam adjacent its opposite end, and serves to clamp the cam in any desired angular position of adjustment.

It will be apparent from the above that the mean drill stem supporting torque imposed upon the hoisting drum, as well as the percentage rate of change induced in the drill stem supporting torque by a given change in the table torque, may be readily adjusted by means of hand wheel 146 and the adjustable cam 139 to produce any desired performance characteristic curve. Consequently, the mechanism of Figures 4 through 7 may be utilized to perform the same functions as that previously described in connection with Figures 1 and 2. The differential gear type of torque responsive element disclosed in Figure 4 is preferred for use in connection with the positive displacement hydraulic transmission for the reason that it can be easily designed to deliver the rather substantial force required to control the valve 127. It will be appreciated, however, that any other form of torque responsive device in the table drive may be substituted for the differential mechanism. Thus, the form of torque responsive device disclosed in Figures 1 and 2 may be substituted for the differential mechanism, or vice versa.

The mechanism for declutching the table drive when the table torque reaches a predetermined maximum value is best shown in Figures 4 through 7. As there shown, the previously mentioned pipe 103 is connected to the discharge port 152 of a control valve 153 and the inlet port 154 of the control valve is connected by means of a line 155 to the high pressure line 95.

As best shown in Figure 7, the valve 153 is provided with a plunger 156 which is urged at all times by a light spring 157 in a direction to close the port 152. The valve plunger is provided with an operating stem 158 of smaller diameter than the port 152, with the result that the pressure applied to the interior of the valve housing through line 155 assists the spring 157 in holding the valve closed. The stem 158 is connected by a lost motion connection 159 to a rod 160 which, in turn, is connected by means of a turnbuckle 161 to an aligned rod 162 that is pivotally connected to the arm 149 of shaft 112. The adjustment of the turnbuckle 161 and the length of the slot in the lost motion connection 159 are such that the valve 153 remains closed until the torque reaction on the shaft 112 rotates that shaft to the maximum desired degree in a clockwise direction, as viewed in Figure 7, at which time the movement of the shaft 112 will unseat the valve plunger 156 and thereby transmit fluid under pressure to the line 103. This disengages the clutch 100.

It will be observed that once the valve plunger 156 is unseated, the pressure acting on the left-

hand end of the plunger will tend to hold the valve open. The spring 157 is made sufficiently light so that it does not overcome this tendency. Therefore, once the valve has been opened by clockwise rotation of the shaft 112, it will remain open, regardless of the position of the shaft.

In order to close valve 156 when desired, it is necessary to close communication between the valve and line 95 by means of the hand valve 163 and open the hand valve 164 which connects the interior of valve 153 to the low pressure reservoir or the low pressure line 94. This relieves all of the pressure acting on the valve 153 and permits the spring 157 to seat the plunger 156 and thereby close the discharge port 152. As soon as the valve 153 is closed, valve 163 may be opened and valve 164 closed to restore the mechanism to normal operating condition.

Means are provided for slowly releasing the brake band 167 to permit a gradual unwinding of the drill stem. This is accomplished by connecting the discharge port of valve 153 by means of a line 165 to the operating cylinder 166 of a hydraulic brake mechanism, shown best in Figures 4 and 6. This mechanism includes the previously mentioned brake drum 120 on shaft 111, a brake band 167, a brake operating lever 168 and a piston 169 in the brake operating cylinder 166. The cylinder 166 may be pivoted by means of a pin 170 to any suitable support and contains a spring 171 which normally holds the brake band 167 in a disengaged condition. It will be apparent that when valve 153 opens, piston 169 will move to the left, as viewed in Figure 6, and apply the brake, thus preventing the previously mentioned reverse rotation of the table incident to the declutching of the table drive. The brake band 167 may be released gradually to permit a slow unwinding rotation of the drill stem by closing the previously mentioned valve 163 and just cracking open the drain valve 164 to permit a slow bleed off of the pressure in cylinder 166.

It will be appreciated that the brake mechanism employed in the form of invention illustrated in Figures 4 through 7 may be incorporated, if desired, in the form of invention illustrated in Figures 1 and 2.

In Figures 8 and 9 is diagrammatically illustrated a further form of the invention in which a variable torque transmitting mechanism comprises an engine driven externally excited eddy current clutch and in which the torque transmitted to the drum by the eddy current clutch is controlled by a torque responsive device in the table drive which is effective to vary the external excitation of the eddy current clutch. Thus, as shown in the drawings, a hoisting drum 172 is connected to the output shaft 173 of any suitable form of change-speed transmission 174. The input shaft 175 of the change-speed transmission is connected to the inner member of an eddy current clutch 176, the outer member of which is connected to the shaft 177 of an engine or motor 178. It will be understood that in accordance with conventional practice, either the inner or the outer member of the eddy current clutch 176 will be provided with a suitable field coil, indicated diagrammatically at 179, by means of which the clutch is excited. The field produced by the coil 179, since it will rotate with respect to the other member of the clutch when there is any difference in their relative speeds, will induce eddy currents in such other member

which, since they will be cutting the lines of force produced by the exciting field, will transmit an electromotive force from one of the clutch members to the other. Consequently, for any given excitation of the field coil 179, the torque transmitted from the shaft 177 to the shaft 176 will be proportional to the difference between the speeds of those shafts.

Shaft 177 is driven by the engine or motor 178 at a relatively high speed in the direction indicated by the arrow thereon, while shaft 175 will be rotated in the opposite direction during normal drilling operations by the unwinding rotation of the hoisting drum 172. The speed of rotation of the shaft 175 incident to penetration of the drill bit is relatively slow even though substantial torque multiplication is provided by the transmission 174. In accordance with the present invention, the speed of shaft 177 is made so much higher than that of the maximum speed of shaft 175 in a reverse direction that variations in the speed of shaft 175 incident to variations in the rate of drill bit penetration will have a negligible effect upon the difference between the speeds of shafts 175 and 177. Consequently, for a constant speed of shaft 177 and a constant excitation of coil 179, the torque transmitted to the drum 172 remains at a substantially constant value. This value is so selected that it supports all of the weight of the drill stem except that portion necessary to provide the desired pressure of the drill bit on the bottom of the hole. The mean value of this torque may be adjusted by adjusting the speed of the engine or motor 178. If an internal combustion engine is used, that adjustment will be effected by changing the position of a fuel feeding control device, such as an intake manifold throttle valve.

The rotary table 180 is driven in any suitable manner from a second engine 181 through a differential mechanism 182 similar to the differential mechanism 102 previously described. The tubular torque reaction shaft 183 of the differential 182 is provided with an arm 184 which is connected to a stationary bracket 185 by a heavy coil spring 186 which resists clockwise rotation of reaction shaft 183 in response to table torque reaction forces. Consequently, the angular position of the arm 184 will vary in proportion to variations in the table torque. The arm 184 is connected by a linkage 187, which includes a turnbuckle 188, to the movable arm 189 of a variable resistor 190 connected in series with the field coil 179 across a suitable source of electric potential, indicated by the lines 191 and 192. Consequently, movements of the arm 184 incident to variations in the table torque will vary the resistance in the field circuit of the eddy current clutch 176 and thereby vary the torque transmitted to the hoisting drum. The mechanism is so designed that substantial percentage increases in the table torque will effect minor percentage increases in the drum torque transmitted through the clutch 176, and vice versa.

With the above described mechanism, the torque transmitted to the drum for any given table torque may not only be adjusted by adjusting the speed of the engine or motor 178, but may also be adjusted by adjusting the turnbuckle 188. In order to adjust the percentage rate of change of the drum torque for a given percentage change in the table torque, it is only necessary to connect the linkage 187 to the arm 184 at any one of a plurality of points spaced radially from the axis of the shaft 183. For this purpose,

a plurality of apertures 193 is provided in the arm 184 for the reception of a bent end of the linkage 187.

While a differential torque responsive means is employed in connection with the mechanism of Figures 8 and 9, it will be apparent that the field excitation of clutch 176 may be controlled by the electrical torque responsive apparatus of Figure 2, in which event the coil 50 of Figure 2 will be the field coil of the eddy current clutch 176. Moreover, if desired, the differential mechanism of Figure 9 and its associated linkage 187 may be connected directly to the engine throttle 51 of Figure 2 in lieu of the electrical torque controlled device there disclosed, or the differentially controlled variable resistance 190 of Figures 8 and 9 may be utilized to control the flow of current in the throttle control solenoid of Figure 2 without departing from the spirit or scope of the invention.

While several forms of the invention have been illustrated and described herein, it will be apparent that variations in the construction and arrangement of parts may be indulged in without departing from the spirit of the invention or the scope of the appended claims.

What is claimed is:

1. Rotary well drilling mechanism including means to rotate the drill stem, an engine-driven variable positive displacement hydraulic transmission for exerting a drill stem supporting force proportional to the hydraulic pressure developed in said transmission, hydraulic means for varying the speed ratio of said transmission, a control valve mechanism for said hydraulic means responsive to the pressure developed in said transmission for maintaining said pressure at a substantially constant value by adjustment of said speed ratio, said control valve mechanism including means for exerting a substantially constant biasing force on said control valve in the direction of actuation which tends to increase the hydraulic pressure in said transmission, means for applying a force on said valve in a direction to oppose said biasing means and which varies in magnitude with the hydraulic pressure in said transmission, and a torque responsive apparatus operatively connected to said drill stem rotating means and connected to said biasing means for increasing the force exerted by said biasing means on said control valve by an amount which is approximately proportional to the torque required to rotate the drill stem.

2. Rotary well drilling mechanism including means to rotate the drill stem, an internal combustion engine having means for controlling the rate of fuel consumption, an eddy current clutch having its input shaft connected to the engine, a hoisting drum connected to the output shaft of said clutch and adapted to support a portion of the weight of the drill stem, said clutch having an externally excited field coil, means for varying the electrical current flow through said field coil, and a torque responsive apparatus operatively connected to said drill stem rotating means and connected to said current flow varying means for increasing the current in said coil by an amount approximately proportional to the torque required to rotate the drill stem.

3. Rotary well drilling mechanism including drive means for rotating the drill stem, a disengageable clutch in said drive means, a hoisting drum for supporting a portion of the weight of the drill stem, torque transmitting means capable of transmitting to the drum a substantially con-

stant torque which is independent of variations in the speed of the drum incident to changes in the rate of drill bit penetration, control means for adjusting the value of the torque transmitted by said means, a torque responsive apparatus operatively connected to said drill stem rotating means and connected to said control means for increasing the torque applied to said drum by said torque transmitting means by an amount which is approximately proportional to the torque required to rotate the drill stem, and means for disengaging said clutch when the drill stem rotating torque reaches a predetermined value.

4. Rotary well drilling mechanism including drive means for rotating the drill stem, a disengageable clutch in said drive means, a brake adapted on engagement to prevent rotation of said drill stem, a hoisting drum for supporting a portion of the weight of the drill stem, torque transmitting means capable of transmitting to the drum a substantially constant torque which is independent of variations in the speed of the drum incident to changes in the rate of drill bit penetration, control means for adjusting the value of the torque transmitted by said means, a torque responsive apparatus operatively connected to said drill stem rotating means and connected to said control means for increasing the torque applied to said drum by said torque transmitting means by an amount which is approximately proportional to the torque required to rotate the drill stem, and means for disengaging said clutch and engaging said brake when the drill stem rotating torque reaches a predetermined value.

5. In a rotary well drilling mechanism, means for rotating the drill stem including a disengageable clutch in the drive connection, means for disengaging said clutch when the torque required to rotate the table reaches a predetermined value, a torque responsive apparatus operatively connected to said drill stem rotating means and to said clutch disengaging means for actuating the clutch disengaging means when the torque required to rotate the table reaches a predetermined value, and means for preventing reverse rotation of the upper end of the drill stem when said clutch is disengaged.

6. Rotary well drilling mechanism, including means to rotate the drill stem, an engine-driven variable positive displacement hydraulic transmission for exerting a drill stem supporting force proportional to the hydraulic pressure developed in said transmission, hydraulic means for varying the speed ratio of said transmission, a control valve mechanism for said hydraulic means responsive to the pressure developed in said transmission for maintaining said pressure at a substantially constant value by adjustment of said speed ratio, said control valve mechanism including means for exerting a substantially constant biasing force on said control valve in the direction which tends to increase the hydraulic pressure in said transmission, means for applying a force on said valve in a direction to oppose said biasing means and which varies in magnitude with the hydraulic pressure in said transmission, and a torque responsive apparatus operatively connected to said drill stem rotating means and connected to said valve mechanism for increasing the total biasing force on said valve in said direction by an amount approximately proportional to the torque required to rotate the drill stem.

7. A power transmission system for use in rotary well drilling, including a hoisting drum

for sustaining at least a portion of the weight of the drill stem, a driving member for rotating the drill stem, a torque transmitting mechanism connected to the hoisting drum, control apparatus for maintaining the torque delivered to the drum by said mechanism at a substantially constant value which is independent of variations in the speed of the drum incident to changes in the rate of penetration of the drill bit, means connected to said control apparatus for adjusting the torque maintained by said control apparatus, a torque responsive apparatus operatively connected to said driving member and responsive to the torque transmitted through the member and connected to said adjusting means and continuously operating on said adjusting means to increase the torque delivered to the drum over that maintained by said control apparatus when no torque is transmitted to the stem by said member by an amount which is approximately proportional to the torque required to rotate the drill stem, and a second adjusting means associated with said control apparatus for adjusting the torque delivered to said drum by said mechanism independently of changes in the torque required to rotate the drill stem.

8. A power transmission system for use in rotary well drilling, including a hoisting drum for sustaining at least a portion of the weight of the drill stem, a driving member for rotating the drill stem, a source of motive power, a torque transmitting coupling having its input connected to the source of power and its output connected to the hoisting drum, said coupling being of the type which transmits a torque that varies with the difference between the speeds of the input and output, control apparatus for maintaining

the speed of the power source at a substantially constant value, means connected to said control apparatus for adjusting the speed maintained by said control apparatus, a torque responsive apparatus operatively connected to said driving member and responsive to the torque transmitted through said member and connected to said adjusting means and continuously operating on said adjusting means to increase the speed of the power source above that maintained by said control apparatus when no torque is transmitted to the stem by said member by an amount which is approximately proportional to the torque required to rotate the drill stem, and a second adjusting means associated with said control apparatus for adjusting the speed at which the power source is maintained by the control means independently of changes in the torque required to rotate the drill stem.

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