

March 11, 1952

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2,588,408

OIL WELL DRILLING TRANSMISSION

Filed May 8, 1947

3 Sheets-Sheet 1

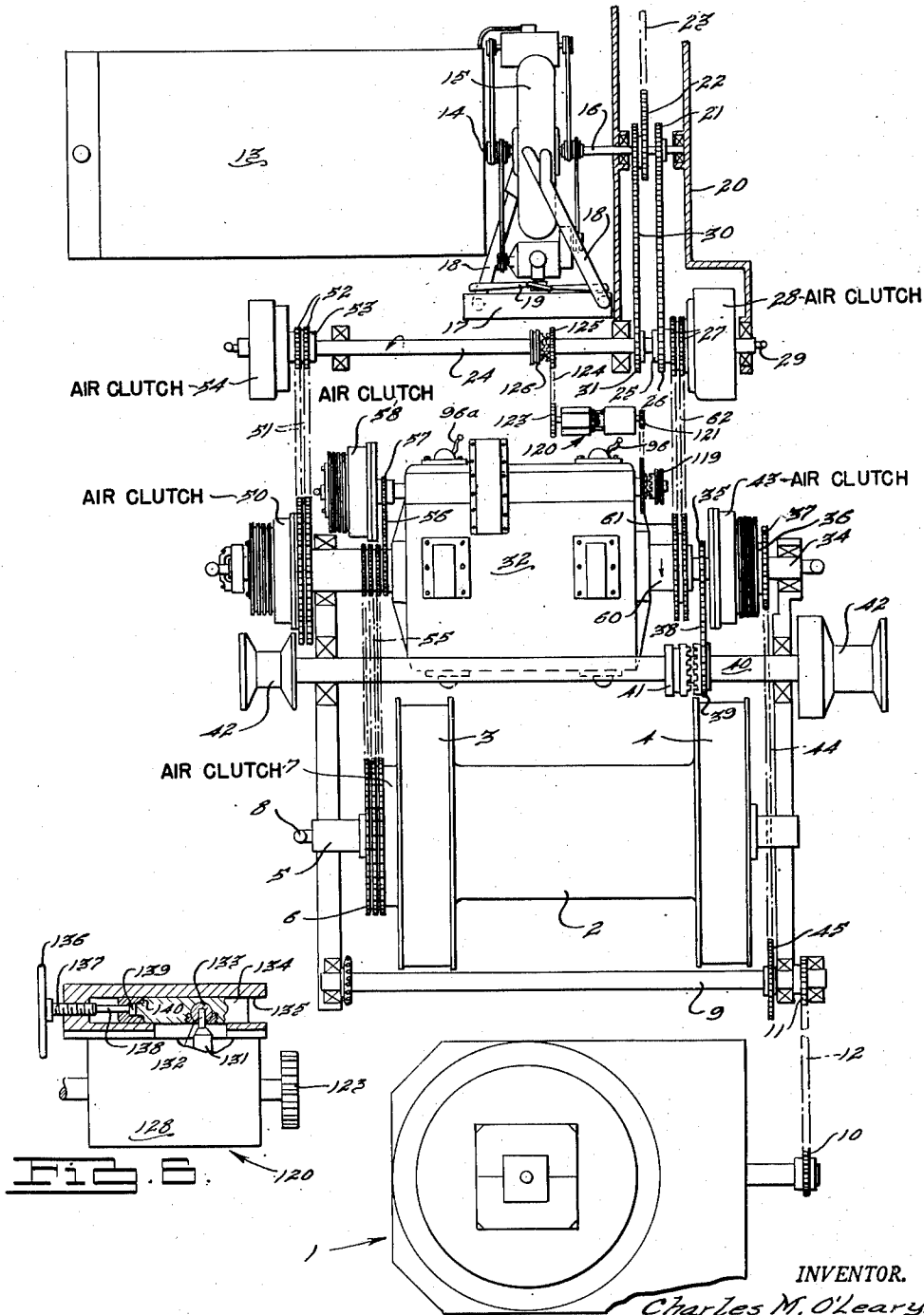


FIG. 1.

FIG. 2.

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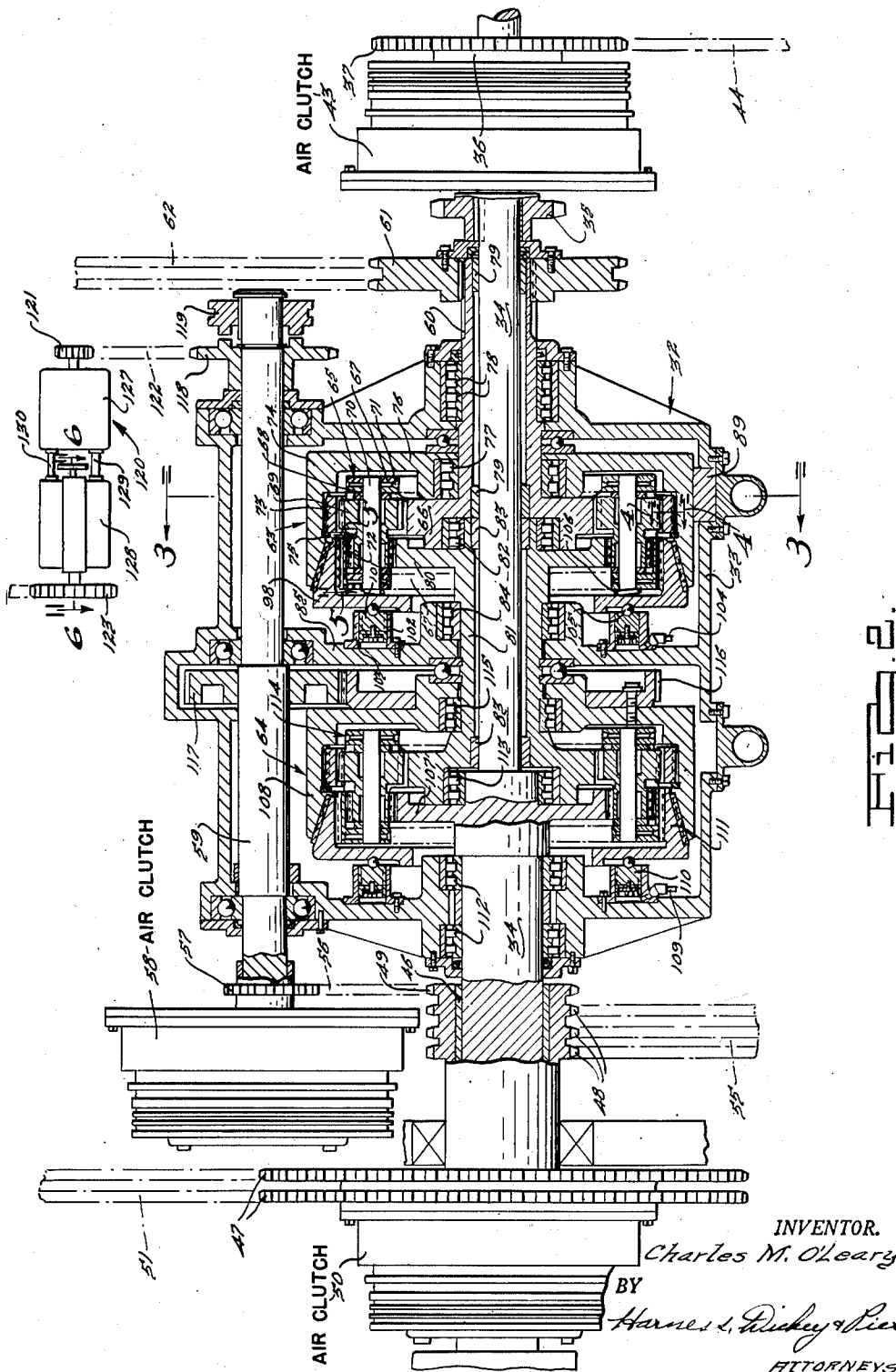
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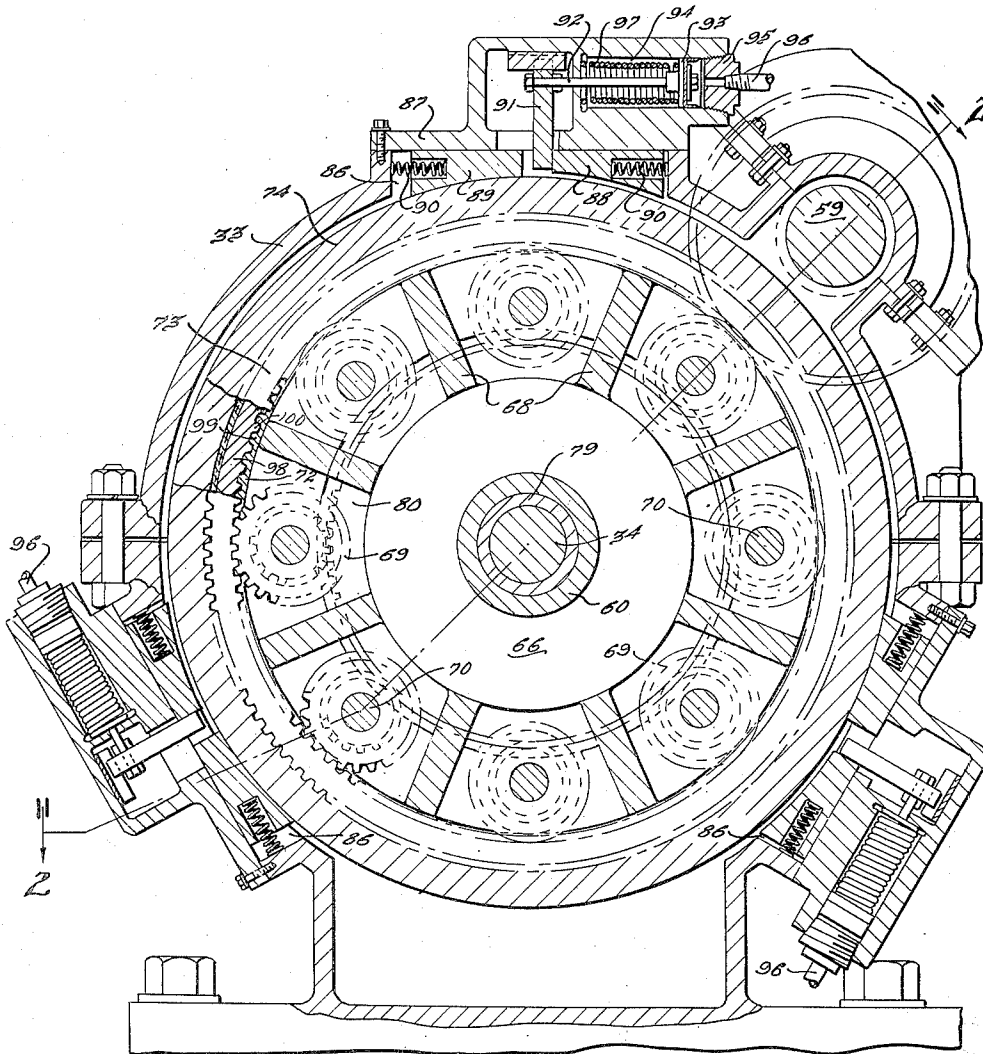


FIG. 3.

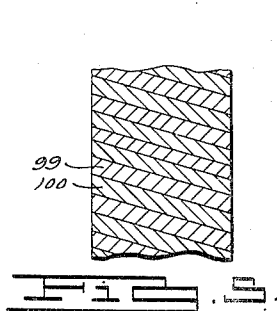


FIG. 4.

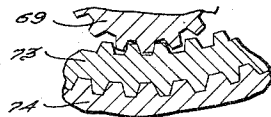


FIG. 5.

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2,588,408

OIL WELL DRILLING TRANSMISSION

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Application May 8, 1947, Serial No. 746,664

13 Claims. (Cl. 255—19)

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The present invention relates to a well drilling machine, and constitutes an improvement over apparatus for the same purpose disclosed in applicant's copending applications, Serial No. 676,450, filed June 13, 1946, and Serial No. 647,677, filed February 15, 1946.

It is the general object of the present invention to provide a rotary well drilling machine characterized by its extreme flexibility, high efficiency and ease of control.

Another object of the invention is to provide, in an apparatus of the type mentioned, an improved form of change-speed transmission mechanism capable of driving either the hoisting drum or the rotary table at a plurality of different forward speed ratios as well as at one or more reverse speed ratios, and in which these results are achieved with a minimum number of parts.

Another object of the invention is to provide apparatus of the type mentioned incorporating means for automatically controlling the pressure of the rotary drilling bit on the bottom of a hole during drilling operations; and, more particularly, to reduce the drill bit pressure in accordance with increases in the torque required to rotate the drill stem but at a smaller percentage magnitude of change.

Another object of the present invention is to provide improved and simplified means for adjusting the sensitivity of the drill stem weight control mechanism.

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

In the drawings:

Figure 1 is a general plan view of the complete apparatus, certain parts being shown more or less diagrammatically;

Figure 2 is a longitudinal section through the change-speed transmission mechanism shown in Figure 1;

Figure 3 is a transverse section taken on the line 3—3 of Figure 2;

Figure 4 is a fragmentary section taken on the line 4—4 of Figure 2;

Figure 5 is a fragmentary section taken on the line 5—5 of Figure 2; and

Figure 6 is a fragmentary section taken on the line 6—6 of Figure 2.

As best shown in Figure 1, the apparatus includes a rotary table 1, which may be of conventional construction and which is adapted to rotate the drill stem during the drilling operation in the usual manner, and a hoisting drum 2, upon

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which is wound the cable which, through suitable block and tackle, not shown, supports the major portion of the weight of the drill stem during the drilling operation, the unsupported portion of the weight of the drill stem being relied upon to develop the necessary pressure between the rotary drill bit and the bottom of the hole. The hoisting drum 2 is also utilized to hoist the drill stem from the well when it is necessary to replace or sharpen drill bits, etc.

The drum 2 is provided in the usual manner with brake flanges 3 and 4 adapted to co-operate with brake bands, not shown. The drum 2 is preferably freely journaled on a stationary shaft 5, which also journals a multiple sprocket 6 adapted to be connected to the drum by operation of an air-operated clutch 7 positioned largely within the brake flange 3. Air for operating the clutch 7 is supplied from a pipe 8 through an opening passing axially through the shaft 5 in the usual manner. The detailed construction of the rotary table, hoisting drum and pneumatic clutch 7 forms no part of the present invention and may be of any desired form. The rotary table 1 is driven from a countershaft 9 by a pair of sprockets 10 and 11 connected by a chain 12.

The apparatus is driven by two or more engines or motors, preferably of the internal combustion type, in the manner shown in greater detail in applicant's copending application, Serial No. 676,450, filed June 13, 1946. However, in Figure 1 only one of the driving engines is shown. Thus, as illustrated in Figure 1, an internal combustion engine 13 has its power output shaft 14 connected to the input of a hydrokinetic torque converter 15. The hydrokinetic torque converter may be of any desired or conventional construction, but is preferably of the type disclosed in greater detail in applicant's copending application, Serial No. 740,673, filed April 10, 1947, now Patent No. 2,558,976 issued July 3, 1951, to which reference may be had for further details of construction. It is sufficient to note here simply that the torque converter 15 automatically transmits torque at an infinitely variable torque ratio from the input shaft 14 to the output shaft 16, this ratio being maximum when the difference between the speeds of the shafts 14 and 16 is greatest and progressively diminishing as the speeds of the shafts approach each other. The torque converter also incorporates a liquid circulating system, including a radiator 17, input and output conduits 18 and a differentially operated radiator fan 19, in order to prevent development of excessive heat within the converter, all as set forth in greater

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detail and claimed in applicant's copending applications, Serial No. 666,626, filed May 2, 1946, and Serial No. 740,673, filed April 10, 1947.

The output shaft 16 of the converter extends within an elongated housing 20 and has fixed thereto within the housing a sprocket 21. A double sprocket 22 is freely journaled on the shaft 16. The sprocket 22 is connected by means of a chain 23 to one or more additional engine and hydrokinetic torque converter power sources identical in construction to that previously described.

The apparatus includes a power input countershaft 24 on which is freely journaled a sleeve 25 which carries a sprocket 26 and a double sprocket 27. The sleeve 25 may be fixed to the shaft 24 by operation of the pneumatically operated clutch 28, which is engaged upon application of air under pressure to a conduit 29 and an associated opening (not shown) passing through the shaft 24, in accordance with conventional practice. The double sprocket 22 is permanently connected to the countershaft 24 by a chain 30 and a sprocket 31 fixed to the shaft 24. As the result of this construction, the engine or engines associated with the chain 23 are at all times connected to the shaft 24, while the engine 13 is at all times connected to the sleeve 25. The shaft 24 and sleeve 25 may be driven independently by the separate power sources; or, upon engagement of clutch 28, may be connected together and the power from two or more power sources supplied to both the shaft and the sleeve.

The apparatus includes a main change-speed transmission, indicated generally at 32. The transmission 32, as best shown in Figure 2, includes a casing 33 in which is journaled a main shaft 34 which projects from the housing at both ends. A sprocket 35 is fixed on the shaft 34 near its right-hand end, and a sleeve 36 is freely journaled on the right-hand end of shaft 34 and carries a sprocket 37. Sprocket 35 is connected by means of a chain 38 to a sprocket 39, which is freely journaled on a shaft 40. Sprocket 39 may be connected to shaft 40 by means of a dog clutch 41 in order to drive a pair of cat heads 42 carried by the shaft. The sprocket 37 and its sleeve 36 may be fixed to the shaft 34 by means of an air-operated clutch 43. Sprocket 37 is connected by means of a chain 44 to a sprocket 45 fixed on the previously mentioned countershaft 9, and thus serves to drive the rotary table 1. The left-hand end of the shaft 34 has freely journaled thereon a sleeve 46 on which are mounted a multiple sprocket 47, a second multiple sprocket 48 and a single sprocket 49. All three of these sprockets, 47, 48 and 49 being fixed to sleeve 46, rotate together. An air-operated clutch 50 will, on actuation, connect the sleeve 46 to the shaft 34. The multiple sprocket 47 is connected by means of chains 51 to a multiple sprocket 52 carried by a sleeve 53 which is freely journaled on the shaft 24. An air-operated clutch 54 will, on actuation, connect the sleeve 53 to the power input countershaft 24. The multiple sprocket 48 is connected by chains 55 to the multiple sprocket 6 of the hoisting drum 2. The single sprocket 49 is connected by means of a chain 56 to a sprocket 57 carried by a sleeve which is freely journaled on a countershaft 59. The sleeve and its sprocket 57 may be connected to the shaft 59 by engagement of an air-operated clutch 58. The shaft 59 is journaled within the casing 33 and projects at both ends therefrom.

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The transmission 32 is also provided with a tubular power input shaft 60 which surrounds the shaft 34 and projects from the right-hand end of the transmission housing, as viewed in Figure 2. The tubular shaft 60 has fixed thereto a multiple sprocket 61, which is connected by means of chains 62 to the multiple sprocket 27 on the sleeve 25.

The transmission 32 includes a pair of substantially identical planetary transmission units, indicated generally at 63 and 64, which units are connected in series between the shaft 60 and the shaft 34. The transmission unit 63 includes a planet cage member 65, which is fixed to a web 66 secured to the inner end of the tubular shaft 60. The planet cage 65 includes a pair of annular side plates 67, which plates are secured to the web 66 by means of a plurality of webs 68 which lie in radial planes about the axis of shaft 34, extend between the side plates 67, and connect the latter to the web 66. The webs 68 are circumferentially spaced, and between each adjacent pair of webs is mounted a planet gear cluster 69, the gear clusters 69 being journaled upon shafts 70 which are fixed at their ends in the side plates 67 by means of pins 71. The planet cage 65 also includes a cylindrical portion 72 which surrounds the outer edges of the webs 68 and extends from the left-hand side plate in Figure 2 inwardly over the smaller of the two gears on each of the gear clusters 69.

The larger of the two gears on each of the gear clusters 69 meshes with a ring gear 73 which is supported by a cylindrical housing member 74. Any desired form of connection may be employed between the ring gear 73 and the housing 74. However, in the preferred construction shown in the drawings, and particularly in Figure 4, the ring gear 73 is provided with external teeth which are adapted to fit mating teeth on the interior of the housing 74. Limited play is provided between the teeth on the ring gear and the housing in order to permit self-alignment of the ring gear. The ring gear is held against displacement from the housing by means of a retaining ring 75.

The housing 74 is provided with an inwardly directed end flange 76 which is journaled by means of bearings 77 on the tubular shaft 60. In this connection, it will be noted that the tubular shaft 60 is supported on the housing 33 by means of bearings 78 and the shaft 34 is journaled within the tubular shaft 60 by means of bearings 79.

The smaller of the two gears in each gear cluster 69 meshes with a sun gear 80 which is fixed to the right-hand end of a tubular shaft 81. A bearing 82 is provided between the web 66 of the planet cage 65 and a right-hand extension of the tubular shaft 81. Bearings 83 are also provided between the tubular shaft 81 and the main shaft 34. The tubular shaft 81 is journaled by means of bearings 84 on a transverse wall 85 formed within the casing.

Means are provided for selectively holding the housing 74 against rotation or connecting the housing to the planet cage in order to obtain two different speed ratios in the drive connection between the shafts 60 and 81. Thus, as best shown in Figure 3, the housing 33 is provided with three circumferentially spaced openings 86, and identical brake mechanisms are associated with each such opening. Thus, as shown at the top of Figure 3, the opening 86 is closed by a cover plate 87 having a flat inner surface. A pair of one-way brake blocks 88 and 89 is positioned within the openings and is adapted to wedge between

the inner surface of the cover plate 87 and the periphery of the cylindrical housing 74. Coil springs 90 normally urge the brake blocks 88 and 89 in wedging direction, but the brake block 88 is normally held out of wedging position by means of a dog 91 carried by a piston rod 92 which is fixed to a piston 93 slidable within a cylinder 94 formed integrally with the cover plate 87. The right-hand end of cylinder 94, as viewed in Figure 3, is closed by means of a plug 95 having an air supply pipe 96 associated therewith. A spring 97 normally acts on the piston 93 to hold the parts in the position illustrated in Figure 3, in which position the dog 91 holds the brake block 88 in inoperative position. With the parts in the position shown in Figure 3, the housing 74 is free to rotate in a counterclockwise direction, but it cannot rotate in a clockwise direction due to the fact that on such rotation the brake block 89 will automatically wedge itself between the periphery of the housing and the inner surface of the closure plate 87. If air under pressure is supplied to pipe 96, the piston 93 will shift to the left, compressing spring 97 and thereby releasing the dog 91 from the brake block 88 and engaging it with the brake block 89. Under these circumstances, counterclockwise rotation of the housing 74 is prevented by the automatic wedging of the block 88 between the periphery of the housing 74 and the inner surface of the plate 87, but the housing is then free to rotate in a clockwise direction due to the fact that the dog 91 prevents wedging action of the block 89. Three identical sets of brake blocks and associated operating mechanism are associated with the housing 74, as shown in Figure 3. Consequently, the braking forces are balanced and impose no radial loads on the elements of the transmission. As will be apparent, all three of the air supply pipes 96 associated with these brake block mechanisms will be connected to a common control valve, with the result that they are in all cases simultaneously operated.

The housing 74 may be connected to the planet cage 65 in order to establish a 1 to 1 or direct drive ratio through the transmission 63. This is accomplished by means of a cone clutch element 98 which has a helical spline connection with the cylindrical portion 72 of the planet cage and which has a cone clutch surface adapted to engage a mating cone clutch surface formed on the left-hand edge of the cylindrical housing 74.

As best shown in Figure 5, the helical splines 99 formed on the cone clutch element 98 and the mating spline teeth 100 formed on the planet cage 65 extend at such an angle that the forces imposed upon the cone clutch element 98 by the clutch face on housing 74 and the teeth 100 on planet cage 65 when the clutch is engaged, housing 74 held stationary, and power is supplied to the transmission by shaft 60 tend to effect disengagement of the clutch element 98. This facilitates clutch release. The clutch 98 is engaged by means of an annular piston 101, which is axially slidable within an annular cylinder formed by a pair of annular members 102 and 103 secured to the transverse wall 85 of the transmission. Air under pressure may be supplied to the interior of the annular cylinder through an air inlet pipe 104 to effect movement of the piston 101 to the right, as viewed in Figure 2, and, consequently, engagement of the clutch element 98 with the mating clutch surface on the housing 74. A ball thrust bearing 135 is provided between the annular piston 101 and the inwardly directed web on the clutch element

98, to eliminate friction and prevent rotation of the annular piston 101. Coil springs 106 are positioned between the inwardly directed end flange or web of the clutch element 98 and the left-hand annular ring 67 of the planet cage, and thus act to urge the clutch element 98 out of engagement with the housing 74 at all times. The springs 106 are helical springs and are positioned in suitable pockets, not shown, formed in the annular ring 67 intermediate the planet shafts 70.

As the result of the above construction, if power is supplied to shaft 60 and air is supplied to pipes 96, the ring gear housing 74 will be held stationary due to the fact that shaft 60 rotates in the direction indicated by the arrow in Figure 1 and hence tends to rotate the ring gear in a counterclockwise direction, as viewed in Figure 3. The clutch blocks 88, under these circumstances, prevent such counterclockwise rotation, and, as a result, the planetary transmission unit 63 provides a speed increasing or overdrive speed ratio connection between the tubular shafts 60 and 81. It is preferred that the relative sizes of the gears employed in the transmission be so chosen that under these circumstances shaft 81 be rotated at twice the speed of shaft 60. The direction of rotation will be the same. When the air under pressure is disconnected from pipes 96 and connected to pipe 104, the clutch 98 will engage and lock the transmission unit 63 in its direct drive or 1 to 1 ratio, thus fixedly connecting shafts 60 and 81.

The planetary transmission unit 64 is identical in construction and mode of operation to the transmission unit 63, previously described, except that the one-way brake blocks which correspond to the brake blocks 89 of transmission 63 are omitted and the sun gear 107 is fixed to the shaft 34. Therefore, the transmission unit 64 will either establish a 2 to 1 overdrive connection or a direct drive connection between the tubular shaft 81 and the shaft 34, depending upon whether or not air is supplied to the brake block operating mechanism associated with the ring gear housing 108 of the transmission unit 64 or to the pipe 109 which supplies air to the annular piston 110 for the cone clutch element 111.

Since the transmission unit 64 is otherwise similar to the transmission unit 63, previously described, further description and illustration are believed unnecessary. However, hereinafter the one-way brake blocks for transmission 64, which correspond to the brake blocks 88 associated with the transmission unit 63, and the air supply pipes for transmission 64, which correspond to pipes 96 of transmission 63, will be referred to by the numerals 88a and 96a, respectively. The left-hand end of the main shaft 34 is journaled by means of bearings 112 in the left-hand end of the housing 33. A bearing 113 is provided between the shaft 34 and the planet cage 114 of transmission 64, and a bearing 115 is provided between the ring gear housing 108 and the tubular shaft 81.

In addition to the above, the planetary transmission unit 64 differs from the unit 63 in that the ring gear housing 108 of the transmission unit 64 has secured thereto a gear 116 which meshes with a gear 117 fixed to the previously mentioned countershaft 59.

The right-hand extremity of the shaft 59 projects from the housing 33 and has freely journaled thereto a sprocket 118, which may be clutched to the shaft by means of a manually

operable dog clutch 119. The sprocket 118 is connected to the power input countershaft 24 through an infinitely variable positive displacement hydraulic transmission, indicated generally at 120, one sprocket, 121, of the transmission being connected to the sprocket 118 by a chain 122, and the other sprocket, 123, of the transmission being connected by a chain 124 to a sprocket 125 freely journaled on the shaft 24. A manually operable dog clutch 126 is adapted to connect the sprocket 125 to the shaft 24.

While any suitable form of change-speed positive drive transmission may be employed as the transmission 120, the preferred form is a positive hydraulic transmission of the wobble plate type. As more or less diagrammatically illustrated in the drawings, transmissions of this type include a hydraulic pump 128 which is connected to a hydraulic motor 127 by means of piping 129 and 130. Either the pump or the motor may be of the fixed positive displacement type, but the other must be a variable displacement unit. In the preferred form of the mechanism, the motor is a fixed displacement hydraulic motor and the pump 128 is a variable positive displacement pump of the wobble plate type, wherein the stroke of the pistons is a function of the angularity of an adjustably mounted wobble plate and means are provided to fix the wobble plate at any desired angle. Thus, as shown fragmentarily in Figure 6, the pump element 128 is provided with a wobble plate member, illustrated fragmentarily at 131. The wobble plate is provided with an upstanding stud 132 on which is fitted a spherical element 133 having a cylindrical bore to receive the stud. The element 133, in turn, is fitted within a spherical socket in a reciprocating plunger 134 which is fitted within a guide member 135 secured to the top of the pump element 128. The position of the plunger 134 may be adjusted by hand wheel 136 which is secured on a threaded shaft 137, the threads of which fit a threaded opening in the guide 135. The threaded shaft is connected by means of a reduced extension 138 to the plunger in any desired manner which will permit rotation of the extension independently of the plunger. Thus, as shown, the extension is provided with a head 139 which is secured by means of an internally threaded ferrule 140 to a threaded projection on the plunger. As a result of this construction, the angularity of the wobble plate 132 may be adjusted by rotation of the hand wheel 136, and that angularity will remain fixed so long as the position of the hand wheel is unchanged. Thus, by increasing the angularity of the wobble plate, the displacement of the pump 127 for each revolution of the input sprocket 121 will be increased and the torque multiplication through the hydraulic transmission 120 decreased, and vice versa.

As will be pointed out hereinafter, the infinitely variable transmission 120 is employed when the apparatus of the present invention is utilized to control the pressure of the drill bit on the bottom of the hole during drilling operations to transmit to the hoisting drum a torque which is a function of the torque required to rotate the table 1. This is achieved by driving the table from tubular shaft 60 through the planetary transmission units 63 and 64 to shaft 34 and thence to the table and simultaneously absorbing the reaction torque imposed upon the ring gear housing 108 of transmission 64 by connecting it to the hoisting drum through the trans-

mission 120. At the same time, a larger portion of the weight of the drill stem is supported by a separate engine from that utilized to drive the table. The infinitely variable transmission 120 is utilized in place of a direct gear connection solely for the purpose of permitting an adjustment of the sensitivity of the drilling control. It is desirable, therefore, to utilize as small an infinitely variable transmission as possible in order to reduce cost, size and weight. This objective is realized in accordance with the present invention by making the gear 117 substantially smaller than the gear 116 and by making the sprocket 121 substantially smaller than the sprocket 118. This greatly reduces the torque imposed on the sprocket 121 of the hydraulic transmission 120. In addition, the sprocket 123 of the transmission 120 is connected to the input countershaft 24, which is a relatively low torque shaft. The drive connection between the shaft 24 and the drum through chains 51 and 55 greatly reduces the torque supplied to the shaft 24 by the drum 2. Since during drilling operations the speed of rotation of the drum is very low, this decrease in the torque transmitted through the transmission 120 will not result in an objectionable increase in speed of the transmission 120, and, therefore, the use of a small transmission unit is possible.

Any suitable or conventional means may be provided for controlling the supply of air to the clutches and the pipes 96, 96a, 104 and 109. It is preferred, however, that each of these clutches and control elements be supplied with air through an independent three-way valve, each valve being independently operable to connect the clutch or other control element either to a source of air under superatmospheric pressure or to the atmosphere. Solenoid operated valves are preferred for this purpose, since they may be controlled remotely from the driller's station. Since such control systems are well known and may be of any conventional or desired arrangement, further illustration or description is believed unnecessary. However, one suitable type of control system is disclosed in applicant's copending application, Serial No. 647,677, filed February 15, 1946.

While the speed ratios of the various drive connections through the apparatus may be varied widely, as desired, to meet any given conditions, the operation and advantages of the invention may best be illustrated by selecting a specific embodiment of the invention. Thus, for purposes of the subsequent description of the operation of the apparatus, it will be assumed that the speed ratios of the various drive connections of the apparatus are as follows:

The ratios of the drive connections between the various converter output shafts, such as the converter output shaft 16 or the corresponding converter output shaft (not shown) associated with chain 23 and the power input countershaft 24, are all 1 to 1 or direct drive ratios. The drive connection provided by chain 62 between sprocket 27 and sprocket 61 on shaft 50 is 2 to 1, i. e. sprocket 27 rotates at twice the speed of sprocket 61. As previously indicated, the planetary transmissions 63 and 64 will each provide either a direct drive or 1 to 1 ratio from the planet cage to the sun gear or a 1 to 2 drive ratio from the planet cage to the sun gear. The drive connection from the gear 116 to the gear 117 is a 1 to 2 drive ratio. The ratio between the sprocket 118 and the sprocket 121 is a 1 to 4 ratio. The ratio between the sprocket 123 and the sprocket 125

is a 1 to 1 ratio. The drive between the sprocket 52 and the sprocket 47 is a 4 to 1 ratio. The drive between the sprocket 48 and the drum sprocket 6 is a 3 to 1 ratio. The drives between the shaft 34 and both the cat head countershaft 40 and the table sprocket 10 are 1 to 1 drive ratios. The drive ratio between the sprocket 57 and the sprocket 49 is a 2 to 1 ratio.

The operation of the apparatus under the varying conditions encountered in deep rotary well drilling is as follows:

Hoisting operations

Low gear drive is obtained by engaging clutches 7 and 54, thus establishing a drive connection between the power input countershaft 24 and the drum independently of the transmission 32. At this time, the clutches 43, 50, 58, 119 and 126 are disengaged and the condition of the planetary gear units 63 and 64 is immaterial. This provides a torque multiplication of 12 to 1 between the shaft 24 and the drum, and the engine torque is multiplied by the hydrokinetic torque converter 15 before it is transmitted to the shaft 24, thus providing a further increase in the torque delivered to the drum. Any desired number of engines may be connected to the drum under these conditions. Thus, if clutch 28 is disengaged, shaft 24 will be driven only by the engine or engines associated with chain 23. By engaging clutch 28, engine 13 may also be connected to the shaft 24.

Second, third and fourth speed ratio drives to the drum are provided through the transmission 32. Thus, to establish the second speed ratio, clutch 50 is engaged and clutch 54 disengaged. The drive is then from the sprockets 27 to the sprockets 61 on the tubular input shaft 60 of the transmission 32. The sprockets 27 may be driven solely from the engine 13 or, by engagement of the clutch 28, from the additional engines associated with the chain 23. For second speed, both of the planetary transmission units 63 and 64 are placed in their 1 to 1 or direct drive ratios by engaging the clutch elements 98 and 111. This is accomplished by supplying air under pressure to the pipes 104 and 109. No air is supplied to the one-way brake block control pipes 96 or 96a. As a result, the tubular input shaft 60 is directly connected to the shaft 34 and thus drives the sprocket 48, which, in turn, is connected by chain 55 to the drum. Since the drive between the sprockets 27 and 61 is a 2 to 1 ratio and the drive between the sprockets 48 and 6 is a 3 to 1 ratio, this establishes a torque multiplication of 6 to 1 between the sprockets 27 and the drum.

Third speed is obtained by disconnecting line 104 from the source of air under pressure and supplying air under pressure to the pipes 96. This disengages the cone clutch 98 of transmission unit 63 and permits the one-way brake blocks 88 to engage the ring gear housing 74 and hold it against rotation, thus establishing a 1 to 2 overdrive connection between the tubular shaft 60 and the tubular shaft 81. The remaining clutches and controls remain in the condition previously described for second gear, with the result that the tubular shaft 81 is directly connected through transmission unit 64 to the shaft 34 and thence to the sprockets 48. This establishes a 3 to 1 torque multiplication between the sprockets 27 and the drum.

Fourth or high speed is similarly obtained by disconnecting pipes 109 from the source of air under pressure and connecting air under pres-

sure to the pipes 96a which render effective the one-way clutch blocks 88a associated with ring gear housing 108 of transmission unit 64. This establishes a 1 to 2 overdrive connection between the shaft 81 and the shaft 34, thereby providing a 1½ to 1 torque multiplication between the sprockets 47 and the drum. The remaining clutches and control elements will remain in the condition described in connection with third gear.

It will be noted that by providing a low gear drive independent of the transmission 32 and by providing substantial torque multiplication between the output of transmission 32 and the hoisting drum, the transmission 32 is relieved of the heaviest loads and hence may be relatively small in size and light in weight.

To drive the drum in reverse, clutches 43, 50, 54, 119 and 126 are disengaged and clutches 7 and 58 engaged. Under these circumstances, the drive to the drum is from sprockets 27 to the input shaft 60 of the transmission 32 through the transmission unit 63, which may be either in its high or low gear ratio, to the transmission unit 64, which must then be in its direct or 1 to 1 ratio, and thence to the countershaft 59 through gears 116 and 117. From the countershaft, the drive is through sprocket 57, chain 56 and sprocket 49 to the sleeve which carries the sprockets 48 and thence to the drum sprockets 6. It will be noted that since the drive from the ring gear 108 of transmission 64 to the shaft 59 is a direct gear drive while the return drive from the shaft 59 to the sleeve 46 is a chain drive, the direction of rotation of the sleeve 46 will be reversed with respect to that of the housing 108 and the tubular input shaft 60. This, therefore, drives the drum 2 in reverse. As previously indicated, for this purpose the transmission unit 64 is locked in its 1 to 1 ratio by supplying air under pressure to pipe 109 and thus engaging the cone clutch element 111. This establishes a positive drive connection between the tubular shaft 81 and the ring gear housing 108. If during reverse operation of the drum the transmission unit 63 is in its low gear or direct drive ratio, the torque multiplication between sprockets 27 and the drum will be 6 to 1. This condition is established by supplying air under pressure to the pipe 104 and thus engaging the cone clutch 98. To obtain a lower torque multiplication, transmission unit 63 is placed in its overdrive condition by disconnecting air under pressure from pipe 104 and connecting air under pressure to pipes 96. This provides a torque multiplication of 3 to 1 between the sprockets 27 and the drum in reverse.

While the control valves which control the various drive connections through the transmission may be operated manually in any desired manner, it is preferred for heavy duty hoisting operations to control the speed ratios of the drive connections to the drum automatically in response to the speed ratio of the hydrokinetic torque converter 15 or the similar converters associated with the additional engines (not shown) in order to maintain the speed ratio of the converter or converters within their range of maximum efficiency. Suitable means for this purpose are disclosed in applicant's above mentioned copending application, Serial No. 647,677, filed February 15, 1946. This automatic control mechanism may be arranged to control all four of the hoisting drum speed ratios, but for most practical purposes it will be sufficient to control automatically only the three higher ratios provided for the apparatus.

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During such operations as lowering the drill stem, the engines and hydrokinetic torque converters may be utilized as a brake to support or stop the lowering movement of the drill stem as a result of their connection to the hoisting drum 2 through the transmission. The transmission speed ratios may be set at any desired value, although preferably at a relatively high speed ratio in order to reduce the speed of the output shaft of the converter. The braking torque applied to the drum is then controlled by adjustment of the engine throttles, and the heat generated in the converters is dissipated by the converter liquid cooling system.

Manual drilling operations

If desired, the apparatus may be utilized in the performance of ordinary drilling operations, in which the pressure of the drill bit on the bottom of the hole is controlled manually by adjustment of the band brakes associated with the hoisting drum. In that case, the drum clutch 7 and clutches 119 and 126 are disengaged and clutch 43 engaged. The transmission then provides six different speed ratios between the output of the torque converter and the table.

The lowest speed ratio is achieved by engaging clutches 54, 58 and the cone clutch element 98 of transmission unit 63. No air is supplied to the pipes 109 or 96. Under these circumstances, the power is delivered from shaft 24 through chain 51 to the sleeve 46 and thence through chain 56 and clutch 58 to the countershaft 59. The countershaft, through gears 117 and 116, then rotates the ring gear housing 108 of the transmission unit 64 in reverse. This tends to rotate the planet cage 114 in reverse, but such reverse rotation is prevented by the one-way clutch blocks 89 of transmission unit 63 since the transmission unit 63 is locked in its 1 to 1 or direct drive ratio by engagement of the clutch element 98. Therefore, the planet cage of transmission unit 64 remains stationary and the sun gear 107 rotates forwardly, i. e. in an opposite direction to the direction of rotation of the ring gear housing 108. This drives shaft 34 and, consequently, the rotary table in a forward direction at a torque multiplication of 8 to 1 between the shaft 24 and the table drive sprocket 10. Since under these conditions the tubular shaft 60 of the transmission is held stationary, it is necessary to disconnect engine 13 from shaft 16 in any desired manner, as, for example, by draining the liquid from the converter 15. The power to drive the table is thus supplied through chain 23 from another engine (not shown). The inconvenience of disconnecting engine 13 in order to establish the above low drive ratios is of minor importance due to the fact that such a high degree of torque multiplication is seldom, if ever, required in ordinary operations.

Second speed is obtained by driving from the power input countershaft 24 through chain 51 and clutch 50 directly to the transmission shaft 34. This will provide a 4 to 1 torque multiplication between the shaft 24 and the table drive sprocket 10. Under these circumstances, clutches 43, 50 and 54 are engaged and clutches 7, 28 and 58 are disengaged. If the power is to be supplied by engine 13, clutch 28 must be engaged, in which event shaft 60 will idle forwardly without any connection to the shaft 34 since the clutch elements 98 and 111 will be disengaged and the ring gear housings 74 and 108 will idle forwardly.

Third speed is established by driving from engine 13 through the converter 15 to sleeve 25 and

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thence to the transmission input shaft 60 with both of the planetary transmission units 63 and 64 in their low gear or direct drive ratios. This will drive the shaft 34 and provide a 2 to 1 torque multiplication to the table drive sprocket 10 from the sleeve 25. Under these circumstances, clutches 50, 54, 58, 119 and 126 are disengaged. If only one engine is required to drive the table, which is usual, clutch 28 will also be disengaged and both of the cone clutch elements 98 and 111 of the transmissions 63 and 64 will be engaged by application of air under pressure to the pipes 104 and 109. No air will be supplied to the pipes 96 and 96a.

Fourth speed is established in the same manner as third speed except that cone clutch element 98 of transmission unit 63 is disengaged and air under pressure supplied to pipes 96, thus placing transmission unit 63 in its 1 to 2 overdrive ratio. This provides a torque multiplication between the sleeve 25 and the table drive sprocket 10 of 1 to 1.

Fifth speed is achieved in the same manner as fourth speed except that the cone clutch element 111 of transmission unit 64 is disengaged and air under pressure is supplied to pipes 96a of transmission unit 64, thus placing that unit in its 1 to 2 overdrive ratio. This provides a 1 to 2 overdrive connection between the sleeve 25 and the table drive sprocket 10.

Higher speed ratio drives for the table will not ordinarily be required. However, it is possible to obtain a still higher speed drive for the table by driving from the sleeve 25 into the input shaft 60 of the transmission and from shaft 24 through sleeves 53 and 46 to the countershaft 59. This is accomplished by engaging clutches 54, 58 and the cone clutch element 98 of transmission unit 63. As a result, the planet cage 114 of transmission unit 64 will rotate forwardly while the ring gear housing 108 of transmission unit 64 will rotate reversely, thereby driving the sun gear 107 forward at a high overdrive speed ratio. Clutch 28 may be engaged, but in that event the engines normally associated with chain 23 should be disconnected unless it is desired to drive the table with more than one engine.

A still higher speed ratio can also be obtained, if desired, by placing the transmission unit 63 in its overdrive ratio under the above circumstances.

Automatic drilling

When it is desired to utilize the apparatus to automatically control the pressure of the drill bit on the bottom of the hole, clutches 7 and 54 are engaged and clutch 50 disengaged. The engine and torque converter (not shown) associated with the chain 23 are then so set by adjustment of the engine throttle that they will supply, through shaft 24 and chains 51 and 55, a sufficient torque to the hoisting drum to support a desired proportion of the weight of the drill stem. This proportion of the weight so supported will remain constant for any given setting of the throttle of the engine due to the fact that the speed of the drum during drilling, and, therefore, the speed of the output shaft of the torque converter, is negligible compared with the engine speed. Under these circumstances, the torque delivered by the torque converter to the drum through the above described connections will, for all practical purposes, be totally unaffected by the minor variations in drum speed normally encountered.

The drill stem supporting torque supplied in the above described manner, which remains at a constant value for any given throttle setting, is

then augmented by supplying to the drum an additional torque which varies in accordance with variations in the torque required to rotate the drill stem. This is accomplished by driving the table from engine 13 through sleeve 25, chain 62 and transmission 32 to the shaft 34. The reaction torque thus imposed on the ring gear of transmission unit 64 is absorbed by the drum through countershaft 59, the hydraulic transmission 120, shaft 24 and chains 51 and 55. This requires that clutches 28 and 58 be disengaged. Clutches 43, 119 and 126 are engaged. No air is supplied either to the pipe 109 or to the pipe 96a of transmission unit 64. Under these circumstances, power supplied to the input shaft 60 of the transmission is transmitted through the transmission unit 63 to the planet cage 114 of the transmission unit 64. Since at this time the ring gear housing 108 of the transmission unit 64 is connected to the hoisting drum through gears 116 and 117, shaft 59, chain 122, transmission 120, shaft 24 and chains 51 and 55, it will be rotated backwardly by the drum at low speed as the cable unwinds during the drilling operation and the sun gear 107 will be rotated forwardly to effect forward rotation of the shaft 34 and, consequently, of the table 1. The table may be driven at either of two speed ratios, depending upon the setting of transmission unit 63, the lower speed ratio being a 1 to 1 drive connection between the sleeve 25 and the sprocket 10 and the higher being a 1 to 2 overdrive between the sleeve 25 and the sprocket 10. The ring gear housing 108 will be subject to a table reaction force tending to rotate it forwardly, and that force will at all times be proportional to the torque delivered to the rotary table 1. That reaction force, in turn, is transmitted through gears 116 and 117, shaft 59 and chain 122 to the hydraulic transmission 120. As a result, the transmission unit 120 will resist unwinding of the drum and thus assist in supporting the drill stem, and the value of this resistance will at all times vary in exact proportion to the table torque. The hydraulic connections between the pump 128 and the motor 127 of the transmission 120 are so arranged that the sprocket 121 of the pump tends to rotate in the opposite direction to the torque applied to sprocket 123 of the motor in order that the torque supplied by the transmission 120 to the shaft 59 will act in a direction to sustain the table torque reaction on the ring gear housing 108.

Actually, the total torque supplied to the hoisting drum from both sources will be insufficient normally to support the entire weight of the drill stem, leaving a residual portion of the weight effective to create the desired drill bit pressure. As a result, as the drilling operation progresses, the drill stem will automatically lower, causing an unwinding or reverse rotation of the drum 2. This, in turn, effects a reverse rotation of the shaft 24. Therefore, the transmission 120 actually receives power from the shaft 24 and delivers it to the transmission unit 64, rather than vice versa, and the pump element 128 drives the motor element 127.

An important feature of this automatic drill bit pressure control resides in the fact that, unlike prior mechanisms for the same purpose, the rate of change of the drill stem supporting torque is entirely independent of drum speed and the percentage rate of change of drill stem supporting torque is substantially less than the corresponding percentage rate of change in the table

torque. This eliminates objectionable hunting, which would otherwise result.

An important feature of the present apparatus resides in the provision of a relatively small, compact, infinitely variable transmission 120, by means of which the rate of change of the drill stem supporting torque for a given change in table torque may be adjusted through an infinite range to any desired value by manipulations of the hand wheel 135. Thus, the total drill stem supporting torque and its degree of responsiveness to table torque may both be adjusted to any desired degree by adjustments of the engine throttle and the position of the hand wheel 135. While an infinitely variable change-speed transmission is preferred for this purpose, it will be apparent to those skilled in the art that any of the many well recognized forms of change-speed transmissions may be employed to advantage in this connection.

Another important feature of the invention resides in the use of a relatively low torque planetary transmission for changing the speed ratios between the engines and the hoisting drum or rotary table in combination with an auxiliary drive for the lowest speed ratios independent of the transmission. Still another feature resides in the provision of an auxiliary reversing drive connection in the form of a countershaft associated with the planetary transmission, which shaft may be employed either to obtain the necessary reverse drives or to transmit to the hoisting drum the table torque reaction during automatic drilling.

While the complete apparatus involves a number of gear driving connections and a number of chain driving connections, it will be apparent to those skilled in the art that gear driving connections may be substituted for any of the chain drives, and that chain drives may be substituted for gear drives except in the planetary transmission, without departing from the principles of the invention. Likewise, other forms of planetary or differential gear sets may be substituted for the type illustrated and described.

It is highly desirable, in change-gear transmissions employed in connection with heavy duty hoisting equipment, that shifts in the gear ratio, particularly from lower to higher ratios, be accomplished during the hoisting operation with a minimum danger of dropping the hoisted load intermediate engagement of the two sets of gears. For this reason, transmissions employing disengageable gears or jaw clutches are undesirable as compared with any form of constant mesh planetary gear transmission employing friction clutches for effecting changes in the gear ratio. For that reason, this form of planetary gear transmission is preferred. However, even with such planetary transmissions it is possible, as the result of defective operation of the shifting mechanism, to disengage one gear ratio and either fail or delay in effecting the next gear ratio. The preferred form of transmission mechanism disclosed in the application incorporates an important feature designed to eliminate or reduce to a minimum this danger. Thus, it will be noted that the conical angle of the cone clutch elements which connect the planet cages to the ring gear housings may, if desired, be a self-locking angle, with the result that when once engaged they will remain engaged until subject to a separating pressure slightly in excess of that supplied by the separating springs. In that event, the cone clutches which maintain the

transmissions in their low gear ratios will remain engaged until after the one-way clutch blocks engage the ring gear housings and tend to bring the ring gear housings to a stop. The force thus exerted will be transmitted through the helical splines 99 and 100 and will be effective to complete the disengaging operation of the cone clutches. Thus, no disengagement can occur until the one-way brake blocks 89 are engaged.

While the transmission does not operate in a similar manner on shifts from a high to a low gear ratio, this is relatively unimportant in a hoisting transmission for the reason that any given hoisting operation is normally started in low gear and the only shifts in the transmission speed ratios which are required during the hoisting operation are shifts to the higher gear ratios as the load speeds up. It should be noted, however, that in order to insure a shift of the transmission from a high gear ratio to a low gear ratio under load, which might occur under unusual circumstances, it is necessary that the springs 97 be of sufficient strength to exert a total force sufficient to overcome the maximum torque reaction on the ring gear when the transmission is in its 1 to 2 overdrive ratio and the air supplied to the pipes 96 is cut off, in order that the springs may effect disengagement of the one-way brake blocks 89. In addition, the engagement of cone clutches at this time must be delayed until the one-way brake blocks 89 can be disengaged. This delay may be effected in any desired manner, as by restricting the air supply line which controls the engagement of the cone clutches.

While only one form of the invention is shown and described herein, it will be apparent that variations in the arrangement of the various parts and the design and construction thereof may be indulged in without departing from the spirit of the invention or the scope of the appended claims.

What is claimed is:

1. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one of said members and one of said input shafts, a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft, said last mentioned drive connection including a change-speed ratio positive torque transmission, and a drive connection between the other input shaft and the drum shaft independent of said change speed transmission.

2. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one of said members and one of said input shafts, a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft, said last mentioned drive connection including an infinitely variable speed ratio positive torque transmission, and a drive connection between the other input shaft and

the drum shaft independent of said change speed transmission.

3. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one of said members and one of said input shafts, a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft, said last mentioned drive connection including an infinitely variable speed ratio positive displacement hydraulic transmission, and a drive connection between the other input shaft and the drum shaft independent of said change speed transmission.

4. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one of said members and one of said input shafts, a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft, said last mentioned drive connection including a change-speed ratio positive torque transmission, the portion of said last mentioned drive connection between said third member and the said last mentioned transmission including a torque reducing drive and the portion between said last mentioned transmission and the drum shaft including a torque increasing drive in order to reduce the torque load imposed on said last mentioned transmission, and a drive connection between the other input shaft and the drum shaft independent of said change speed transmission.

5. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one of said members and one of said input shafts, a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft, said last mentioned drive connection including an infinitely variable speed ratio positive torque transmission, the portion of said last mentioned drive connection between said third member and the said last mentioned transmission including a torque reducing drive and the portion between said last mentioned transmission and the drum shaft including a torque increasing drive in order to reduce the torque load imposed on said last mentioned transmission, and a drive connection between the other input shaft and the drum shaft independent of said change speed transmission.

6. Mechanism for operating a rotary well drilling machine and automatically controlling the drill bit pressure, including a hoisting drum operating shaft, a table operating shaft, a pair of independently operable power input shafts, a differential gear set having three interconnected rotary members, a drive connection between one

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a drive connection between the second of said members and the table shaft, and a positive drive connection between the third of said members and said drum shaft; said last mentioned drive connection including an infinitely variable speed ratio positive displacement hydraulic transmission, the portion of said last mentioned drive connection between said third member and the said last mentioned transmission including a torque reducing drive and the portion between said last mentioned transmission and the drum shaft including a torque increasing drive in order to reduce the torque load imposed on said last mentioned transmission, and a drive connection between the other input shaft and the drum shaft independent of said change speed transmission.

7. A transmission for a rotary well drilling machine, comprising a supporting frame, a main shaft journaled on the frame, a table driving element journaled on the frame, a drum driving element journaled on the frame, a power input shaft, driving means including a disengageable clutch for connecting said power input shaft to said drum driving element, a planetary gear set comprising a pair of rotary gear members connected by planet gears which are journaled on a rotary planet cage member, one of said members being connected to said main shaft, driving means including a disengageable clutch for connecting said power input shaft to another of said members, a driving connection including a disengageable clutch between the third member of the planetary gear set and said drum driving element independent of said main shaft, said driving means for connecting the power input shaft to said drum element being operative independently of said third member of the planetary gear set, means selectively operable to prevent relative rotation between said members, a brake operable to hold one of said members stationary, and driving connections including independently operable clutches for connecting the main shaft to said drum driving element and said table driving element, respectively.

8. A transmission for a rotary well drilling machine, comprising a supporting frame, a main shaft journaled on the frame, a table driving element journaled on the frame, a drum driving element journaled on the frame, a power input shaft, driving means including a disengageable clutch for connecting said power input shaft to said drum driving element, a sun gear on said main shaft, a ring gear member journaled on said frame, a planet cage member journaled on said frame, planet gears journaled on said cage and operatively connecting said sun and ring gears, driving means including a disengageable clutch for connecting said power input shaft to said planet cage, a reversing driving connection including a disengageable clutch between the ring gear member and said drum driving element and independent of the main shaft, a brake operable to hold said ring gear member stationary, a clutch operable to prevent relative rotation between said members, and driving connections including independently operable clutches for connecting the main shaft to said drum driving element and said table driving element, respectively.

9. A transmission for a rotary well drilling machine, comprising a supporting frame, a main shaft journaled on the frame, a table driving element journaled on the frame, a drum driving element journaled on the frame, a power input shaft, driving means including a disengageable

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clutch for connecting said power input shaft to said drum driving element, a planetary gear set comprising a pair of rotary gear members connected by planet gears which are journaled on a rotary planet cage member, one of said members being connected to said main shaft, driving means including a disengageable clutch for connecting said power input shaft to another of said members, a driving connection including a disengageable clutch and a change-speed ratio transmission between the third member of the planetary gear set and said drum driving element independent of said main shaft, means selectively operable to prevent relative rotation between said members, a brake operable to hold one of said members stationary, and driving connections including independently operable clutches for connecting the main shaft to said drum driving element and said table driving element, respectively.

10. A transmission for a rotary well drilling machine, comprising a supporting frame, a main shaft journaled on the frame, a table driving element journaled on the frame, a drum driving element journaled on the frame, a power input shaft, driving means including a disengageable clutch for connecting said power input shaft to said drum driving element, a sun gear on said main shaft, a ring gear member journaled on said frame, a planet cage member journaled on said frame, planet gears journaled on said cage and operatively connecting said sun and ring gears, driving means including a disengageable clutch for connecting said power input shaft to said planet cage, a driving connection between the ring gear member and said drum driving element independent of the main shaft and including a disengageable clutch and a change-speed ratio transmission, a brake operable to hold said ring gear member stationary, a clutch operable to prevent relative rotation between said members, and driving connections including independently operable clutches for connecting the main shaft to said drum driving element and said table driving element, respectively.

11. A transmission for a rotary well drilling machine, comprising a supporting frame, a main shaft journaled on the frame, a table driving element journaled on the frame, a drum driving element journaled on the frame, a power input shaft, driving means including a disengageable clutch for connecting said power input shaft to said drum driving element, a sun gear on said main shaft, a ring gear member journaled on said frame, a planet cage member journaled on said frame, planet gears journaled on said cage and operatively connecting said sun and ring gears, driving means including a disengageable clutch for connecting said power input shaft to said planet cage, a reversing driving connection including a disengageable clutch between the ring gear member and said drum driving element and independent of the main shaft, a second driving connection between the ring gear member and said drum driving element independent of the main shaft and including a disengageable clutch and a change-speed ratio transmission, a brake operable to hold said ring gear member stationary, a clutch operable to prevent relative rotation between said members, and driving connections including independently operable clutches for connecting the main shaft to said drum driving element and said table driving element, respectively.

12. A two-speed transmission comprising a sun

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gear, a plurality of planet gear clusters each comprising a pair of gears of different sizes fixed together with the smaller gear of each cluster in mesh with said sun gear, a planet cage for supporting and journaling said planet clusters, an internal gear member surrounding said planet cage and in mesh with the larger gears of said clusters, a clutch element mounted on said planet cage outwardly of the planet clusters, a clutch surface on said internal gear member adapted for engagement with said element, said clutch element having a helical spline connection with said cage, the helix angle being such that the load imposed on said clutch element when engaged with said surface tends to cause disengagement of the clutch element from said surface when torque is applied to the planet cage and resistance is offered to rotation of the internal gear, means for engaging and disengaging said clutch element and surface, and a brake adapted on operation to hold said internal gear member against rotation.

13. A two-speed transmission comprising a sun gear, a plurality of planet gear clusters each comprising a pair of gears of different sizes fixed together with the smaller gear of each cluster in mesh with said sun gear, a planet cage for supporting and journaling said planet clusters, said cage being an integral unit comprising a pair of parallel spaced annular side plates connected by a plurality of circumferentially spaced axially extending webs, said plates being provided with aligned openings, removable pins having their ends fixed within said openings in said side plates and constituting journals for said gear clusters, said cage including a cylindrical wall portion extending axially from one of the side plates over the smaller gears of said clusters,

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an internal gear member surrounding said planet cage and in mesh with the larger gears of said clusters, a clutch element mounted on said cylindrical wall portion of said planet cage outwardly of the planet clusters, a clutch surface on the interior of said internal gear member adapted for engagement with said element, said clutch element having a helical spline connection with said cylindrical wall portion of the cage, the helix angle being such that the load imposed on said clutch element when engaged with said surface tends to cause disengagement of the clutch element from said surface when said planet cage is the driving member and resistance is offered to rotation of the internal gear, means for engaging and disengaging said clutch element and surface, and a brake adapted on operation to hold said ring gear housing against rotation.

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