

May 10, 1966

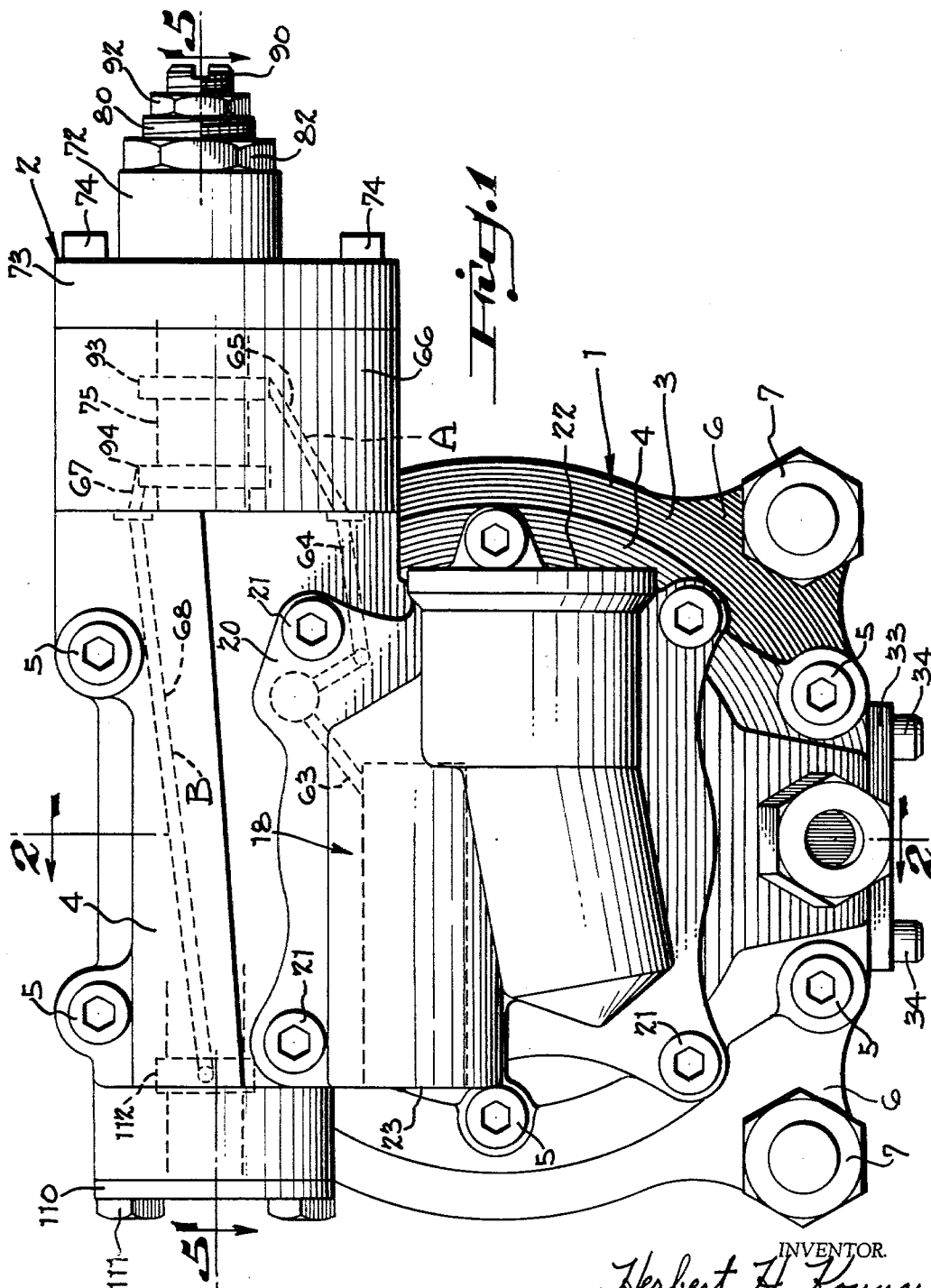
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3,250,227

TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

Filed Aug. 9, 1963

7 Sheets-Sheet 1



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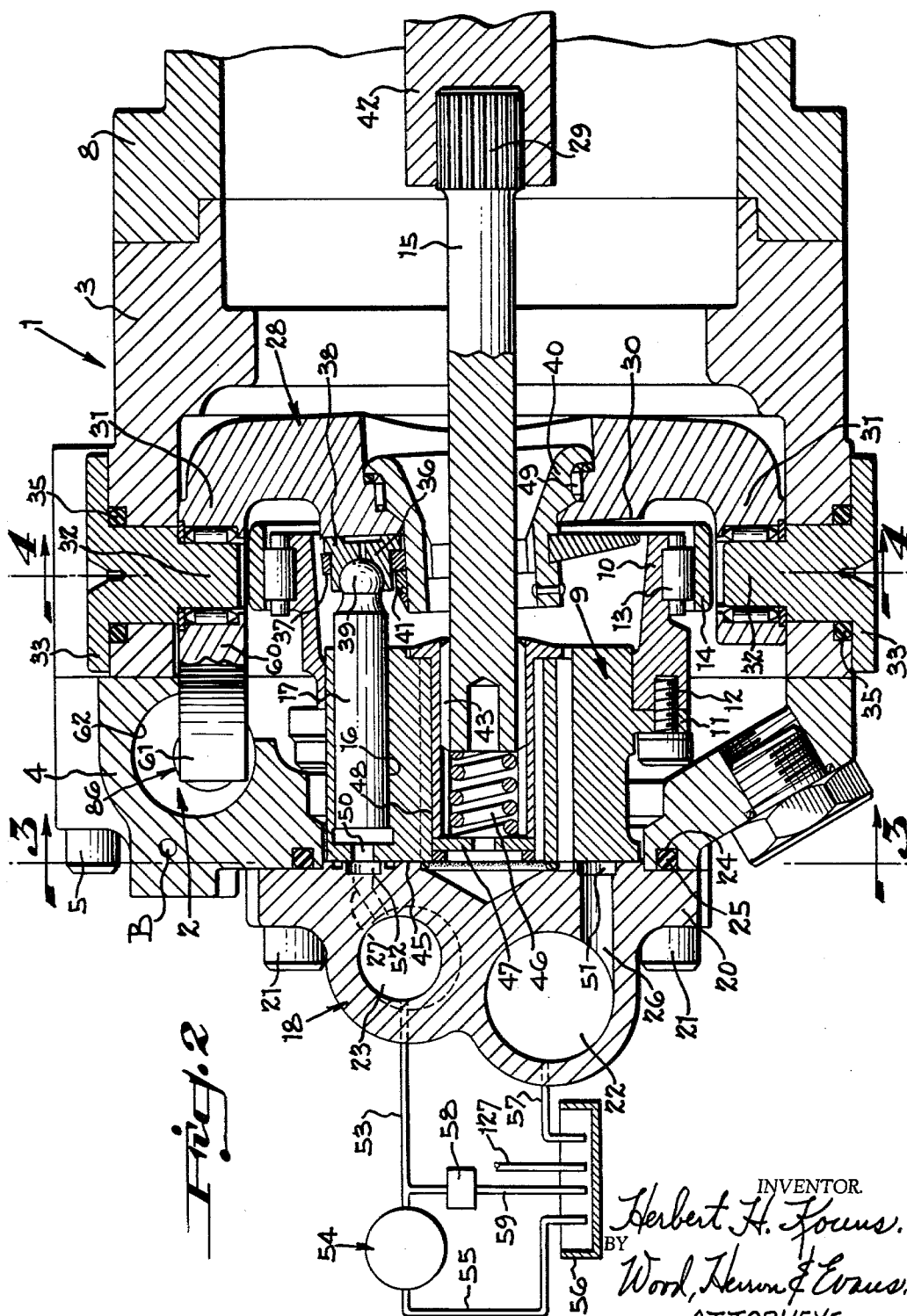
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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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7 Sheets-Sheet 2



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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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7 Sheets-Sheet 3

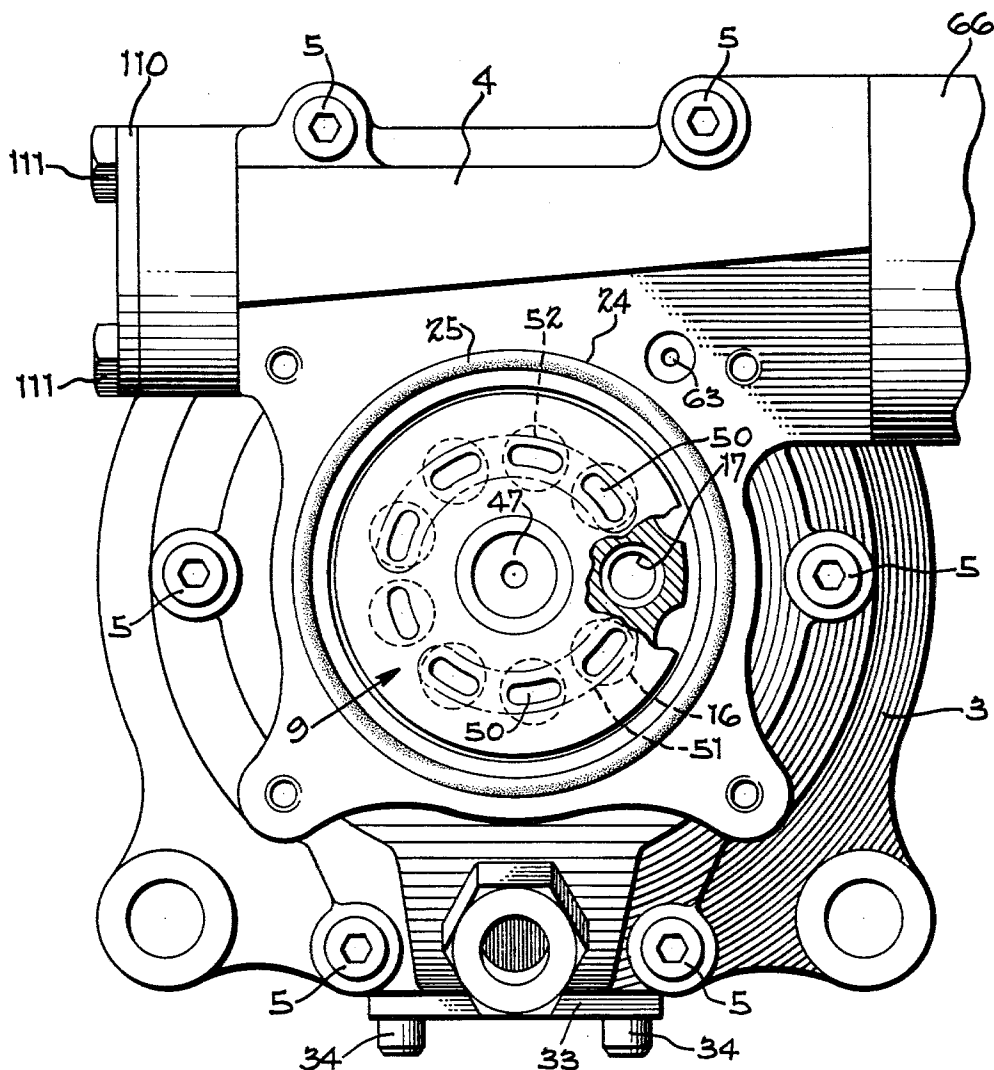


Fig. 3

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7 Sheets-Sheet 4

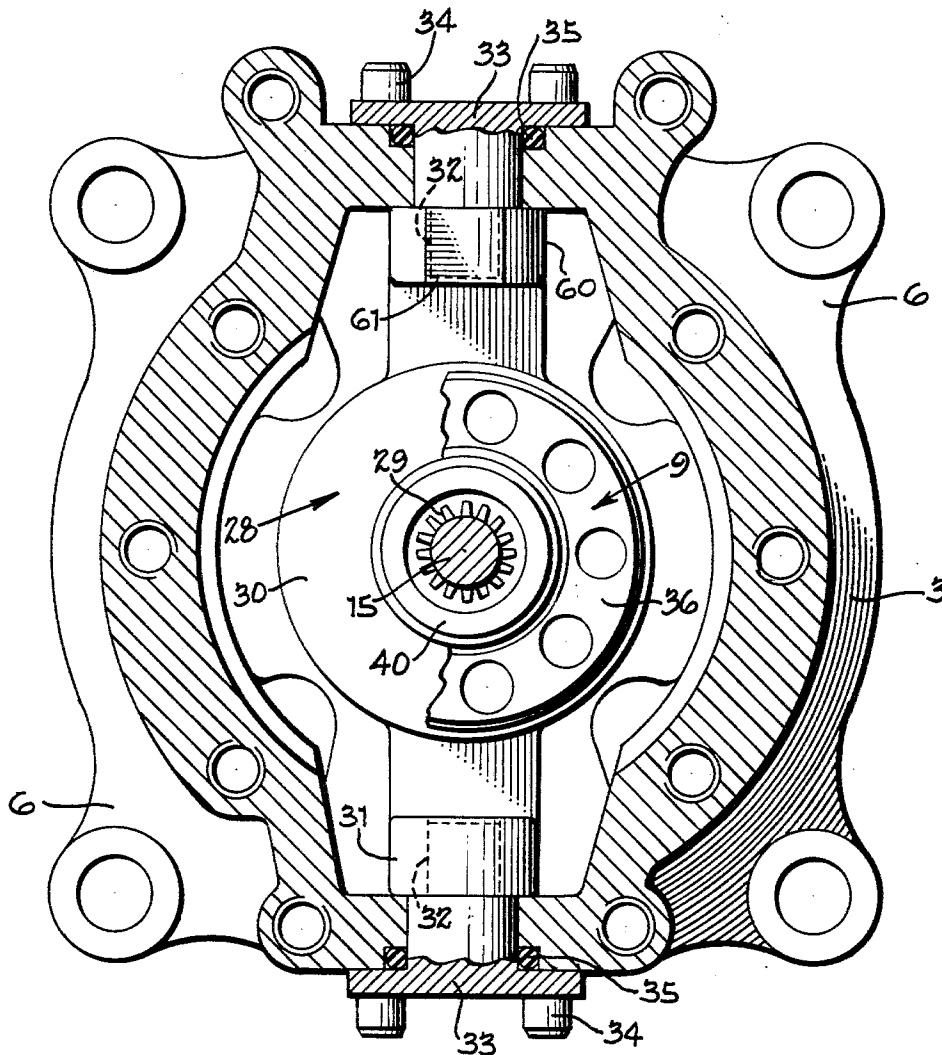


Fig. 4

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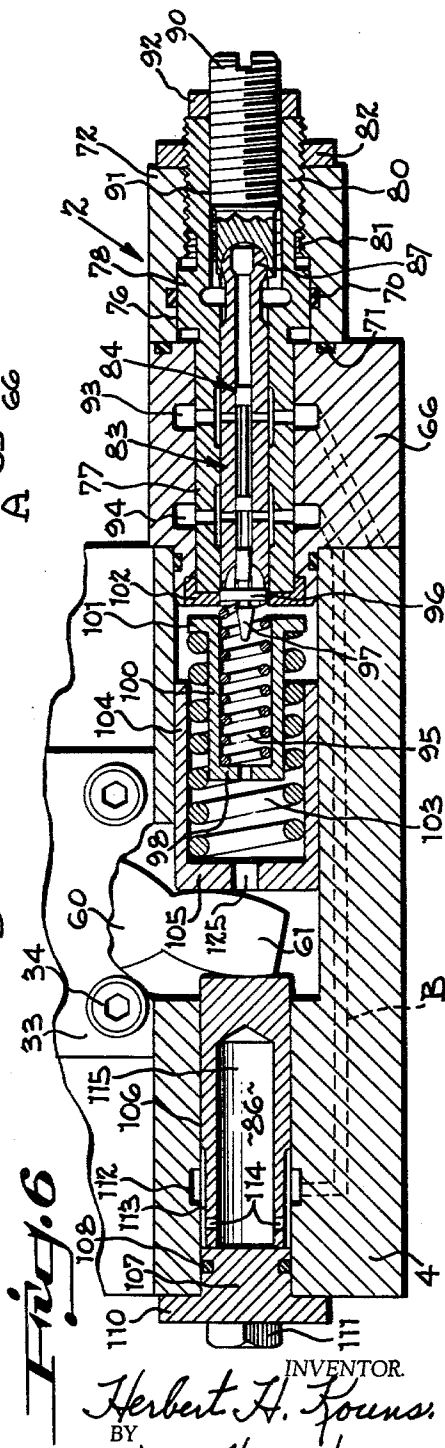
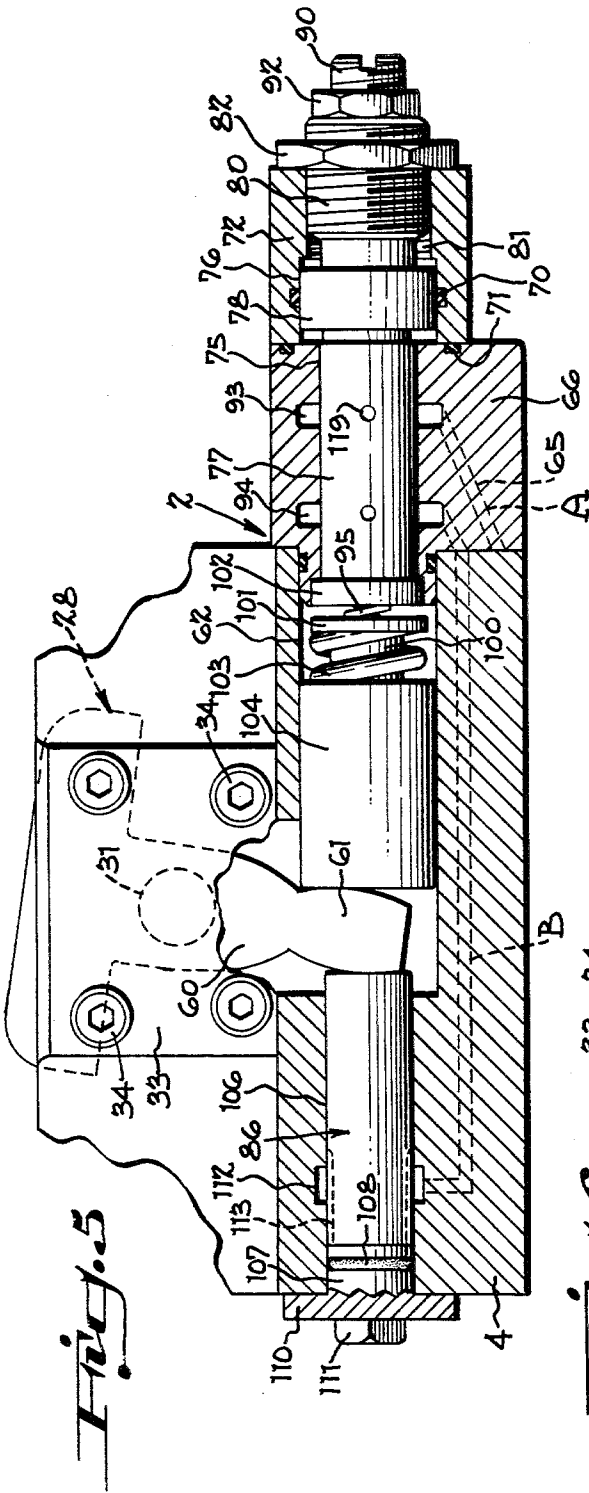
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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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7 Sheets-Sheet 5



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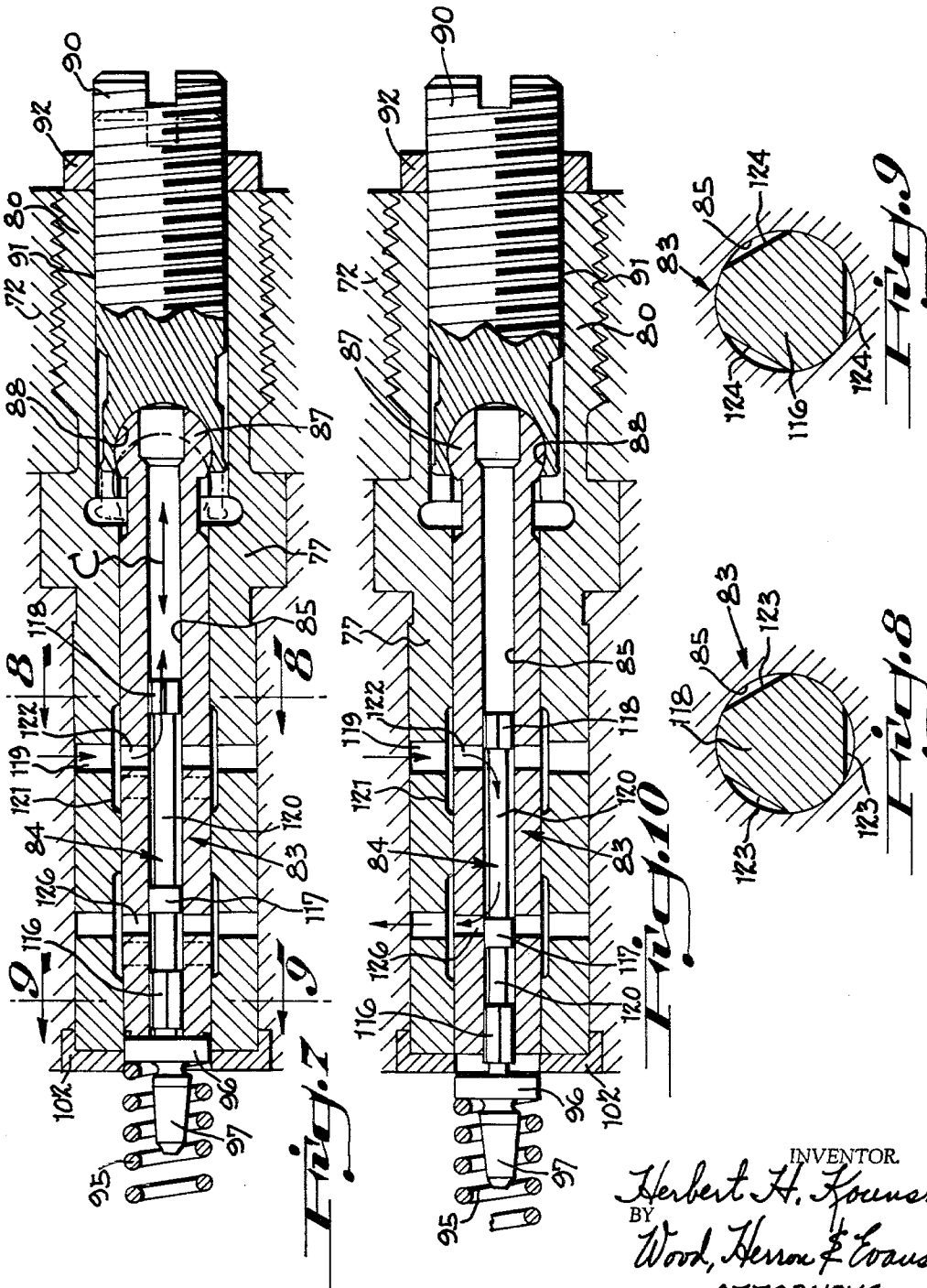
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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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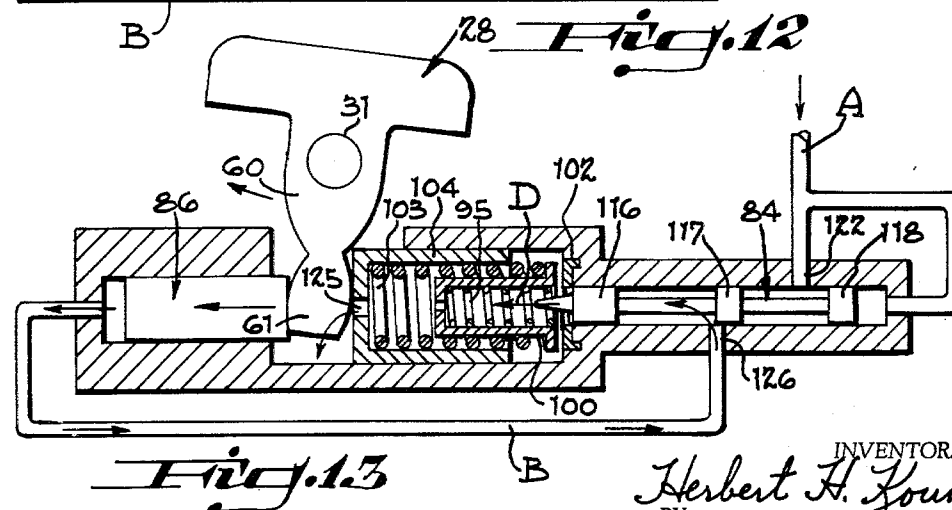
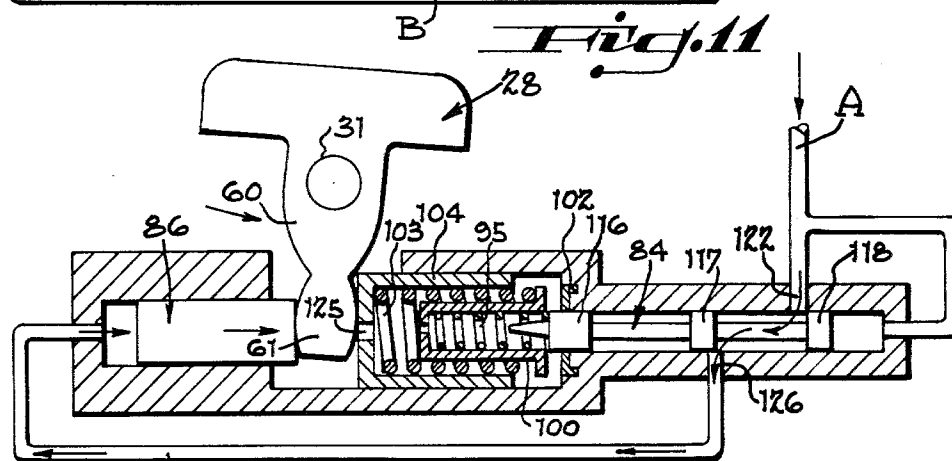
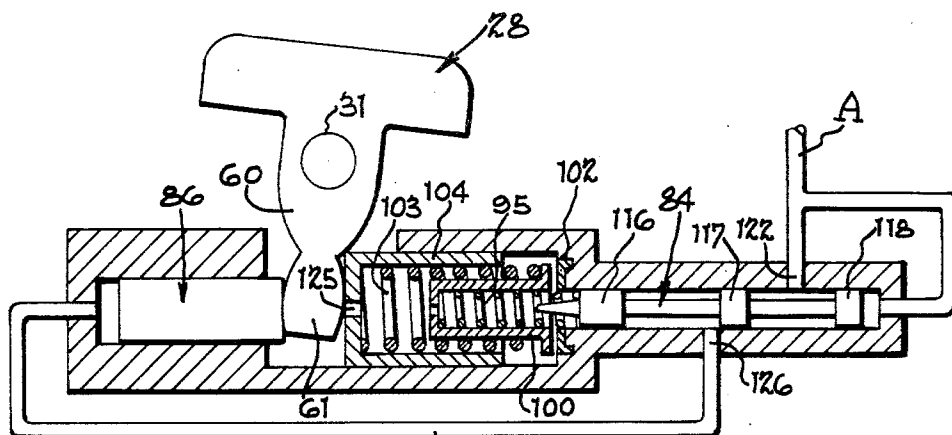
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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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7 Sheets-Sheet 7



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TORQUE CONTROL APPARATUS FOR HYDRAULIC POWER UNITS

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9 Claims. (Cl. 103-162)

This invention relates to hydraulic control apparatus for regulating the displacement of piston-type hydraulic power units by varying the stroke of the pistons in response to load variations. The present apparatus as disclosed primarily in its utility in controlling the displacement of a hydraulic pump, although the same hydraulic principles may be employed in regulating the displacement of a generally similar hydraulic motor.

The hydraulic pump, which has been selected to illustrate the principles of the torque control apparatus comprises, in general, a rotatable cylinder barrel including a series of pistons slidably confined within bores located within the cylinder barrel in a circle concentric to the axis of barrel rotation. Rotary motion is imparted to the barrel by a source of power while reciprocating motion is imparted to the pistons by an inclined cam plate, also termed a "swash plate," which slidably engages the outer ends of the pistons. The degree of inclination of the swash plate, with reference to the axis of rotation of the barrel, determines the extent of the piston stroke and pump displacement.

Torque control or horsepower limiting apparatus of this general type is known in the art and is utilized to prevent the power unit (motor or pump) from exceeding its rated horsepower. In the case of a pump of the present example, the torque control apparatus prevents the load or horsepower output of the pump from exceeding the rated horsepower of the motor which drives the pump. In other words, if the pump is driven by a motor rated at a given horsepower, the output of the pump is regulated by the torque control apparatus so that the motor is not operated above its rated capacity. Thus, if the pump is operating at a given displacement and output pressure, and is suddenly called upon (due to increased loading of the hydraulic motor which is driven by the pump) to deliver a higher output pressure, this higher pressure at the same displacement and speed may overload the power motor and cause damage to the motor and also to the hydraulic pump. However, the torque control apparatus automatically reduces the pump displacement and increases the output pressure, such that the product of pump pressure and displacement is maintained at a substantially constant predetermined value which is related to the capacity of the driving motor.

In the present example, the torque control apparatus of the invention forms a part of the pump and is utilized to regulate the inclination of the swash plate in response to the load or back pressure against which the pump is acting. By way of example, the torque control apparatus responds to back pressure in the hydraulic output passageway of the pump to reduce the stroke of the pistons, and consequently the displacement of the pump when the back pressure reaches a predetermined value. This action thus limits the amount of torque which is required to rotate the cylinder barrel while increasing the pressure from the hydraulic output line of the pump so as to prevent overloading of the driving motor or pump.

The hydraulic pumps for which the torque control apparatus is intended may be utilized as a hydraulic power source for driving the hydraulic motors used in controlling

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aircraft, for driving the motors used in actuating certain components of machine tools, and in general, for operating any hydraulic motor in which the load resistance may vary during operation.

One of the primary objectives of the present invention has been to provide a torque control apparatus of simplified, compact design which is highly sensitive to changes in the back pressure developed in the pressure or output passageway of the pump.

A further objective of the invention has been to provide a torque control apparatus including a pilot valve and an opposed stroke control piston which are disposed generally in axial alignment and housed within a single cross bore formed in the pump casing so as to simplify construction of the pump casing.

In order to control the inclination of the swash plate, the plate is provided with an actuating arm arranged to be shifted from a position of maximum inclination (maximum displacement) to a position of minimum inclination and displacement. The torque control apparatus, in general, comprises a pilot valve spool interposed in the cross bore at one side of the actuating arm, combined with a stroke control piston slidably confined in the opposite end portion of the cross bore, such that the swinging end of the swash plate arm is interposed between the valve spool and stroke control piston. The pump casing includes a pressure passageway providing communication between the high pressure output passageway of the pump and pilot valve spool. The arrangement is such that the spool is shifted axially toward the swash plate arm in response to back pressure in the output passageway. A control passageway, also formed in the pump casing, leads from the valve spool to the outer end of the stroke control piston. As back pressure increases to a predetermined setting, and the valve spool is shifted toward the arm, fluid pressure is transmitted from the pilot valve spool to the outer end of the stroke control piston, causing the piston to shift the swash plate arm from its full stroke position toward the valve spool to a shorter stroke position.

Compression springs are interposed between the swash plate arm and valve spool, the arrangement being such that, as the stroke control piston shifts the arm in the direction to decrease the stroke, the compression springs are compressed to bias the valve spool in the opposite direction, creating a servo-motion between the stroke control piston and valve spool. By virtue of this action, the valve spool becomes balanced by the back pressure acting in a direction to shift it toward the arm and the back pressure acting upon the stroke control piston in the opposite direction, thereby to hold the swash plate at a given inclination in response to a given back pressure.

A further objective of the invention has been to provide a pilot valve structure which may be adjusted in a convenient manner to vary the stroke of the pump, and also to simplify the construction of the pilot valve itself.

According to this aspect of the invention, the pilot valve comprises two concentric sleeves confined within the cross bore of the pump casing. The outer sleeve is ported and provides communication with a high pressure passageway of the pump casing and with a control passageway leading to the stroke control piston. The inner sleeve is adjustable axially with respect to the outer sleeve and includes control ports communicating with the ports of the outer sleeve. The pilot valve spool is slidably confined within the inner sleeve and controls the admission of hydraulic pressure from the high pressure outlet passageway to the control passageway leading to the stroke control valve. The arrangement is such that axial adjustment of the inner sleeve with respect to the outer sleeve

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regulates the operation of the torque control apparatus, and more specifically controls the angle of inclination of the swash plate at a given back pressure, thus controlling the operation of the pump.

The various objectives and advantages of the present invention will be more fully apparent to those skilled in the art from the following detailed description taken in conjunction with the drawings.

In the drawings:

FIGURE 1 is an end view of a variable displacement hydraulic pump which is provided with the torque control apparatus of the present invention.

FIGURE 2 is a sectional view taken along line 2—2 of FIGURE 1, further illustrating the internal construction of the pump.

FIGURE 3 is an end view similar to FIGURE 1 taken along line 3—3 of FIGURE 2 with the upper portion of the pump removed to illustrate more fully its internal construction.

FIGURE 4 is a sectional view taken along line 4—4 of FIGURE 2, further illustrating the internal construction of the pump, particularly the tiltable cam or swash plate which regulates the output of the pump.

FIGURE 5 is a fragmentary sectional view taken along line 5—5 of FIGURE 1, detailing the hydraulic torque control apparatus of the invention which regulates the output of the variable displacement pump.

FIGURE 6 is a sectional view similar to FIGURE 5, further detailing the hydraulic torque control apparatus.

FIGURE 7 is an enlarged fragmentary sectional view similar to FIGURE 6, detailing the concentric sleeves and pilot valve spool of the torque control apparatus.

FIGURE 8 is an enlarged cross sectional view taken along line 8—8 of FIGURE 7, detailing the relief areas formed on one land of the pump spool.

FIGURE 9 is a similar enlarged cross sectional view taken along line 9—9 of FIGURE 7, illustrating the relief areas of another land of the spool valve.

FIGURE 10 is a fragmentary sectional view, generally similar to FIGURE 7 and taken along line 10—10 of FIGURE 7, showing the adjustment of the inner sleeve of the control valve to increase the back pressure setting of the torque control apparatus, the valve spool being shown in the balanced position of FIGURE 7, prior to responding to the changed setting of the sleeve.

FIGURES 11—13 are diagrammatic views illustrating the hydraulic torque control apparatus in the positions which are assumed under various operating conditions.

Variable displacement pump

The torque control apparatus of this invention is illustrated in relation to a variable displacement, piston type hydraulic pump (or motor) shown in FIGURES 1—4. In the present disclosure, the structure is described primarily in its utility as a pump in which the torque compensating apparatus regulates the stroke of the pistons in response to the back pressure or load resistance imposed by a hydraulic motor which is driven by the pump. However, when utilized as a hydraulic motor, the torque control or limiting apparatus similarly changes the stroke of the pistons in response to the load so as to increase the torque and reduce the speed as the load increases. Generally speaking, the pump is similar in principle to the structure disclosed in Patents 2,696,189 and 2,699,123 and also in the pending application of Cecil E. Adams et al. for "Control for Variable Displacement Pump or Motor" Serial No. 225,484, filed on September 24, 1962.

As best shown in FIGURES 1 and 2, the variable displacement pump is indicated generally at 1 and the torque limiting apparatus, which is incorporated in the pump, is indicated generally at 2. In general, the pump comprises a sectional casing or body having a base section 3 and an intermediate section 4, the two sections being secured together by screws 5. The base section includes a flange 6 adapted to be secured by screws 7 to the mount-

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ing collar 8 of a source of power, such as an electric motor (not shown).

A rotatable cylinder barrel, indicated generally at 9 (FIGURE 2) is journaled within the sectional pump body and includes a skirt portion 10 attached to a flange 11 of barrel 9 by a series of screws 12. The cylinder barrel 9 is journaled for rotation in a roller bearing 13 having an outer race 14 confined in the base section 3. The skirt portion 10 of the barrel 9 forms the inner race of the roller bearing, adapting the cylinder barrel 9 to be rotated by a drive shaft 15 which is coupled to the power motor. The motor, in turn, is secured to the mounting collar 8 by means of screws (not shown) the arrangement being such that the cylinder barrel 9 is rotated at the same speed as the motor. The cylinder barrel 9, as explained later, includes a series of cylindrical bores 16 spaced radially in concentric relationship to the drive shaft 15, each cylindrical bore having a piston 17 adapted to reciprocate to response to the rotary motion of the barrel 9.

The intermediate section 4 of the pump body includes a head, indicated generally at 18 (FIGURES 1 and 2). The head 18 is provided with a flange 20 attached by screws 21 to the pump section 4. The head 18 includes a hydraulic fluid intake passageway 22 and a pressure outlet passageway 23. The ends of these passageways are bored and tapped (not shown) to provide connections with the pressure and return lines of the hydraulic motor or power unit (FIGURE 2) which is driven by the pump, as explained later. In order to prevent leakage of hydraulic fluid under pressure, the upper surface of pump body section 4 includes a circular groove 24 (FIGURES 2 and 3) in which is seated a conventional seal ring 25, which is maintained under compression by the flange 20 of head 18. To provide communication with the cylindrical bores 16, the intake passageway 22 of head 18 includes a bore 26 (FIGURE 2) communicating with the upper end of the rotatable cylinder barrel 8. A similar bore 27 provides communication between the upper end of barrel 8 and the outlet passageway 23 of head 18.

Reciprocating motion is imparted to the pistons 17 by a tiltable cam or swash plate, indicated generally at 28 (FIGURE 2). Swash plate 28 is generally in the form of a yoke having a circular cam surface 30. Swash plate 28 includes a pair of bearing lugs 31—31 at diametrically opposite sides which are journaled upon respective trunnions 32—32. This arrangement permits the swash plate 28 to be shifted from a neutral position to a tilted or inclined position to regulate the stroke of the pistons 17, thereby to regulate the displacement of the pump, as explained later. Each trunnion 32 includes a mounting plate 33 (FIGURES 2 and 4) disposed to the exterior of the base section 3 of the pump body and secured in place by screws 34. In order to prevent leakage, a seal ring 35 is seated in a groove formed in section 3 and engaged in compression under the mounting plate 33 of each trunnion 32.

The inner end of each piston 17 of cylinder barrel 9 includes a shoe 36 which is in sliding engagement with the cam surface 30 of swash plate 28 (FIGURE 2). It will be understood at this point, that the angle of inclination of the cam surface 30, with reference to the axis of the drive shaft 15, determines the stroke of the pistons 17, thereby regulating the output of the pump. As is best shown in FIGURE 2, there is provided a ball and socket connection between the pistons 17 and the respective shoes 36. For this purpose, the inner end of each piston 17 is machined to a spherical shape 39 interfitting a similar socket formed in the shoe 36. The shoe is compressed in this area so as to couple the spherical portion 39 to the shoe.

During rotation of the cylinder barrel 9 by the drive shaft 15, cylinder barrel 9 carries the pistons 17 and their shoes 36 in a circular path, with the shoes in sliding en-

gagement with the cam surface 30 of the swash plate 28, thus imparting reciprocating motion to the pistons. The extent of the reciprocation is determined by the angulation of the cam surface 30 of the swash plate 28. The shoes 36 are held in bearing engagement against the bearing surface 30 by a hold-down plate 37 (FIGURE 2) which is provided with bores embracing the individual shoes 36. Each shoe includes a flange 38 interposed between the hold-down plate 37 and the cam surface 30.

The hold-down plate 37 is secured to the swash plate 28 by a retainer collar 40 (FIGURE 2), which is concentric with the drive shaft 15. It will be understood that the hold-down plate 37 necessarily rotates with the cylinder barrel 9 and pistons 17. In the present example, the retainer collar 40 rotates with the cylinder barrel 9 but tilts with the swash plate 28. For this purpose, the collar 40 is journaled in the swash plate 28 upon roller bearings 49.

The drive shaft 15 includes a splined portion 29 at its inner end keyed to a coupling element 42, which is in driving connection with the motor shaft (FIGURE 2). The opposite end of drive shaft 15 includes a similar splined portion 43, which slidably interfits a bore formed in the outer portion of cylinder barrel 9. As explained later, hydraulic fluid under pressure is forced by the pistons 17 outwardly through output passageway 23 during the compression stroke of the pistons, as determined by the slant or angle of swash plate 28 with reference to the axis of drive shaft 15. At the same time, fluid is drawn from a sump by way of the intake passageway 22 during the rearward or retracting stroke of the pistons. More specifically, hydraulic fluid is drawn from intake passageway 22 into the cylinder bores 16 in one arcuate area, during which the pistons 17 are being retracted, and the hydraulic fluid is expelled under pressure during advance motion of the pistons across a second arcuate area which communicates with the output passageway 23 (FIGURE 2).

For this purpose, the outer end (FIGURE 2) of the cylinder barrel is machined to form a flat bearing surface 45 which is in sliding engagement with a similar slide bearing surface formed on the head 18. In order to prevent leakage of hydraulic fluid, the cylinder barrel 9 is urged into bearing engagement with the sealing surface of head 18 by means of a compression spring 46 having one end seated against the end of drive shaft 15, and having its opposite end seated against a washer 47 which is secured within the bore 48 of the cylinder barrel 9.

In order to provide communication between the cylinder bores 16 and the intake passageway 22 and outlet passageway 23, each cylinder bore is provided with a port 50 (FIGURES 2 and 3). The cylinder head 18 includes an arcuate intake passageway 51 (shown in broken lines in FIGURE 3) and an arcuate outlet passageway 52, also shown in broken line in FIGURE 3. The arcuate passageways 51 and 52 are concentric with the axis of shaft 15 and communicate with the ports 50 of the cylinder bores 16. The passageways 51 and 52 communicate respectively with the intake passageway 22 and output passageway 23 by way of the bores 26 and 27, as described earlier. Briefly therefore, during rotation of the cylinder barrel 9, the pistons 17 retract as their ports 50 traverse the arcuate intake slot 51 and advance as they traverse the arcuate outlet slot 52.

As shown diagrammatically in FIGURE 2, a conduit 53 connects the output passageway 23 with a hydraulic motor 54, which performs the work. From the hydraulic motor 54, the exhaust fluid flows by way of a conduit 55 to a sump 56. The sump 56 is connected by way of a conduit 57 with the intake passageway 22 of the variable displacement pump. It will be understood at this point, that drainage from the pump and also from the torque control apparatus is conducted to the sump 56. In order to protect the apparatus from accidental overloading, the pressure conduit 53 includes a pressure relief valve 58, which is connected by way of a conduit 59 to the sump

56. The relief valve 58 is conventional and is arranged to bypass the fluid from the pressure conduit 53 to the sump in the event that the hydraulic motor 54 becomes overloaded, thereby to prevent damage to the equipment.

As noted earlier, the displacement of the variable displacement pump, in terms of volumetric output, is determined by the angulation of the swash plate 28, which is mounted for rocking motion with respect to the trunnions 32. In other words, when the plane of the swash plate is disposed at right angles to the axis of the drive shaft 15, the pistons 17 remain stationary, and as the plane of the swash plate 28 is progressively inclined, then the pistons are reciprocated within the cylinder barrel to an extent determined by the inclination of the swash plate. The volumetric displacement of the pump from zero to maximum is determined by the torque control apparatus of the invention, as previously indicated at 2.

The swash plate is interconnected with the torque control apparatus 2 by an arm 60 (FIGURES 2 and 4). Arm 60 projects upwardly from the swash plate at the axis of tilting motion of the swash plate, as provided by the trunnions 32-32. The outer or swinging end of arm 60 includes an actuating head portion 61 (FIGURES 5 and 6) which is interconnected with the torque control apparatus.

Torque control apparatus

As noted earlier, the torque control apparatus of this invention regulates, in an automatic manner, the output of pump 1 in terms of volumetric displacement and pressure. By way of example, if the load on the hydraulic motor 54 (FIGURE 2) increases to an extent which normally would tend to overload the power motor of pump 1, then the torque compensator 2 shifts the swash plate 28 in a direction to decrease the stroke of the pistons 17, so as to decrease the displacement of the pump, while at the same time increasing the pressure delivered by way of the output passageway 23 to the motor 54. On the other hand, if the load on the hydraulic motor 54 decreases, then the torque compensator 2 shifts the swash plate 28 in a direction to increase the stroke of the pistons 17, thereby increasing the volumetric output of hydraulic fluid by way of the output passageway 23 so as to increase the speed of the hydraulic motor 54 with a proportionate decrease in torque.

As explained later in detail, the torque compensating apparatus 2 (FIGURES 5 and 6) is adjustable so as to enable the user to regulate the apparatus as required by the specific operating conditions of his equipment. In other words, the apparatus 2 may be adjusted to provide a relatively high volume output at correspondingly lower pressure for a work load which requires relatively high speed operation of the hydraulic motor 54, or it can be adjusted to provide higher operating pressures at proportionately lower volume if the equipment so requires.

As shown generally in FIGURES 1-3, the torque control apparatus 2 resides within a cylindrical cross bore 62 machined in the intermediate section 4 of the pump housing. As explained later in detail, the torque control apparatus 2 is actuated in response to back pressure which is developed in the pressure conduit 53 (FIGURE 2) which leads from the output passageway 23 to the hydraulic motor 54. The several passageways which interconnect the pump, with the compensating apparatus comprise bores which are machined directly into the castings which form the intermediate housing section 4 and the head 18. These passageways are disclosed diagrammatically in FIGURES 11-13 in the form of conduits, as described below.

Described generally, the head 18 is bored as at 63 (FIGURE 1) to provide a pressure passageway communicating with the output passageway 23. The intermediate section 4 of the pump housing is also bored to provide a passageway 64 which registers with the pas-

sageway 63. Passageway 64, in turn, registers with a passageway 65 formed in a collar 66, which forms a part of the torque control apparatus 2. For simplicity, the several bores 63, 64 and 65 are indicated at A in FIGURE 1 and also in FIGURES 11-13. The passageway A (section 65) communicates with the shiftable spool of the pilot valve, as described later.

From the pilot valve, hydraulic fluid is advanced by way of a passageway 67 (FIGURE 1) formed in the collar 66. Passageway 67 registers with a passageway 68 formed in the intermediate section 4 and leading to the stroke control piston, which also forms a part of the torque control apparatus. For convenience, the passageways 67 and 68 are indicated at B in FIGURE 1 and in the diagrammatic FIGURES 11-13.

Referring to FIGURES 1, 5 and 6, the collar 66, which forms a part of the torque control apparatus, is attached to the end portion of the intermediate section 4. A cap 72 having a flange 73 is secured by screws 74 to the end of the collar 66, the screws 74 passing through collar 66 into section 4. The collar 66 and the cap 72 include bores 75 and 76 which are concentric with the cross bore 62 of the intermediate housing section 4.

As detailed in FIGURES 5, 7, and 10, the torque control apparatus includes an outer ported sleeve, indicated generally at 77, mounted in the bore 75 of collar 66. The outer end portion of sleeve 77 includes a head portion 78 confined in the bore 76 of cap 72, and includes a threaded end portion 80 engaging a screw threaded bore 81 formed within the cap 72. The outer ported sleeve 77 is locked in its adjustment position by a nut 82 screwed upon its threaded end portion 80 and in clamping engagement with the end of cap 72. The cap 72 includes an annular groove in which is seated a seal ring 70 embracing the head portion 78 to prevent leakage of hydraulic fluid. A similar seal ring 71 also seated in an annular groove, seals the cap 72 with respect to the collar 66.

As explained later in detail, an inner, pilot valve sleeve, indicated generally at 83, is adjustably mounted within the outer ported sleeve 77 (FIGURES 6, 7 and 10). A pilot valve spool, indicated generally at 84, is mounted for axial motion within the bore 85 of the pilot valve sleeve 83 and controls the admission of hydraulic fluid under pressure to the stroke control piston, indicated generally at 86 in FIGURES 5 and 6. The axial position of the pilot valve sleeve 83 regulates the setting of pump 1 (in terms of displacement and output pressure) in response to the required operating characteristics (FIGURES 7 and 10).

For adjustment purposes the outer end of the adjustable pilot valve sleeve 83 includes a spherical head portion 87 confined in a spherical cavity 88 formed in the end portion of an adjustment screw 90. The adjustment screw 90 is threaded into a bore 91 formed in the end portion 80 of the outer ported sleeve 77. A lock nut 92 is threaded upon the end portion of the adjustment screw 90 and is engaged against the end of the portion 80 to lock the inner pilot valve sleeve 83 in its adjusted position.

It will be noted at this point, that the collar 66 includes an annular groove 93 communicating with the pressure passageway A and a second annular groove 94 communicating with the control passageway B. In general, the annular groove 93 supplies hydraulic fluid at output pressure from the output passageway 23 to the pilot valve spool 84 so as to shift the spool in response to back pressure. The spool, in response to its shifted position, supplies fluid pressure (passageway B) to the stroke control piston 86 which acts upon the head 61 of arm 60 so as to regulate the angular position of the swash plate 28 in response to back pressure.

Briefly, under minimum back pressure, the apparatus shifts the actuating arm 60 to the maximum stroke po-

sition, as shown in FIGURES 5 and 6, so as to provide maximum pump displacement at minimum pressure. As back pressure increases, the pilot valve spool 84 responds by transmitting additional fluid under pressure to the stroke control piston 86, thereby to shift the arm 60 in the direction indicated by the arrow (FIGURES 5 and 6) so as to decrease the stroke and correspondingly increase the pump output pressure.

The pilot valve spool 84 is normally biased toward the right (FIGURES 6, 7 and 10) by a compression-type pilot valve spring 95 seated against the head 96 formed on the end of the valve spool. The spring 95 is held in alignment with the valve spool by a tapered spur 97 which projects outwardly beyond the head 96. The opposite end of the valve spring 95 is seated against the closed end 98 of a floating hat-shaped sleeve 100. Sleeve 100 includes a flange 101 which provides a stop engageable with a cap 102 seated at the end of the outer ported sleeve 77.

A compression-type arm biasing spring 103 (FIGURES 6, 7 and 10) has one end seated against the flange 101 of sleeve 100. The arm biasing spring 103 presides within a spring guide 104 and its opposite end is seated against the closed end 105 of spring guide 104. The spring guide 104 is in the form of a cylindrical sleeve slidably confined in the cross bore 62, and its closed end 105 is in bearing engagement with the head 61 of the swash plate arm 60.

The stroke control piston 86 engages the opposite side of head 61 (FIGURES 6, 11, 12 and 13), opposing the spring guide 104, such that there is created a biasing action between the spring 103 and stroke control piston 86. The stroke control piston 86 is slidably confined in a second section 106 of cross bore 62, and is generally aligned therewith but of smaller diameter. The end of cross bore 106 includes a closure cap 107 having a seal ring 108 seated in an annular groove and under compression within bore 106. The closure cap includes a flange 110 and is secured in place by screws 111 passing through flange 110.

In order to supply hydraulic pressure from the control passageway B, the cross bore 106 includes an annular groove 112 in communication with passageway B and surrounding the stroke control piston 86 (FIGURE 6). From the annular groove 112 the hydraulic fluid pressure is transmitted by way of the longitudinal grooves 113 to the end of piston 86. The piston 86 may include radial openings 114 leading from the grooves 113 to its internal bore 115 to allow the fluid pressure to act upon the major diameter of the piston.

As best shown in FIGURES 7-10, the shiftable spool 84 of the pilot valve includes three lands indicated at 116, 117 and 118 slidably interfitting the bore 85 of the adjustable valve sleeve 83. The spool 84 includes necked portions 120 having a diameter smaller than the lands to provide the valving action, as explained later. When the pump 1 is at rest, or running at low pressure with maximum displacement, the valve spool 84 assumes the position shown in FIGURES 6, 7 and 11. In this position, hydraulic fluid from the pump flows by way of passageway A through the annular groove 93 and through the ports 119 of the outer ported sleeve 77 (FIGURE 7). From ports 119, the fluid passes to the relief groove 121 of the outer ported sleeve 77. From the relief groove 121, the hydraulic fluid passes into ports 122 of the inner pilot valve sleeve 83. From ports 122, the hydraulic fluid passes into the bore 85 of valve sleeve 83 between the lands 117 and 118 of the valve spool 84.

As explained later with reference to FIGURES 7-10 the land 118 of valve spool 84 includes relief areas 123 (FIGURE 8). The relief areas permit the fluid pressure from port 122 to pass through the bore 85 to the rearward end of spool 84, as indicated by the arrow in FIGURE 7. This permits the back pressure to act upon

the full diameter of the shiftable valve spool 84 (the land 117 being sealed with reference to the bore 85) thereby urging the valve spool 84 toward the left against its biasing spring 95.

In order to drain hydraulic fluid from the pilot valve, the land 116, at the forward end of valve spool 84 includes relief areas 124 (FIGURE 9) similar to a relief area 123 of land 118. Hydraulic fluid thus passes from the pilot valve sleeve 83, through the aperture 125 (FIGURE 6) of the spring guide 104 to be returned to the circulatory system of the pump. The drain passageway is indicated diagrammatically at 127 in FIGURE 2, leading to the sump 56 of the hydraulic system.

As explained later in detail, back pressure in conduit 53 and passageway A acts upon the pilot valve spool 84, tending to shift the spool toward the left (FIGURES 10 and 12) counter to the action of pilot valve spring 95. As the spool is moved toward the left, the land 117 uncovers the ports 126 of inner valve sleeve 83, thus providing communication from the ports 126 to the annular passageway 94 and passageway B, which communicates with the stroke control piston 86.

Operation

In order to adjust the output pressure of the pump, the inner valve sleeve 83 is shifted axially with respect to the outer ported sleeve 77, thus changing the relative position of the ports 122 and 126 with respect to the lands 116 and 117. To increase the output pressure, the lock nut 92 is loosened and the adjustment screw 90 is rotated in a direction to shift the inner valve sleeve 83 toward the left, as shown in broken lines in FIGURE 7, thus increasing the axial motion which must be imparted to the valve spool 84 under back pressure. Since the pilot valve spool 84 is biased by the valve spring 95, the back pressure (outlet passageway 23 and passageway A) necessarily is increased before the pilot valve spool 84 is shifted axially a sufficient distance to uncover the ports 126. As noted above, ports 126 deliver the hydraulic pressure to stroke control piston 86 so as to decrease displacement and increase pump pressure. On the other hand, when the adjustment screw 90 is rotated in a direction to shift the pilot valve sleeve 83 toward the right, then the required axial travel of the pilot valve spool to uncover ports 126 is reduced. As a consequence, the back pressure of outlet passageway 23 and passageway A necessary to shift spool 84 is reduced, so as to decrease the maximum output pressure of the pump and to increase its displacement.

As noted earlier, the actuating arm 60 of swash plate 28 automatically assumes the position of FIGURES 5 and 6 when the pump is shut down or when there is a minimum load on the hydraulic motor 54 (FIGURE 2), thus providing maximum pump displacement at low pressure. As the load on the hydraulic motor 54 increases, back pressure in output passageway 23 and pressure passageway A correspondingly increases. As shown by the arrow C in FIGURE 7, the fluid pressure from passageway A and port 122 acts upon the full diameter of land 117 by operation of the relief areas 123 (FIGURE 8) thus tending to shift the pilot valve spool 84 toward the left and causing land 117 to uncover port 126, as noted above. As fluid pressure is transmitted to the stroke control piston 86, the piston and swash plate control arm 60 are shifted toward the right, as previously indicated by the arrows in FIGURES 5 and 6. This motion tends to compress the coil spring 95 (by operation of spring guide 104 and arm biasing spring 103), thereby tending to shift the pilot valve spool 84 back toward the right in a direction to cover port 126. Thus, the coaction between the pilot valve spool 84 and stroke control piston 86 provides, in effect, a servo-motion leading to a state of balance between the stroke control piston 86 and pilot valve spool 84. Otherwise expressed, motion of the pilot valve spool toward the left (FIGURES 10 and 12) causes

a corresponding motion of the stroke control piston toward the right, thus tending to shift the valve spool back toward the right through operation of the valve spring 95.

It will be understood that the right hand motion (FIGURE 13) of the actuating arm 60 reduces the piston stroke and displacement of the pump and increases output pressure so as to compensate for the increased load on the hydraulic motor 54. As the torque control apparatus reacts to the load resistance, as noted above, a state of equilibrium is established between the pump and motor, as determined by the setting of the adjustment screw 90. In other words, the spool is balanced by the force of the compressed spring 95 acting in one direction and by the force of hydraulic pressure acting upon the spool in the opposite direction.

When the pilot valve spool 84 is thus balanced, the land 117 blocks off the ports 126 to prevent further admission of fluid to the stroke control piston 86. However, a decrease in back pressure will cause the spring 95 to shift spool 84 toward the right. This motion permits fluid pressure to pass from stroke control piston 86 and passageway B, through ports 126 and through the relief areas 124 of land 116, and to drain by way of cross bore 62 and aperture 125 (drain 127—FIGURE 2). The drainage flow is indicated by the arrow D in FIGURES 2 and 13. As a consequence, the pilot valve spool floats substantially in a state of balance and responds immediately to any change in the load resistance of hydraulic motor 54 (and back pressure) to control the stroke and output pressure of the pump.

As the load on the hydraulic motor 54 approaches the maximum for which the torque control apparatus is adjusted, the flange 101 of the hat-shaped sleeve 100 comes into contact with the cap 102 (FIGURE 6). Under this condition, the actuating arm 60 of the swash plate is in the position of intermediate stroke and maximum pressure. As noted earlier, the arm biasing spring 103 is formed of spring wire having a diameter several times greater than the valve control spring 95. Under normal operation, with the hat-shaped sleeve 100 floating, the motion imparted by the stroke control piston to arm 60 produces a deflection of both the heavy spring 103, and the control spring 95, thus raising the bias load of spool 84, and the pump discharge pressure. However, when the valve control sleeve 100 reaches its limit of motion, then further motion of the actuating arm 60 results in further compression of the spring 103 only. Discharge pressure of the pump then remains constant at minimum stroke.

As noted earlier, the pump 1 is also adapted to be used as a hydraulic motor. In the present example, hydraulic pressure may be transmitted to the motor by way of the intake passageway 22 so as to act upon the pistons 17 of cylinder barrel 9, thereby imparting rotary motion to the barrel by the reaction of the pistons against the swash plate 28. The hydraulic fluid in this case is discharged from the motor by way of the outlet passageway 23. The shaft 15 is thus rotated by the cylinder barrel 9 to drive the element which is connected to the shaft 15.

Having described my invention I claim:

1. A torque control apparatus for a hydraulic power unit, said power unit having a pressure passageway and having a series of displacement pistons, said power unit having a movable element adapted to be shifted in a flat plane in response to back pressure in the pressure passageway to control the stroke of said pistons and the displacement of the power unit, said control apparatus comprising:

an axially shiftable stroke control piston connected to the movable element of the power unit and adapted to shift the same for increasing or decreasing the stroke of said pistons and thereby the displacement of the power unit;

a pilot valve including an axially shiftable valve spool disposed in axial alignment with said stroke con-

trol piston and engaging said movable element in opposition to said stroke control piston;

- a compressible spring element interposed between the axially shiftable valve spool and movable element, adapting the valve spool to act upon the movable element through said spring element;
- said power unit including means providing hydraulic communication between said pressure passageway and said shiftable valve spool, whereby the valve spool is shifted toward said movable element in response to back pressure in said pressure passageway to urge the movable element through the compressible spring element toward said stroke control piston;
- said axially shiftable valve spool adapted to transmit hydraulic pressure in response to said movement of the valve spool to said stroke control piston in a direction to force the same toward the movable element opposite to the force exerted by said valve spool and spring element, whereby the axially shiftable valve spool and axially shiftable stroke control piston bias one another against said movable element to provide a servo action and thereby a balanced relationship to hold the movable element at a position providing a given piston stroke in response to a give back pressure in said pressure passageway.

2. A torque control apparatus for a hydraulic power unit, said power unit having a pressure passageway and having a series of displacement pistons, said power unit having a casing including a cross bore and having a movable element adapted to be shifted in a flat plane in response to back pressure in the pressure passageway to control the stroke of said pistons and thereby the displacement of the power unit, said torque control apparatus comprising:

- an axially shiftable stroke control piston slidably confined in one end portion of said cross bore, said piston engaging the movable element of the power unit and adapted to shift the same for controlling the stroke of the pistons of the power unit;
- a pilot valve residing in an opposite end portion of said cross bore in axial alignment with said stroke control piston;
- an axially shiftable pilot valve spool disposed in said pilot valve in axial alignment with said stroke control piston and engaging said movable element on the side opposite said stroke control piston;
- a compressible spring element interposed between the axially shiftable valve spool and movable element, adapting the valve spool to act upon the movable element to the said spring element;

said power unit including means providing hydraulic communication between said pressure passageway and said shiftable valve spool, whereby the valve spool is shifted toward said movable element in response to back pressure in said pressure passageway;

said axially shiftable valve spool adapted to transmit hydraulic pressure to said stroke control piston to force the piston toward the movable element in a direction opposing the shiftable valve spool, whereby the axially shiftable valve spool and axially shiftable stroke control piston bias one another against said movable element and thereby provide a servo action, thereby to hold the same at a given stroke control position in response to a given back pressure in said pressure passageway.

3. A torque control apparatus for a hydraulic power unit, said power unit having a pressure passageway and having a series of displacement pistons, said power unit having a casing including a cross bore and having a movable element adapted to be shifted in a flat plane through an arcuate path in response to back pressure in the pressure passageway to control the stroke of said pistons and

thereby the displacement of the power unit, said torque control unit comprising:

an axially shiftable stroke control piston slidably confined in one end portion of said cross bore, said piston engaging the movable element of the power unit and adapted to shift the same axially for controlling the stroke of the pistons of the power unit;

a pilot valve sleeve residing in an opposite end portion of said cross bore in axial alignment with said stroke control piston, said pilot valve sleeve including ports communicating with said pressure passageway and with said stroke control piston;

an axially shiftable pilot valve spool confined in said pilot valve sleeve generally in alignment with said stroke control piston and engaging said movable element on the side opposite said stroke control piston; means for adjusting said pilot valve sleeve axially relative to the valve spool;

said power unit including means providing hydraulic communication between said pressure passageway and the ports of said pilot valve sleeve;

said ports adapting the valve spool to be shifted toward said movable element in response to back pressure in said pressure passageway;

a land on said axially shiftable valve spool, said land being acted upon by hydraulic pressure from the ports of the valve sleeve in a direction to shift the valve spool toward the movable element, said land, upon being shifted toward the movable element, adapted to transmit hydraulic back pressure from the ports of the valve sleeve to the stroke control piston to urge the piston in a direction opposing the shiftable valve spool, whereby the axially shiftable valve spool and axially shiftable stroke control piston bias one another axially against said movable element and providing a servo action, thereby to hold the same at a given stroke control position in response to a given back pressure in said pressure passageway as determined by the adjustment of the pilot valve sleeve.

4. A torque control apparatus in accordance with claim 3 in which there is provided an outer sleeve confined in a fixed position in the cross bore generally in alignment with the stroke control piston, said outer sleeve having ports in communication with the pressure passageway of the power unit and with the stroke control piston, said pilot valve sleeve being mounted within said outer sleeve for axial adjustment relative to the outer sleeve, with the ports of the outer sleeve and valve sleeve maintaining communication with said power passageway and with said stroke control piston.

5. A torque control apparatus for a hydraulic power unit, said power unit having a pressure passageway and having a series of displacement pistons, said power unit having a casing including a cross bore and having a movable element adapted to be shifted in a flat plane through an arcuate path to control the stroke of said pistons and thereby the displacement of the power unit, said torque control unit comprising:

an axially shiftable stroke control piston slidably confined in one end portion of said cross bore, said piston engaging the movable element of the power unit and adapted to shift the same for controlling the stroke of the pistons of the power unit;

a pilot valve residing in an opposite end portion of said cross bore in axial alignment with said stroke control piston;

an axially shiftable pilot valve spool disposed in said pilot valve generally in axial alignment with said stroke control piston and engaging said movable element on the side opposite said stroke control piston;

a first compressible spring element interposed between the axially shiftable valve spool and movable element, adapting the valve spool to act upon the mov-

able valve element through said compressible spring element,
 a second compressible spring element concentric with the first spring element and having an end seated against the movable element of the power unit;
 means connecting the opposite end of said second spring element to the first spring element, whereby the motion of the movable element of the power unit is transmitted through said first and second spring elements to the pilot valve spool;
 said power unit including means providing hydraulic communication between said pressure passageway and said shiftable valve spool, whereby the valve spool is shifted through the first spring element toward said movable element in response to back pressure in said pressure passageway;
 said axially shiftable valve spool adapted to transmit hydraulic pressure to said stroke control piston to force the same toward the movable element in a direction opposite to the force exerted by said valve spool, whereby the axially shiftable valve spool and axially shiftable stroke control piston bias one another against said movable element and provide a servo action and thereby a balanced relationship to hold the movable element at a given position to provide a given piston stroke in response to a given back pressure in said pressure passageway.

6. A torque control apparatus in accordance with claim 5 in which there is provided a sleeve element seated against the end portion of said first compressible spring element opposite the pilot valve spool and slidably confined in the cross bore, said sleeve element having an exterior flange formed thereon, said sleeve element telescopically interfitting said second compressible spring element, the end of the second spring element opposite the movable elements of the movable element are transmitted through said second compressible spring element to the said sleeve and through said first spring element to the pilot valve spool.

7. A torque control apparatus for variable displacement hydraulic pump, said pump having a casing including an intake and an output passageway, said casing including a cross bore and having a pressure passageway communicating with a first end portion of said cross bore, said casing having a control passageway connecting said first end portion of the cross bore to a second end portion thereof, said pump casing including a plurality of pistons, a tiltable swash plate adapted to control the stroke of said pistons, said swash plate including an actuating arm for controlling the tiltable motion of the swash plate, said torque control apparatus comprising:

- a stroke control piston slidably confined in the second end portion of said cross bore, said piston engaging the actuating arm of the swash plate and adapted to shift the swash plate for controlling the stroke of the pump pistons;
- a pilot valve including an axially shiftable valve spool disposed in the first end portion of the cross bore generally in alignment with said stroke control piston;
- an adjustment screw threaded through an end portion of said casing;
- a ball and socket joint connecting the adjustment screw to said axially shiftable valve spool and adapted to shift the same axially in forward or reverse directions in response to rotation of said adjustment screw;
- said pressure passageway adapted to transmit hydraulic fluid pressure to the first end portion of the cross bore adjacent the end of said valve spool to force the valve spool toward said actuating arm;
- said shiftable valve spool adapted to transmit hydraulic pressure to said control passageway in response to motion of the valve spool toward said actuating arm;
- said control passageway communicating with the second end portion of the cross bore adjacent the end of said stroke control piston and adapted to force the stroke

control piston toward the actuating arm in a direction opposing the shiftable valve spool, whereby the valve spool and stroke control piston bias one another axially against said actuating arm to hold the swash plate at a given stroke control position in response to a given back pressure in said output passageway.

8. A torque control apparatus for a variable displacement hydraulic pump, said pump having a casing including an intake and an output passageway, the casing having a cross bore and having a pressure passageway connecting the output passageway with a first end portion of the cross bore, the casing having a control passageway connecting said first end portion of the cross bore to a second end portion of the cross bore, said pump having a movable element adapted to be shifted in a flat plane through an arcuate path to control the hydraulic displacement of the pump, said torque control apparatus comprising:

- a shiftable displacement control piston mounted in the second end portion of said cross bore, said piston engaging the movable element of the pump and adapted to shift the same for controlling the displacement of the pump;

- a pilot valve sleeve residing in the first end portion of said cross bore generally in alignment with said displacement control piston, said pilot valve sleeve including ports communicating with said pressure passageway and with said control passageway of the pump casing;

- an axially shiftable pilot valve spool confined in said pilot valve sleeve generally in alignment with said displacement control piston and engaging the movable element of the pump on the side opposite said displacement control piston;

- means for adjusting said pilot valve sleeve axially relative to the valve sleeve, said pressure passageway of the pump casing transmitting hydraulic back pressure from said output passageway through the ports of said pilot valve to said shiftable valve spool;

- said adjustment means comprising an adjustment screw threaded through an end portion of said casing and having a socket formed in the inner end portion thereof;

- said pilot valve spool having a spherical head interfitting said socket and adapted to be shifted axially in forward or reverse directions in response to rotation of the adjustment screw;

- a land on said valve spool acted upon by said back pressure whereby the valve spool is urged hydraulically toward the movable element of the pump in response to the back pressure of the output passageway;

- said land of the axially shiftable valve spool adapted to transmit hydraulic pressure from the ports of the valve sleeve through the control passageway of the casing to the second end portion of the cross bore, whereby the back pressure acts upon the displacement control piston to force the same toward said movable element of the pump, whereby the axially shiftable valve spool and axially shiftable stroke control piston bias one another against opposite sides of the movable element to hold the same at a given displacement position in response to a given back pressure in said output passageway and pressure passageway as determined by the adjustment of the pilot valve sleeve.

9. A torque control apparatus in accordance with claim 8 in which said pressure passageway and ports of the valve sleeve communicate with one side of said land to urge the valve spool toward the movable element and in which the end of the valve spool adjacent said movable element is provided with a second land having relief areas, whereby a drop in the hydraulic back pressure transmitted by the pressure passageway adapts the displacement control piston and movable element of the pump to shift in a direction opposing the shiftable valve spool, said second land adapting the hydraulic pressure to be exhausted from the

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axially shiftable displacement control piston through the control passageway and the relief areas of the second land and from the valve sleeve as determined by the adjustment of the pilot valve sleeve.

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