May 16, 1961 F. T. SMITH 2,984,222

CONSTANT WORK OUTPUT ROTARY HYDRAULIC DEVICE

Filed May 8, 1957 4 Sheets-Sheet 1

Fig-1

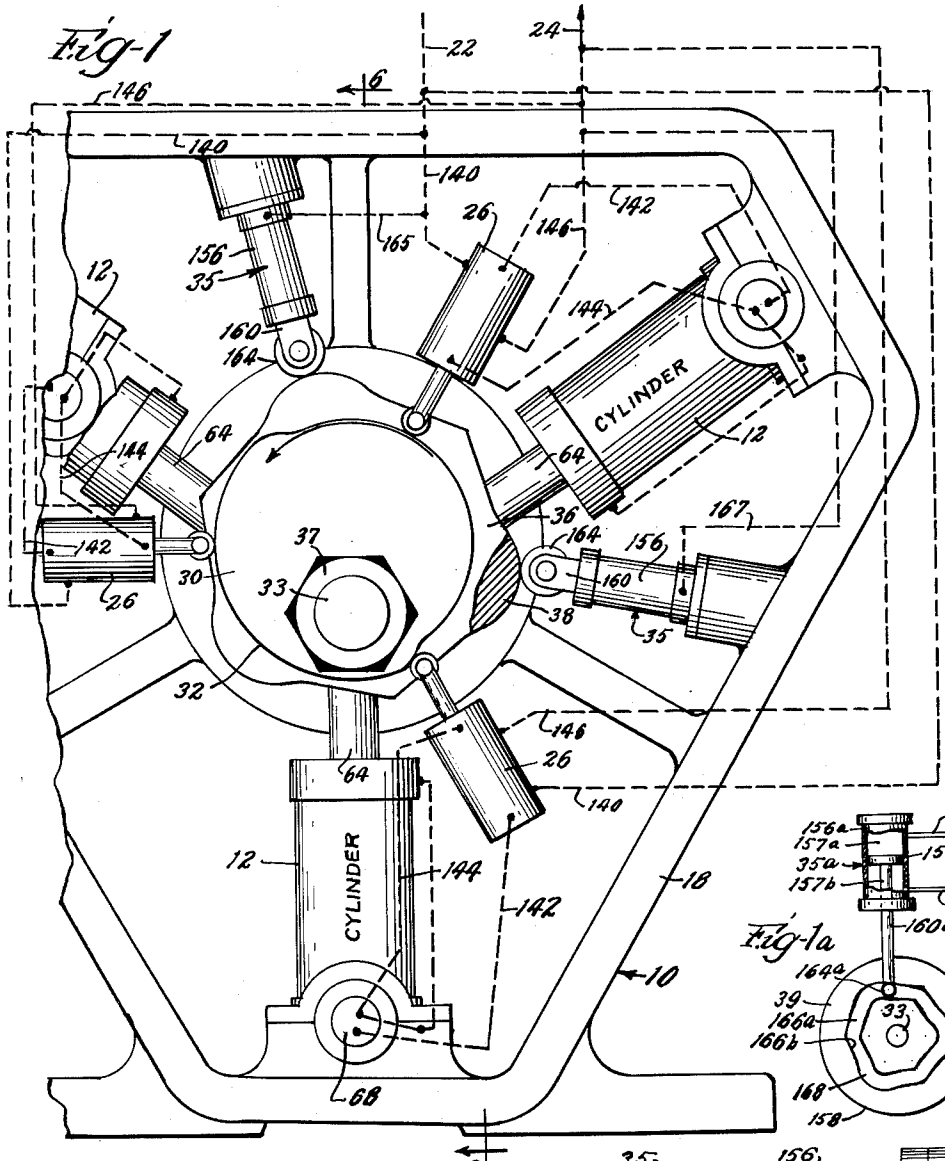


Fig-1a

Fig-2

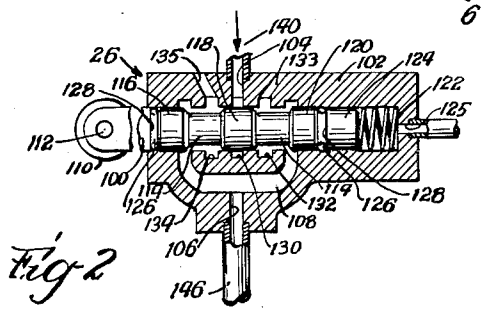
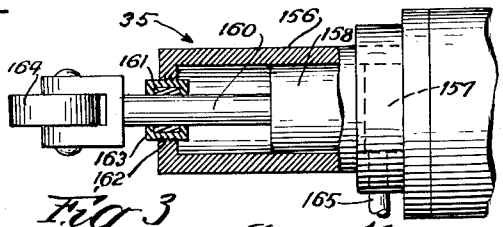


Fig-3



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CONSTANT WORK OUTPUT ROTARY HYDRAULIC DEVICE

Filed May 8, 1957 4 Sheets-Sheet 2

Fig-4

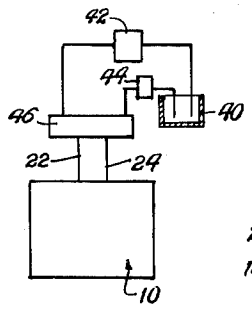


Fig-8

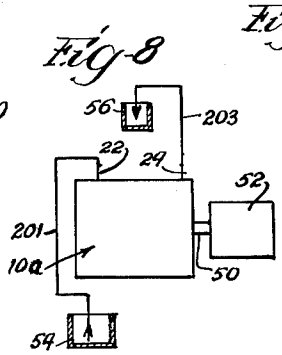


Fig-7

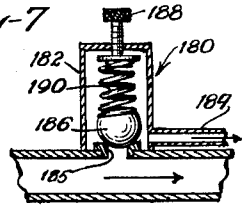


Fig-6

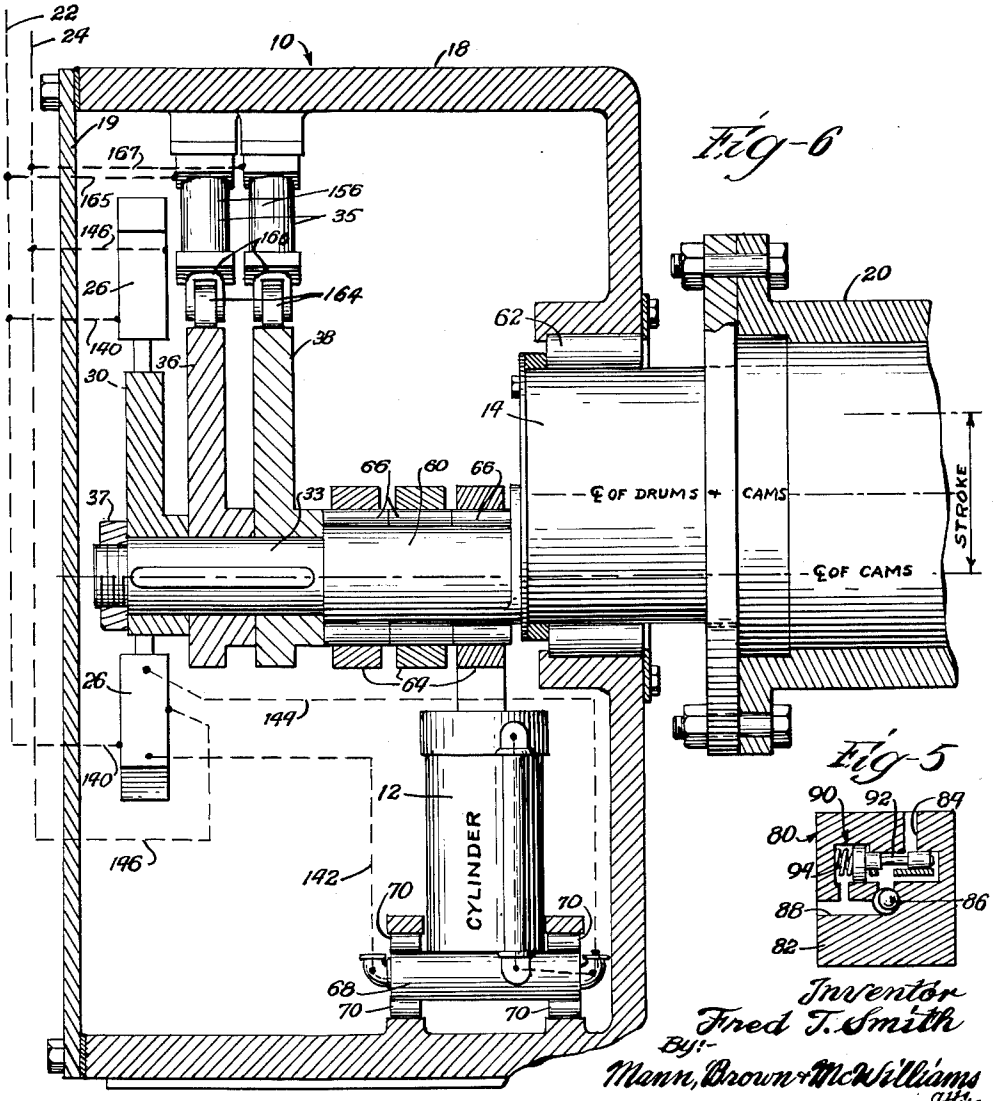


Fig-5

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CONSTANT WORK OUTPUT ROTARY HYDRAULIC DEVICE

Filed May 8, 1957 4 Sheets-Sheet 3

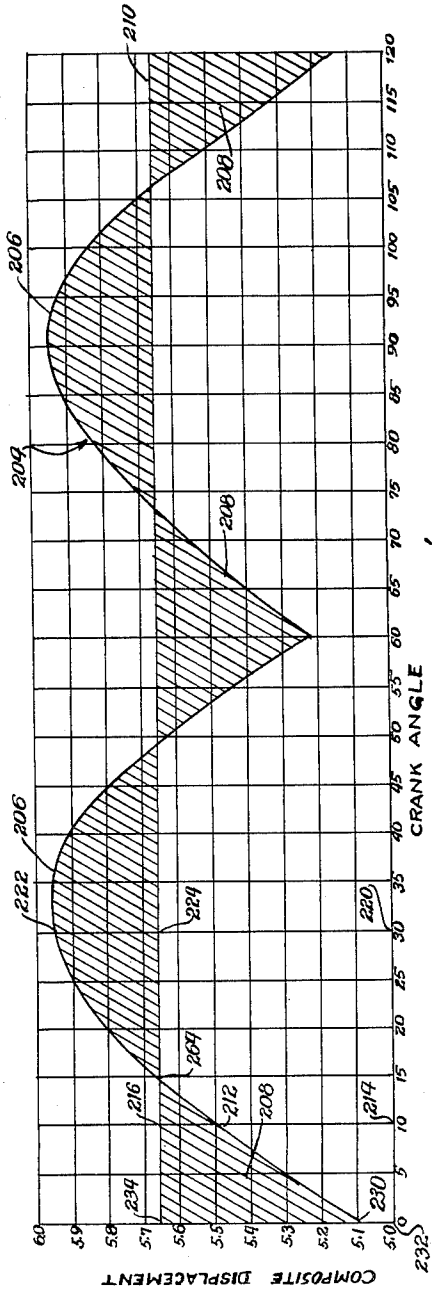


Fig-9

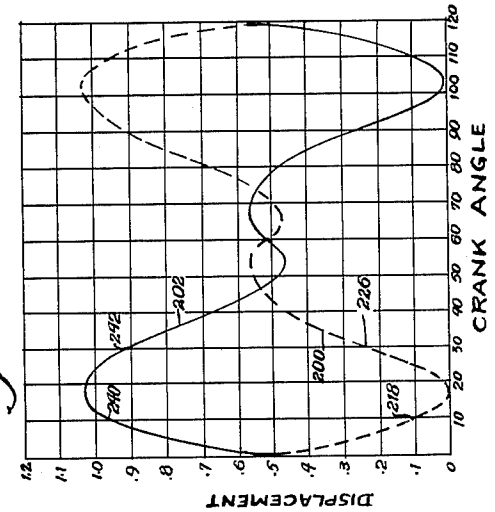


Fig-10

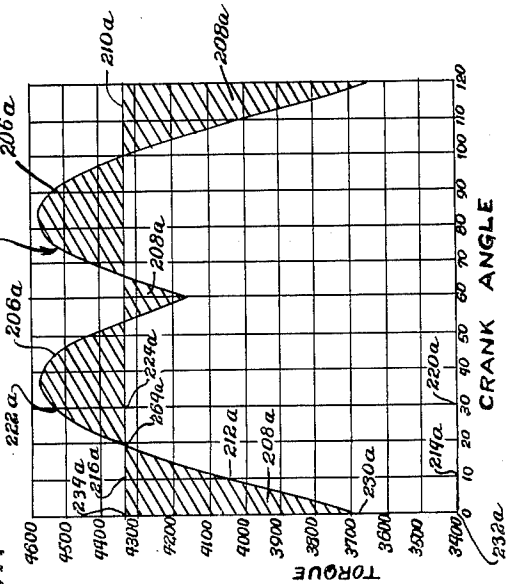


Fig-9A

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CONSTANT WORK OUTPUT ROTARY HYDRAULIC DEVICE

Filed May 8, 1957 4 Sheets-Sheet 4

Fig-11

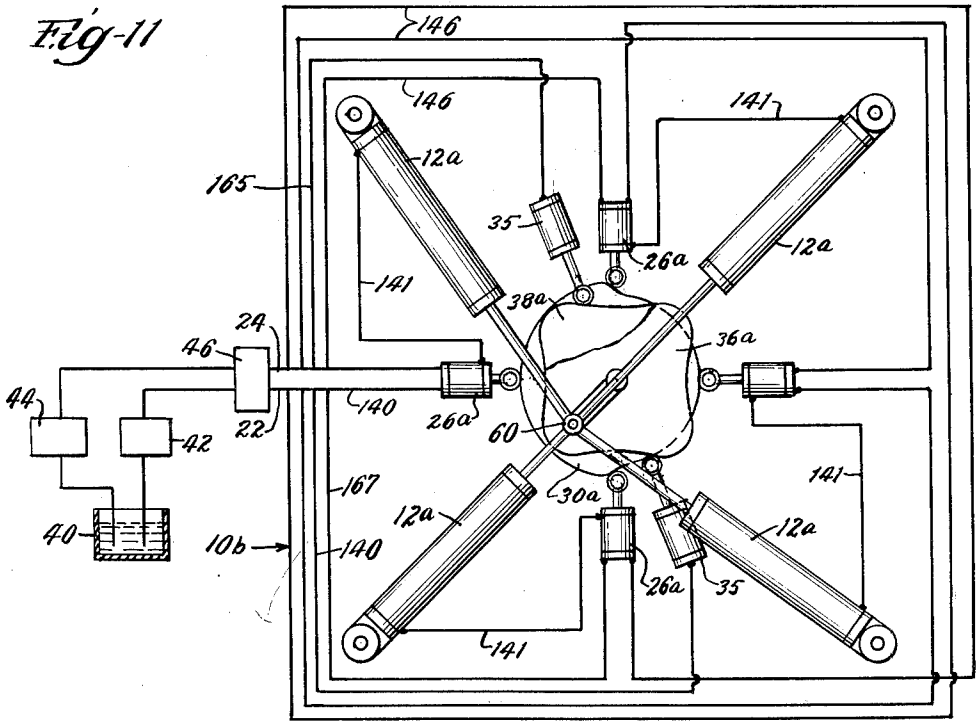


Fig-12

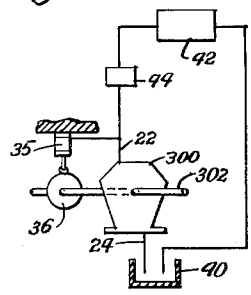


Fig-13

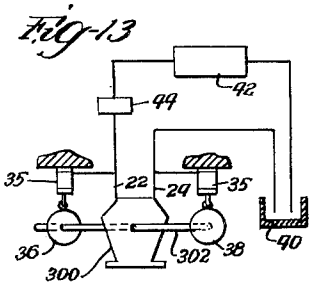


Fig-14

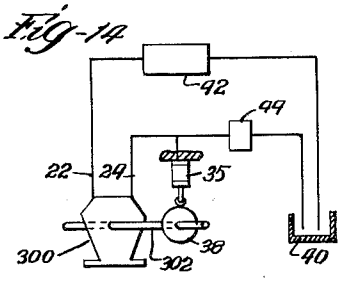


Fig-15

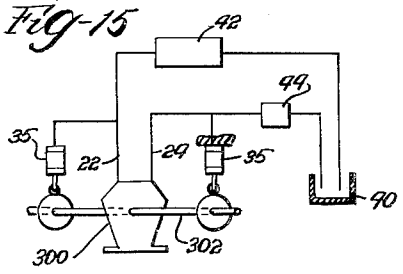
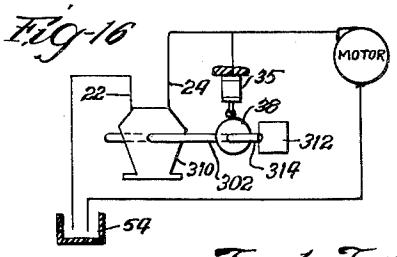


Fig-16



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United States Patent Office 2,984,222 Patented May 16, 1961

2,984,222

CONSTANT WORK OUTPUT ROTARY HYDRAULIC DEVICE

Fred T. Smith, Park Forest, Ill., assignor to Whiting Corporation, a corporation of Illinois

Filed May 8, 1957, Ser. No. 657,862

18 Claims. (Cl. 121—118)

My invention relates to a constant work output rotary hydraulic device operable either as a motor or as a pump, and more particularly, to a positive displacement hydraulic device of the type that includes a crank shaft or its equivalent actuated by or actuating pistons or equivalent structures, which provides a predetermined and uniform or constant, rather than a pulsating, work output for and through each cycle of operation of the device, or revolution of the crank shaft.

One of the prominent characteristics of conventional hydraulic motors and pumps is that they produce a pulsating output. Thus, hydraulic motors conventionally supply a pulsating torque output, while hydraulic pumps produce a pulsating liquid discharge.

Hydraulic devices of this type conventionally comprise a sliding block type linkage including a crank shaft rotated by, or rotated to actuate, a plurality of pistons reciprocating in cylinders that may be arranged in line or radially about the crank shaft. The pistons are connected to the shaft by piston rods and crank arms. If the hydraulic liquid is the driving agency, it acts against the pistons which in turn apply torque to the shaft to rotate same to provide power. If the pistons are the driving agency, they are reciprocated by rotating the shaft, which actuates them through their respective piston rods and crank arms; the pistons then perform work on the liquid and act as a pump..

In either case, the conversion between rotary and reciprocating movement in sliding block linkages means that each piston will necessarily come to a complete stop at two particular positions of the crank shaft before the direction of its motion is reversed, and its rectilinear speed is at a maximum midway between these two crank shaft positions. Since the rate of piston movement along its cylinder is equivalent to the rate of displacement of the cylinder in which it is mounted, the displacement, and consequently, the displacement or liquid acceptance capacity of each piston and cylinder has the well-known curve (for instance, liquid flow or displacement plotted against degree of crank angle revolution) in which the displacement is at a minimum at the ends of the piston stroke and is at a maximum adjacent the middle of the stroke. This means that over the cycle of operation of the device, whether it operates as a motor or as a pump, liquid discharge is made pulsating in nature by the pistons and cylinders forming a part of the device. Moreover, the lever arms between the individual pistons and the shaft vary throughout the cycle of operation, and it is common knowledge that the summation of these lever arms over the cycle of operation of devices of this type is in effect a fluctuating mechanical advantage, which means that the force transmitted through hydraulic devices of this type will provide an output that necessarily fluctuates over the cycle of operation of the device. In motors, the latter has the effect of providing a pulsating torque output at the shaft regardless of how carefully hydraulic liquid flow is controlled. Consequently, hydraulic devices of this type heretofore have been limited

in application to situations where the adverse effect of the pulsating output is negligible.

A principal object of my invention is to provide a hydraulic device of the type described above in which said varying mechanical advantage is compensated for.

A further principal object of my invention is to provide a hydraulic device that operates through a constant mechanical advantage throughout the operating cycle of the device regardless of whether it serves as a motor or as a pump and regardless of whether it operates in a forward or reverse direction.

Still a further object of the invention is to provide a constant mechanical advantage rotary hydraulic motor, and hydraulic circuits for operating same, to provide a uniform, rather than a pulsating, torque output at the shaft.

A further important object of the invention is to provide a constant mechanical advantage rotary hydraulic pump.

Another object of the invention is to provide a positive displacement hydraulic device in which the total hydraulic liquid receiving capacity or displacement of the unit is made predetermined and uniform for each cycle, and portion of a cycle, of operation of the device.

Yet another object of the invention is to provide a hydraulic drive unit capable of providing a constant or uniform angular torque output over each cycle, and portion of a cycle, of operation of the device.

Another and further object of my invention is to provide a motor or drive unit having stepless speed control and capable of producing a very high torque at low speeds.

Still another object of my invention is to provide a motor or drive unit that will produce both constant torque and speed, and also provide stepless speed control without altering the torque provided.

Yet a further object of the invention is to provide a method of completely eliminating output pulsations in positive displacement hydraulic pumps and motors.

Still another object of the invention is to provide a hydraulic device having the foregoing characteristics which is composed of few and simple parts, which is inexpensive of manufacture, and which is adapted for use in a wide variety of applications.

In accordance with the principles of my invention, I provide a rotary hydraulic device including a plurality of cylinders and pistons and one or more pulsation compensators in which, if the device is operated in accordance with a basic rule or expression I have developed, the output pulsations heretofore characteristic of positive displacement rotary hydraulic apparatus are eliminated, whether the hydraulic device operates as a motor or as a pump.

Other objects, uses, and advantages will become apparent from a consideration of the following detailed description and the appended drawings.

In the drawings:

Figure 1 is a front elevational view, largely diagrammatic in nature, illustrating the internal arrangement of one embodiment of my invention arranged to operate as a motor, with the front housing plate omitted for clarity of illustration and showing diagrammatically a portion of the hydraulic circuiting;

Figure 1*a* is a diagrammatic fragmental front elevational view illustrating a modified form of the invention;

Figure 2 is a diagrammatic cross-sectional view of one of the three separate hydraulic liquid flow orienting valves employed in the device shown in Figure 1 for controlling the operation of the respective hydraulic cylinders;

Figure 3 is a diagrammatic detailed view, partially in

section, of one of the output compensator devices employed in the hydraulic device of Figure 1;

Figure 4 diagrammatically illustrates a hydraulic circuit for operating the devices of Figure 1 or 1*a* as a motor;

Figure 5 is a diagrammatic cross-sectional view of a hydraulic liquid volume control valve employed in connection with the devices shown in Figures 1 or 1*a* for operating same as a motor where constant volume liquid flow conditions are required;

Figure 6 is a diagrammatic cross sectional view approximately along the line **6—6** of Figure 1, with some parts displaced and other parts shown in elevation to better illustrate the structural features of the form of the invention of Figures 1–5, and with the hydraulic circuiting partially illustrated;

Figure 7 is a diagrammatic cross-sectional view of a hydraulic liquid pressure control valve that may be employed in the hydraulic circuit where constant pressure conditions are required;

Figure 8 diagrammatically illustrates a separate hydraulic circuit for operating a device substantially identical to those of Figures 1 or 1*a* as a pump;

Figure 9 is a hydraulic displacement or liquid receiving capacity graph employed in developing the compensator controlling cams forming a part of the motor shown in Figure 1, showing only the upper portion thereof for 120 degrees of crank shaft rotation;

Figure 9*a* is a torque graph which may be employed in developing the compensator controling cams of the embodiment of Figure 1, showing only the upper portion thereof for 120 degrees of crank shaft rotation;

Figure 10 is a graph illustrating portions of two cam follower movements, greatly magnified, of the pulsation compensators employed in the motor of Figure 1, the dashed line illustrating cam follower movement required when a compensator is on the pressure side of the hydraulic system and the unbroken line illustrating cam follower movement required when a compensator is on the exhaust side of the hydraulic system;

Figure 11 is a diagrammatic view illustrating the principal components of a modified form of motor in which the principles of my invention are incorporated; and

Figures 12, 13, 14, 15, and 16 are diagrammatic sketches illustrating the basic arrangements necessary to achieve performance in accordance with the fundamental rule or expression I have developed.

GENERAL DESCRIPTION

Reference numeral **10** of Figures 1, 4, and 6 generally indicates a specific embodiment of a hydraulic motor in which the principles of my invention have been embodied. As shown more particularly in Figures 1 and 6, the motor **10** comprises three double acting cylinders **12** radially arranged about a crank shaft **14** (see Figure 6) which in turn is journalled as at **16** in an appropriate motor housing **18** and rotates in the embodiment illustrated in these figures a cable drum **20** of any appropriate design.

The hydraulic motor **10** is operated to rotate the crank shaft **14** in the direction of the arrow of Figure 1 by hydraulic liquid under pressure entering the motor through intake or pressure line **22** and leaving the motor through exhaust or return line **24**. The pressure line **22** internally of the housing **18** communicates with a hydraulic liquid flow orientating valve **26** for each cylinder **12** which orients the liquid flow to the head and crank or rod ends of the respective cylinders in a conventional manner. Hydraulic liquid flow from the cylinders is returned through the valves **26** to the return line **24**. The valves **26** are actuated by a cam **30** having a cam surface **32**. The cam **30** is keyed to stub shaft **33** of crank shaft **14** in any suitable manner.

Application of hydraulic liquid under pressure to pressure line **22** results in displacement of the pistons within cylinders **12**. The force applied to each piston by the hydraulic liquid is transmitted to the crank shaft through the well known sliding block linkage between the cylinders **12** and the crank shaft. Since the mechanical advantage between the pistons and the crank shaft varies over the cycle of operation of the motor, the angular torque output of the shaft will tend to fluctuate or pulsate over its cycle of operation.

The angular torque output T_0 of the crank shaft **14** at any instant is equivalent to the expression T_1-T_2, wherein T_1 is the torque applied to the crank shaft by the pressure side of the system while T_2 is the countertorque applied to the crank shaft by the back pressure in the exhaust side of the system. Since torque is equal to force times a lever arm and mechanical advantage is functionally a "lever arm," T_1-T_2 equals $F_1A_1-F_2A_2$, wherein F_1 is the total intake side force provided by the driving hydraulic liquid acting on the pistons of cylinders **12**, A_1 is the summation of the mechanical advantages through which the pistons of cylinders **12** act through at that instant in applying torque to the crank shaft **14**, F_2 is the total counterforce opposing the movement of the pistons (provided by the back pressure on the discharge side of the hydraulic system) and A_2 is the summation of the mechanical advantages through which F_2 acts in applying torque to the crank shaft.

I have found that T_0 may be made constant over the cycle of operation by holding at least one of the factors A_1 or A_2 constant and controlling the values of F_1 and F_2 over the cycle of operation of the motor, thus providing the following governing expression

$$T_0=C=F_1A_1-F_2A_2$$

Any positive displacement hydraulic motor may be provided with a constant cyclical torque output by the application of governing expression or rule thereto. Moreover, the discharge pulsations of hydraulic pumps may be eliminated by following the same expression or rule.

In practicing the principles of my invention, I compensate for the varying mechanical advantage that the intake and discharge hydraulic pressures act through. In the illustrated embodiments this is done by providing separate compensators responsive to the intake and discharge pressures respectively, that add and subtract torque increments to the shaft in proportion to the variation in the composite displacement of the motor or pump cylinders. The compensators are in free communication with the hydraulic liquid passing through the hydraulic cylinders. In the motor of Figure 1, these compensators are generally indicated at **35** and take the form of single acting cylinders and pistons (see Figure 3) that are actuated respectively by separate cams **36** and **38** and are in free and open communication with pressure and exhaust lines **22** and **24**, respectively. Compensators **35** of motor **10** are provided to make both A_1 and A_2 of my governing expression or rule constant. Cams **36** and **38** are also keyed to stub shaft **33** of the crank shaft **14** in any suitable manner, the three cams being retained in position in the illustrated embodiment by nut **37**.

Alternately, in the case of three double acting cylinders, or six single acting cylinders, a single double acting compensator **35***a* (see Figure 1*a*) may be substituted for the pair of compensators **35**, compensator **35***a* being actuated by a single cam plate **39** keyed to stub shaft **33** in place of cams **36** and **38**.

The application of my formula or governing expression to hydraulic motors requires that the following conditions be observed:

(A) A compensator **35** must be employed on either the intake or exhaust side of the hydraulic system.

(B) If a compensator **35** is employed on the intake or pressure side of the hydraulic system, the compensator must be in free communication with the intake flow to all the motor cylinders. If a compensator **35** is employed on

the exhaust or discharge side of the hydraulic system, the compensator must be in free communication with the exhaust flow from all of the motor cylinders.

(C) The pressure of the hydraulic liquid on the compensators employed should be other than atmospheric, and preferably is greater than atmospheric.

(D) Where hydraulic liquid pressure or volume control valve are required on either the intake or discharge sides of the hydraulic system to control liquid flow in accordance with my governing expression or formula, the compensator on that particular side of the system must be in communication with the hydraulic liquid between the motor cylinders and the liquid control valve.

(E) Where a compensator **35** is employed only on the exhaust or discharge side of the hydraulic system, the liquid control valve employed must be a constant volume flow control valve, and it must be on the discharge side of the system.

(F) If compensators **35** are employed on both sides of the system, and a liquid control valve is employed on the discharge side of the system, it may be either a constant volume flow control valve or a constant pressure control valve if the motor is provided with a constant pressure feed. If the motor is not provided with a constant pressure feed, the discharge side control valve must be a constant volume flow control valve.

(G) Where a compensator **35** is employed only on the intake or pressure side of the system and the back pressure on the discharge side is zero, and when compensators are employed on both sides of the system, if employed, the intake or pressure side liquid control valve may be either a constant volume flow control valve or a constant pressure control valve.

These conditions are illustrated by the diagrammatically illustrated circuit arrangements of Figures 4 and 12–15.

Thus, the motor **10**, when incorporated in a hydraulic circuit in accordance with my governing expression, such as that illustrated in Figure 4, will provide a constant or uniform angular torque output over its cycle of operation. That is, the torque available over the cycle of operation of the motor will be uniform and non-pulsating. Motor **10** may be run in either a forward (in the direction of the arrow of Figure 1) or reverse direction with the same results. When the motor is run in the direction of the arrow of Figure 1, hydraulic liquid forming the pressure liquid is drawn from a source **40** (see Figure 4) by an appropriate hydraulic liquid supply pump **42** and forced through a four-way control valve **46** of any appropriate design and into the intake or pressure line **22**. Hydraulic liquid passes from the pressure line **22** into the conduits leading to the various valves **26**, each of which is of the type diagrammatically illustrated in Figure 2, from which the liquid is distributed to the head and crank or rod ends of the respective cylinders as dictated by the cam **30**, and thence back through the valves **26** to return line **24**, through four-way control valve **46** and thence through a hydraulic liquid control valve **44** back to the source of hydraulic liquid **40**. If the pump **42** does not provide a constant pressure feed, valve **44** must be a constant volume flow control valve (for instance, of the type shown in Figure 5), and the discharge side compensator **35** must be in communication with the discharge liquid flow between valve **44** and all of the cylinders **12**.

The compensators **35** of the embodiment of Figure 1 are connected to the pressure and return lines respectively, and in accordance with my invention, when the motor **10** operates in a forward direction, its operation will correspond to the expression $T_0=C=F_1A_1-F_2A_2$, wherein A_1 and A_2 are made constant over the cycle of operation of the device and F_2 is varied by valve **44** to compensate for the variations, if any, in pressure on the intake or pressure side of the system. If pump **42** is a constant pressure pump or if a constant pressure control valve (which may be of the type shown in Figure 7) is interposed upstream of valve **46**, valve **44** may be a constant pressure control valve, since F_1 and F_2 would then also be constant; under such conditions, valve **44** could be eliminated as discharge to tank is normally under constant pressure.

When the motor **10** is operated in reverse direction, as may be done by appropriately manipulating valve **46** to make line **24** the intake or pressure line and line **22** the return or exhaust line, the operation of the mechanism **10** as a motor will provide a constant torque in the opposite direction in accordance with the same governing expression since none of the conditions of operation, other than the direction of operation, are changed.

The compensator **35***a* of the embodiment of Figure 1*a* is essentially a double acting cylinder, one end of which is connected to line **22** and the other end of which is connected to line **24**. The action of compensator **35***a* under the action of its controlling cam **39** is essentially identical to the combined action of compensators **35**.

Briefly, the compensators **35** and **35***a* perform as follows in eliminating the variations in mechanical advantage that the hydraulic liquid acts through in applying torque to the crank shaft.

In accordance with my invention, the compensator **35** or the side of compensator **35***a* that is connected to the intake or pressure side of the system makes constant over the cycle of operation of the motor the liquid acceptance capacity of the pressure side of the system; it does so by withdrawing and adding increments of hydraulic liquid in proportion to the variations in the composite displacement curve (see Figure 9) of the motor cylinders over the cycle of operation (or 360 degrees of crank angle). The compensator in adding and subtracting hydraulic liquid from the pressure side of the system adds and subtracts torque increments to and from the crank shaft through cam **36** to compensate for the differences in lever arm length that the individual pistons act through over the cycle of operation of the motor. The compensator **35** (or the side of compensator **35***a*) that is connected to the return side of the system acts in the same manner with respect to the discharge flow of the motor and back pressure applying countertorque to the crank shaft, and in so doing makes the liquid discharge of the motor uniform and smooth.

The result is that the mechanical advantage through which the intake and exhaust pressures act through on the shaft is made constant over the cycle of operation of the motor, and if hydraulic flow is controlled as specified above, a truly uniform torque output is provided by the crank shaft. That is, the angular or instantaneous torque available at each degree of crank angle is the same, or the torque increments uniformly increase or decrease, depending upon the hydraulic feed to the motor cylinders.

The result is the same whether the motor runs in either forward or reverse direction. Moreover, the device or mechanism **10** will operate as a pump, as would occur on the application of torque to the crank shaft by a load applied to drum **20**, either in forward or reverse direction. The compensator **35** (and the side of compensator **35***a*) that is connected to the exhaust side of the hydraulic system performs its same function on operation as a pump, so liquid discharge is made constant or uniform, rather than pulsating. The other compensator (and the other side of compensator **35***a*) merely idles as it is acting under approximately atmospheric pressure. The device should be acting under a pressure head, however, so that the intake side compensator will engage and follow its cam surface.

Figure 8 is provided purely for purposes of illustrating an appropriate hydraulic arrangement for operating the motor **10** as illustrated as a pump by applying torque to the crank shaft. As a practical matter, the functioning of the mechanism as a motor or as a pump depends solely on whether or not the crank shaft is the driving or driven member of the system.

In the arrangement of Fig. 8, reference numeral **10***a*

generally indicates the hydraulic mechanism of Figure 1 (or 1*a*) driven by motor **52** through shaft **50**; reference numeral **54** indicates a source of hydraulic liquid, and reference numeral **56** indicates a receptacle to which the hydraulic liquid is pumped. The pump **10***a* can be operated in either forward or reverse directions, as dictated by the direction of operation of motor **52**, and the liquid discharged by the pump **10***a* when run in either direction will be uniform or constant, as the compensator on the exhaust side of the system will always control and eliminate the pulsations of the liquid discharge.

Pump operation of the devices **10** or **10***a* conforms to my governing expression $C=F_1A_1-F_2A_2$ if the torque input to the shaft is made constant. F_1 will equal zero or may be disregarded for all practical purposes and $F_2A_2=-C$, F_2 being the force provided by the back or tank pressure of the pump discharge which is ordinarily constant, A_2 being made constant by the discharge side compensator and C being the torque applied to the shaft by the force provided by the back pressure acting on the crank shaft through the pistons of the pump.

Furthermore, any positive displacement hydraulic pump may be provided with a uniform non-pulsating discharge by applying these principles of the invention to it.

Single acting cylinders may be employed as well as double acting cylinders, for either pump or motor operation with the same results if the arrangement is in accordance with the principles just discussed. Such an embodiment is generally indicated by reference numeral **10***b* in Figure 11, wherein single acting cylinders **12***a* act on, or are actuated by, crank arm **60** of a crank shaft **14**. Intake and exhaust conduits **22** and **24** communicate with liquid orientating or distributing valves **26***a* that control the liquid flow to and from the respective cylinders **12***a* in a conventional manner. A single liquid conduit **141** extends between each cylinder **12***a* and its controlling valve **26***a*, these conduits serving both to supply liquid to the respective cylinders and return same to the respective valves **26***a*. Compensators **35** are employed in the same manner described above, although their controlling cams have a somewhat different design, as hereinafter made clear. Device **10***b* may be employed both as a pump and as a motor. If it is operated as a motor, it must be incorporated in a hydraulic circuit conforming to the conditions above specified, which are satisfied by the circuit illustrated in Figure 11 wherein valve **44** is a constant volume flow control valve, and pump **42** and four-way control valve **46** are employed. The device **10***b* may also be substituted for the mechanism **10***a* in the illustration of Figure 8 for operation as a pump. Valves **26***a* are controlled by a cam similar to cam **30**, and may be any conventional valve that performs the required functions.

Specific description of motor

Referring first to the hydraulic motor arrangement of Figure 1, the three double acting cylinders **12** are of conventional design and are mounted 120 degrees apart about the crank shaft **14**. Crank shaft **14** includes crank arm **60**, and for purposes of illustration, crank shaft **14** is shown journalled in housing **18** by roller bearings **62** (see Figure 6). Each double acting cylinder **12** includes a piston of any conventional type (not shown) to which is secured in the usual manner a piston rod **64** that is received over the crank arm **60** of the crank shaft. For purposes of illustration only, roller bearings **66** are shown interposed between the piston rods and the crank arm **22**, though any kind of bearing arrangement between the respective piston rods and crank arm **60** will be satisfactory for purposes of this invention. The cylinders **12** are each fixed to the respective shafts **68**, each of which is journalled at its ends by, for instance, roller bearings **70** in turn secured to the motor housing **18**. This mounting of the cylinders **12** permits the pivotal movement occasioned by the action of the piston rods on the crank arm **60**, or vice versa, as the case may be.

The housing **18** includes an appropriate end plate **19** (see Figure 6), which is omitted in the showing of Figure 1.

The flow orientating valves **26**, as indicated hereinbefore, supply hydraulic liquid alternately to the head and crank ends of the respective cylinders. One valve **26** is provided for each cylinder, and all of the valves cooperate with the cam **30**, which is designed in the usual way to insure their proper operation. A valve **26** is diagrammatically illustrated in Figure 2 and includes a valve plunger **100** slidably mounted in the valve body **102** that is provided with an inlet **104** and an outlet **106** that lead from valve chamber **108**. The plunger **100** rotatably carries a cam roller **110** secured thereto by a pin **112**. The roller **110** engages the cam surface **32** of cam **30**. The plunger **100** is formed with annular recesses **114** and spaced lands **116**, **118**, and **120**. The plunger **100** is urged to the left of Figure 2 by an appropriate spring **122** that bears against the inner end **124** of the member **100**. O-ring seals **126** received in grooves **128** formed in the plunger **100** seal off the valve chamber **108**, and the chamber in which the spring **122** is disposed may be drained through an opening **125**.

Liquid entering a valve **26** from the pressure line enters inlet **104** that communicates with an annular feed groove or recess **130** formed in the valve body **102**. In the position illustrated in Figure 2, pressure is closed to both ends of the cylinder operated by the valve **26**. When the plunger **100** moves to the left of Figure 2, the right hand groove **114** is positioned to bring annular groove or recess **130** into communication with the annular groove or recess **132** that communicates with the head end of the cylinder through passageway **133** and appropriate piping arranged as diagrammatically illustrated in Figure 1. In this position, pressure is applied to the head end of the cylinder, and hydraulic fluid at the crank end of the cylinder is returned through appropriate piping arranged as diagrammatically illustrated in Figure 1 to passageway **135** and thence to annular groove or recess **134**, through the valve chamber **108**, and thence to the outlet **106**. When the plunger **100** moves to the right, the other groove **114** brings into communication the grooves and recesses **130** and **134** which supply fluid under pressure to the crank end of the cylinder served by the valve **26** and then the first groove or recess **114** is positioned to permit return of fluid from the crank end of the cylinder through an appropriate conduit through the passageway **133**, the groove or recess **132**, the valve chamber **108**, and the outlet **106**.

In the position of the valve **26** shown in Figure 2, both the head and crank ends of the cylinder are open to the return line **24**.

The pressure line **22** is connected to the inlet **104** of each valve **26** by an appropriate conduit **140**. The conduits **140** and the pressure line **22** comprise what may be termed pressure conduit means that extends between, for instance, the valve **46**, and the respective valves **26**. Appropriate pressure and return conduits **142** and **144** extend between the respective valves **26** and the cylinders **12**, and each valve **26** is provided with a return conduit **146** that is connected with return line **24**.

In the embodiment of Figure 11, conduits **140** and **146** connect each valve **26***a* with the pressure and return lines **22** and **24**, respectively. The cylinders **12***a*, valves **26***a* and compensators **35** in practice would be arranged in a housing similar to that of Figure 1 and in a similar manner.

The compensators

The compensators **35** each comprise a cylinder **156** fixed to the housing **18** in any suitable manner and in which is reciprocally mounted a single acting piston **158** of any conventional design, the cylinder and piston form-

ing an expandable and contractable hydraulic liquid receiving chamber **157**. A piston rod **160** extends from piston **158** through, for instance, a brass bushing **161** held in the end **162** of the cylinder **156** by an appropriate steel retainer **163**. The piston rod **160** at its end rotatably carries a roller **164** that engages the cam surface of one of the cams **36** or **38**, depending on whether the device **35** is to operate on the intake or discharge side of the motor **10**.

As indicated in Figure 1, the head end of one of the cylinders **156** is placed in communication with the pressure or intake line **22** by an appropriate conduit **165**, while the head end of the other cylinder **156** is placed in communication with the return line **24** by an appropriate conduit **167**. In the illustrated motor, the compensator in communication with line **22** is controlled by cam **36**, while the other compensator is controlled by cam **38**. This is also true in the embodiment of Figure 11, though the cams will have a different design.

The compensator **35***a* comprises a cylinder **156***a* that may be fixed to housing **18** in any suitable manner and in which is mounted a double acting piston **158***a* of any conventional design, the cylinder and piston forming expandable and contractable chambers **157***a* and **157***b*. The piston rod **160***a* of this embodiment may extend through a bushing similar to bushing **161**, and it carries at its end a roller **164***a* that engages spaced cam surfaces **166***a* and **166***b* that define an endless groove **168**. Conduit **165** is connected to the head end of cylinder **156***a*, while conduit **167** is connected to the crank end thereof.

Design of cams controlling compensators for motor

As described above in general terms, it is the operation of the compensators **35** or compensator **35***a* that makes constant over the cycle of operation of the motor the mechanical advantage that the hydraulic liquid acts through in applying torque to the crank shaft, and provides a uniform liquid discharge. The compensator piston displacement and consequently, the cam lift required for the compensators **35** or **35***a* to perform their functions bears a definite relationship to the total piston displacement of the hydraulic motor and thus to the liquid receiving capacity of the hydraulic system, to the work output of the pistons, and to the torque applied to the crank shaft, all of which have impressed upon their values the variations in mechanical advantage between the pistons and the crank shaft over the cycle of operation of the motor. The cams in actuating the compensator pistons provide in effect additional lever arms through which the hydraulic liquid acts on the crank shaft, and when the compensator piston displacement is developed as hereinafter described, the additional lever arms provided will make constant over the cycle of operation the mechanical advantage through which hydraulic liquid acts on the crank shaft.

One way of computing the required compensator piston displacement is best described in connection with the graphs of Figures 9, 9*a* and 10 that were developed in connection with the design of the embodiment of Figure 1. Figure 9 is a diagrammatic illustration of the hydraulic displacement or liquid capacity curve that may be used to compute or develop required compensator piston displacement, while Figure 9*a* is a diagrammatic illustration of the torque curve that also may be used to compute or develop required compensator piston displacement. Figure 10 is a diagrammatic graph in which the compensator piston movement required for the compensator on the intake side of the hydraulic system is illustrated by the curve **200**. The curve **202** illustrates the compensator piston movement required for the compensator that is on the exhaust side of the hydraulic system. Once the curve **200** of Figure 10 is developed, it may be incorporated in the cam surface of cam **36** in any conventional manner which will provide the necessary cam lift required. Curve **202** is incorporated in the cam surface of cam **38** in like manner for the same purpose.

If the embodiment of Figure 1*a* is to be employed, curve **200** may be incorporated in cam surface **166***a*, while curve **202** is incorporated in cam surface **166***b*. Where three double acting cylinders, or six single acting cylinders are employed, curves **200** and **202** will be found to be substantially symmetrical and approximately the reverse of each other both horizontally and vertically, thus making curve **202** what may be termed the substantial converse of curve **200**, and permitting a cam arrangement such as that illustrated in Figure 1*a*. However, this is true only in this specific embodiment of the invention. Curves **200** and **202** of this embodiment of the invention will be exactly symmetrical if the areas of each side of the compensator piston are made the same, but in practice the variations of the two curves from symmetry are so slight that they will not show up in the cam surfaces **166***a* and **166***b* if standard tolerances are employed.

Alternately, in the embodiment of Figure 1*a*, curve **200** may be incorporated in cam surface **166***b* and curve **202** may be incorporated in cam surface **166***a*.

Parenthetically, it may be mentioned that the graph of Figure 10 is for only 120 degrees of crank angle (a three cylinder double acting embodiment being illustrated in Figure 1), though the curves shown merely repeat themselves twice over the remaining 240 degrees.

Thus, when the motor **10** rotates in the direction of the arrow of Figure 1, the cam **36** controls the mechanical advantage through which the incoming hydraulic liquid acts on the crank shaft, while the cam **38** controls the mechanical advantage through which the exhaust flow of hydraulic liquid acts in applying torque to the crank shaft. Cam **36** also controls the liquid acceptance capacity of the motor while cam **38** controls the liquid discharge thereof. On reverse operation, the functions of the cams are reversed since the curve **200** when reversed about a vertical axis at 180 degrees of crank angle will be the same as curve **202** and vice versa, as described more specifically hereinafter. The cams **36** and **38** in actuating the respective compensator pistons provide additional lever arms for the hydraulic liquid to act through on the crank shaft. If the respective compensator pistons are actuated in accordance with curves **200** and **202**, the additional lever arms provided will exactly compensate for the variations in mechanical advantage between the pistons of cylinders **12** and the crank shaft over the cycle of operation of the motor.

The graphs of Figures 9, 9*a*, and 10 are developed as follows:

The composite displacement of the respective double acting pistons of the motor **10** (which is the curve illustrated in Figure 9) and the spread of the composite displacement over a cycle of operation may be used as a basis for determining the information required to permit the mechanical advantage between the pistons and the shaft to be controlled. To determine this total displacement, and its spread over the cycle of operation, the displacement of each piston over each degree of crank angle rotation is measured and tabulated. Moreover, since each cylinder **12** of the illustrated motor is of the double acting type, the crank end of the cylinder must also be accounted for in the same manner. Another factor that must be allowed for is that the respective pistons are at different positions in their strokes for any degree of crank angle rotation.

More specifically, the resulting data needed for plotting Figure 9 is obtained by determining and tabulating the displacement of each piston, or the distance that each piston moves, for or between each degree of rotation of the crank shaft, and then adding all of the displacements for each degree of crank angle together to arrive at the data which, when plotted against degrees of crank angle,

will provide the composite piston displacement curve for the hydraulic mechanism.

Thus, taking first one of the cylinders, the distance that the head end of the piston moves for each degree of rotation of the crank shaft is separately tabulated. The same tabulation is made for the other two cylinders, displacing them 120 degrees in the tabulation with respect to the first cylinder and to each other, since they are mounted 120 degrees apart. Then the displacement of the crank ends of the pistons for each degree of rotation of the crank shaft is established and tabulated, the figures being appropriately displaced according to the position of the particular piston with respect to the others. The total figures for each degree of rotation of the crank shaft are added up, and the result is the total displacement of the hydraulic mechanism for each degree of rotation of the crank shaft. The total displacement is equivalent to the liquid receiving or acceptance capacity of the cylinders **12**, as well as to the liquid discharge flow.

When the total displacement or acceptance capacity figures are plotted against degree of crank angle rotation, a curve such as curve **204** of Figure 9 is developed. It should be kept in mind, however, that the curve **204** is for only 120 degrees of crank angle, though the curve merely repeats itself twice through the remaining 240 degrees, or once for each cylinder **12**.

Stating it in other words, it being known that a piston moves at modified simple harmonic motion, and that the volume of hydraulic liquid in the cylinder at any given time has a known relationship to that modified simple harmonic motion, one takes the displacement of the piston for each increment of movement, and adds to that displacement the displacement for the other pistons at that position of the crank. The figures are then totaled to arrive at the basic information used in plotting curve **204**.

The curve **204** as plotted may be used to obtain the information required to arrive at the compensator piston displacement or movement necessary to operate compensators **35** (or compensator **35***a*), whether the hydraulic mechanism is operated either as a pump or as a motor or in forward or reverse directions, and whether a particular compensator is connected to either the intake or exhaust side of the hydraulic system.

The data for plotting curve **200** may be obtained from the graph of Figure 9 as follows: Referring to Figure 9, it will be noted that the curve **204** has the well known peaks **206** and valleys **208** that are normally associated with displacement curves of hydraulic cylinders that act on a crank shaft or its equivalent. Since the displacement or liquid receiving capacity of a cylinder is directly related to the work output of the cylinder, curve **204** also represents the composite work output curve of the hydraulic cylinders **12** under constant pressure test conditions; the reason for this is that work is performed as the respective pistons move through each increment of displacement and under constant pressure or constant volume flow conditions the composite work output curve of the cylindeds would have the peaks and dips impressed on the resulting figures by the composite displacement figures.

As one of the requirements for providing a constant work and torque output of the hydraulic motor **10** is the elimination of the peaks **206** and valleys **208** of the curve **204**, it became apparent to me that the displacement or liquid receiving capacity of the hydraulic cylinders must be changed so that the curve **204** will become a straight and horizontal hydraulic liquid displacement or capacity line.

I have found that the hydraulic liquid receiving capacity, and thus the effective composite displacement of the hydraulic system, may be made constant over 360 degrees of crank angle rotation by establishing straight line **210** and adjusting same so that the shaded areas underneath curve **204** and above line **210**, for 360 degrees of crank angle rotation, are exactly equal to the shaded areas above curve **204** and below line **210**. The line **210** then becomes a median displacement or hydraulic liquid capacity line and when the total effective composite displacement of the hydraulic system is kept at this figure, the peaks and valleys characteristic of pulsating composite cylinder output will be entirely eliminated.

In practice, the establishment of line **210** may be implemented by employing conventional devices for measuring areas on graphs. It will be found that the area under curve **204** will exactly equal the area under line **210** when these lines are established as described above.

The curve **200** of graph **10**, which is to provide the design for the compensator on the intake side of the system, is then evolved by analyzing the curves **204** and **210** for the function desired for each degree of crank angle rotation, and plotting the results against degrees of crank angle rotation of the crank shaft. For instance, referring to Figure 9, at 10 degrees of crank angle rotation, the hydraulic cylinders collectively without the compensators **35** (or **35***a*) have a hydraulic liquid receiving capacity represented by the distance between points **212** and **214**. If the composite liquid acceptance capacity of the cylinders is to be made uniform or constant, at 10 degrees of crank angle rotation the hydraulic liquid receiving capacity must be increased by the amount represented by the distance between points **212** and **216**. Therefore, the compensator, for instance, a compensator **35**, connected to the intake of the hydraulic system, to perform its function, must increase the total liquid receiving capacity of the mechanism by this amount, and accordingly, the piston **158** of the compensator **25** must move away from the head end of the cylinder **156** a corresponding amount, depending upon the area of the head end of the piston. This distance will establish point **218** on the graph of Figure 10.

This compensator piston movement permits the hydraulic liquid acting on the compensator piston to apply a force to the crank shaft through a lever arm, the length of which is determined by the angle of the cam surface with respect to the axis of rotation of crank shaft **14** at 10 degrees of crank angle. Since the cam roller **164** will be moving down into a valley of the cam surface at this position of crank angle, torque will be applied to the crank shaft in the same direction that it rotates, and thus is added to the shaft. The additional lever arm adds to the mechanical advantage through which the hydraulic liquid acts at this point in the cycle of operation.

At 30 degrees of crank angle rotation, the composite liquid receiving capacity of the hydraulic mechanism is represented by the distance between points **220** and **222**; as the hydraulic liquid within the mechanism, or its acceptance capacity, at this point will then be too great if the constant composite acceptance capacity is to be maintained, the liquid capacity represented by the distance between points **222** and **224** must be subtracted from the composite capacity of the hydraulic mechanism. This means that the piston **158** of said compensator **35** must be moved toward the head end of cylinder **156** and the amount of movement of said piston **158** required to reduce the capacity the amount represented by the distance between the points **222** and **224** will establish point **226** on the graph of Figure 10.

This compensator movement requires the crank shaft to do work on the compensator piston to force same toward the head end of its cylinder under the pressure of the hydraulic liquid in the cylinder. The hydraulic liquid is therefore acting on the crank shaft through a lever arm, the length of which is determined by the angle of the cam surface with respect to the axis of rotation of crank shaft **14** at 30 degrees of crank angle. Since the cam roller **164** will be moving up from a valley of the cam surface at this position of crank angle, torque will be subtracted from the crank shaft, and this lever arm

subtracts from the mechanical advantage of the pistons of cylinders **12** at this point in the cycle of operation.

It will thus be seen that a complete analyzation of the graph of Figure 9 in this manner for each degree of crank angle rotation will permit one to plot the curve **200**, which defines the compensator piston movement required to make the liquid receiving capacity of the hydraulic mechanism constant over its cycle of operation, or through one revolution of the crank shaft. By incorporating curve **200** in cam **36**, the compensating lever arm will be provided for making the mechanical advantage constant that the incoming hydraulic liquid acts through on the crank shaft. It may be added that the foregoing assumes hat the entire graph represented by the showing of Fgure 9 is used, including the aforesaid omitted lower portion.

Analyzation of the curves **204** and **210** for liquid discharge operation (to which they apply since liquid acceptance capacity is equivalent to discharge flow) for each degree of crank angle rotation will establish curve **202**. Thus, referring to Figure 9, at 10 degrees of crank angle the hydraulic cylinders **12** collectively without the compensators **35** (or **35***a*) have a liquid discharge capacity represented by the distance between points **212** and **214**. Since line **210** represents the median displacement line, if the liquid discharge of the cylinders is to be made constant or uniform, at 10 degrees of crank angle rotation, hydraulic liquid must be added to the discharge in an amount equivalent to that represented by the distance between points **212** and **216**. Therefore, the compensator, for instance, a compensator **35**, connected to the exhaust side of the hydraulic system must force an equivalent amount of liquid into the discharge, and accordingly the piston of this compensator must move toward the head end thereof a corresponding amount, depending upon the area of the head end of the piston. This will establish a point **240** on curve **202**, and at this position of crank angle the exhaust hydraulic liquid is acting through a lever arm to apply countertorque to the shaft in addition to that supplied by the back pressure acting through cylinders **12**. The length of this lever arm is determined by the angle of the cam surface with respect to the axis of rotation of crank shaft **14** at 10 degree of crank angle rotation. Since the cam roller **164** will be moving up from a valley of cam surface at this position of crank angle, countertorque will be added to the crank shaft, and this lever arm adds to the mechanical advantage that the exhaust side hydraulic liquid acts through on the crank shaft.

At 30 degrees of crank angle rotation, the liquid discharge is represented by the distance between points **220** and **222**, and if the liquid discharge is to be made constant, hydraulic liquid must be withdrawn from the discharge in an amount equivalent to that represented by the distance between points **222** and **224**. Therefore, the compensator, for instance, said exhaust side compensator **35**, connected to the exhaust side of the hydraulic system must withdraw this amount of liquid from ths discharge, and accordingly the piston **158** of this compensator must move away from the head end thereof a corresponding amount, depending upon the area of head end of the piston. This will establish point **242** on curve **202**, and at this position of crank angle, the exhaust hydraulic liquid is acting through a lever arm to subtract from the countertorque that is applied to the crank shaft by the back pressure acting through cylinders **12**. The length of this lever arm is determined by the angle of the cam surface with respect to the axis of rotation of the crank shaft at 30 degrees of crank angle rotation. Since the cam roller **164** will be moving down into a valley of the cam surface at this position of crank angle, countertorque is subtracted from the crank shaft, and this lever arm subtracts from the mechanical advantage that the exhaust side hydraulic liquid acts through on the crank shaft.

A similar analyzation of curves **204** and **210** for each degree of crank angle will permit one to plot curve **202**, which defines the compensator piston movement necessary to make the liquid discharge of the hydraulic mechanism constant or uniform over its cycle of operation. By incorporating curve **202** in cam **38**, the compensating lever arms will be provided for making the mechanical advantage constant that the exhaust hydraulic liquid acts through on the crank shaft.

The embodiment of Figure 1*a* may be employed in making this analyzation with the same results, though it should be kept in mind that both ends of the single cylinder **156***a* are employed, and not just the head end thereof.

Thus, in accordance with my invention, the compensator pistons are actuated in accordance with curves **200** and **204** by employing the crank shaft, which is provided with cams **36** and **38** to effect the actual piston movement and this has the effect of providing the aforementioned additional lever arms through which the hydraulic liquid may act on the crank shaft. When the respective pistons are actuated in accordance with curves **200** and **202**, it will be found that the additional lever arms provided will exactly compensate for the variations in mechanical advantage between the pistons of cylinders **12** and the crank shaft.

The same result may be obtained by applying the same analyzation to the composite torque input curve of motor **10**, which is the curve shown in Figure 9*a*. When composite torque input under constant pressure test conditions is plotted against degree of crank angle rotation, the composite torque input curve **204***a* of the mechanism to the shaft (without the compensators) results.

If the total or composite torque input to the crank shaft is to be made constant over the cycle of operation of the mechanism, a median torque input line represented by line **210***a* must be established, and the torque increments added to and subtracted from the crank shaft in accordance with the variations between these two curves.

It may also be assumed for purposes of this disclosure that curve **204***a* represents the composite torque curve for both the pressure and exhaust sides of the system, since differences in operating pressures merely amplify the torque curves, and have no affect on mechanical advantage.

In practice, the required torque increments to be added and subtracted from the shaft may be determined by following a procedure similar to that described above with respect to displacement or capacity increments, a torque input curve **204***a* being established by the torque input of each cylinder for each degree of crank angle under constant pressure test conditions being separately tabulated and then added, with the data of the respective cylinders being appropriately displaced, to provide a composite torque input curve when the resulting data is plotted against crank angle of rotation. The median torque input line **210***a* may then be established in the same manner as the median liquid capacity line. Under constant pressure test conditions, the torque curve will show the precise variations in mechanical advantage over the cycle of operation of the motor, and by operating the motor in accordance with the median torque line, the mechanical advantage through which the hydraulic liquid acts on the shaft will be made constant.

With the foregoing in mind, the graph of Figure 9*a* may be analyzed as follows with respect to the compensator for the intake side of the hydraulic mechanism: When the crank shaft is turned between zero and 10 degrees of crank angle rotation, the torque input by the pressure side of the system varies from point **230***a* to point **212***a*, and the work input to the shaft is represented by the area between points **212***a*, **214***a*, **232***a*, and **230***a*; the torque input at 10 degrees should have been that indicated by point **216***a* and should have been constant between zero and 10 degrees of crank angle rotation, if

a constant torque output is to be maintained. Thus, torque must be added to the shaft at each increment of crank angle rotation between zero and 10 degrees to bring the torque applied by the pressure side of the system up to line **210***a* and therefore work must be applied to or done on the shaft in the amount indicated by the area between points **212***a*, **216***a*, **234***a*, and **230***a*.

In accordance with my invention, the compensator that is on the pressure or intake side of the system applies this increment of torque and work to the shaft as it is actuated to increase the acceptance capacity of the motor. As discussed above with respect to the compensator **35** on the intake side of the system, the piston **158** moves away from the head end of cylinder **156** at 10 degrees of crank angle rotation, which means that the cam roller **164** will be moving down into a valley of the cam surface formed on the cam **36** with which it cooperates. Torque is thus being transmitted into, and work done on, the crank shaft and the movement of the compensator piston that will apply the required torque to the shaft at 10 degrees of crank angle will establish point **218** on curve **200**. I have found that the torque added to the crank shaft by the compensator on the intake side of the motor maintains the uniform torque input (of the pressuure side of the system to the shaft) established by line **210***a*.

At 30 degrees of crank angle, the torque input of the pressure side of the hydraulic system without the compensator is the amount represented by point **222***a* and the work done in moving the shaft between zero and 30 degrees of crank angle is represented by the area under curve **204***a* between points **222***a*, **220***a*, **232***a*, and **230***a*. The torque input should have been that represented by point **224***a* and the excess work applied to the shaft is represented by the area between points **222***a*, **224***a*, and **264***a*. The compensator on the intake side of the system cancels out this excessive torque and work input from the shaft as its piston is moved toward the head end of its cylinder in reducing the liquid receiving capacity of the hydraulic system at this point the required amount in the cycle of operation; the torque reduction required at 30 degrees of crank angle to maintain the uniform torque input curve established by the median torque input line will establish point **226** on curve **200**.

Similar remarks apply to the remaining portions of the curves of Figure 9*a*, and consequently, it will be seen that the curve **200** when applied to a cam **36** will insure that the pressure or intake side of the hydraulic system applies a constant torque to the crank shaft **14**. Again, it should be kept in mind that the above discussion assumes one is working with the complete graph of Figure 9*a*, including the omitted lower portion.

Thus, the compensator on the pressure side of the hydraulic system in establishing a constant liquid receiving capacity for the hydraulic system also makes constant or uniform the total torque input to the crank shaft by the pressure side of the hydraulic system.

The reason for this is clear if the added and subtracted capacity and torque increments are considered in terms of work increments. Of course, torque multiplied by angular displacement equals work, and work is done when a piston moves through the volume equivalent to a capacity increment. When the compensator increases the liquid acceptance capacity of the hydraulic system, the compensator piston is moved through a space or volume equivalent to the increase in capacity, which means that the hydraulic system does work on the compensator piston, and a corresponding increment of work is applied to the crank shaft. This increment of work is equivalent to the difference between the work input to the crank shaft by the pressure side of the system acting through the cylinders **12** without the compensator at a particular position of crank angle rotation, and the work input required by an established pressure side torque input median line. When the compensator decreases the liquid acceptance capacity of the hydraulic system, the compensator piston is moved through a space or volume equivalent to the decrease in capacity, and a corresponding increment of work is applied to the hydraulic system by the compensator piston; of course, a corresponding work increment is subtracted from the work applied to the crank shaft by the pressure side of the system through cylinders **12**. This latter work increment is equivalent to the difference between the work input to the crank shaft by the pressure side of the hydraulic system acting through cylinders **12** without the compensator at a particular position of crank shaft rotation and the work input required by an established pressure side torque input median line.

The discharge side of the system may be analyzed in like manner with respect to a torque curve **204***a*. The hydraulic pressure on the exhaust side of the system applies countertorque to the crank shaft both through the pistons of cylinders **12** and the exhaust side compensator when this compensator is displaced to add hydraulic liquid to the discharge flow. When the exhaust side compensator piston is displaced to remove hydraulic liquid from the discharge flow, torque is added to the crank shaft to compensate for the excess countertorque applied thereto by the back pressure of the discharge side. The torque increments required at each degree of crank angle will establish curve **202**.

Thus, the compensator on the discharge side of the hydraulic system in making the liquid discharge constant or uniform over the cycle of operation of the motor, also makes uniform the torque input to the crank shaft or countertorque by the exhaust side of the hydraulic system.

The same analyzation applied to compensators **35** applies also to compensator **35***a*, the only real difference between these two forms of the invention in so far as they apply to three double acting cylinders or six single acting cylinders being that the cam surfaces operate on a single double acting cylinder. The sides of the compensator **35***a* that are connected to the intake and exhaust sides of the motor perform the same functions as the respective compensators **35**. Also, since the cam surfaces **166***a* and **166***b* are substantially the reverse of each other, the motor employing compensator **35***a* may be operated in either direction with equal facility. Therefore, no further description of the arrangement of Figure 1*a* is deemed necessary.

With respect to the embodiment of Figure 11, the compensators **35** perform in the same manner, and the design of their controlling cams is arrived at in like manner.

Operating conditions of motor

While the compensators above described, when applied to a positive displacement hydraulic motor, make constant the mechanical advantage that hydraulic liquid acts through in applying torque to the motor crank shaft, over the cycle of operation of the motor, the conditions of operation selected by the designer must control hydraulic pressure in accordance with my governing expression if constant cyclical torque output is to be achieved. These conditions of operation, which are enumerated hereinbefore, are best illustrated by diagrammatic Figures 12–15 illustrating simplified one-way hydraulic circuits. In these figures, motor **300** is intended to represent a mechanism similar to that shown in Figure 1 but without the compensators applied unless specifically indicated. In each of these figures, pump **42** draws hydraulic liquid from source **40** and forces it into pressure line **22** of the motor, and after passing through the cylinders **12** (not shown) of the motor, the liquid passes from return line **24** back to the source **40**. In these circuits, off-on flow control valves are omitted for simplicity of illustration, it being assumed that liquid flow is stopped by stopping

pump 42. Motor 300 in each case drives a shaft 302 that extends from both sides of the motor.

In the circuit of Figure 12, the pressure on the exhaust side of the motor is assumed to be atmospheric which means that F_2A_2 equals zero and the governing expression applying is $F_1A_1=C$. Therefore, a compensator 35 acting on a cam 36 keyed or otherwise coupled to shaft 302 must be applied to the pressure side of the motor to make A_1 constant, and F_1 is made constant by employing a liquid control valve 44 between pump 42 and the motor cylinders. Valve 44 in the illustration of Figure 12 may be either a constant volume liquid flow control valve or a constant pressure control valve, since either will make the pressure of the hydraulic liquid on the pressure side of the motor, and thus F_1, constant over the cycle of operation of the motor. If pump 42 is a constant pressure pump, such as a centrifugal pump operated at constant r.p.m., or a constant volume supply pump, such as a positive displacement pump, valve 44 is not required.

The circuit of Figure 13 is similar to that of Figure 12 except that it is assumed that the pressure on the discharge side of the system is greater than atmospheric, which means that a compensator 35 cooperating with cam 38 keyed or otherwise coupled to shaft 302 is required. As the hydraulic liquid discharging to source will be under constant pressure, F_2 will be constant, and a liquid control valve is not required between the exhaust side compensator and the cylinders of motor 300.

In the circuit of Figure 14, which is applicable to conditions wherein it is desirable not to have a hydraulic liquid control valve between pump 42 and motor 300, the valve 44 is on the discharge side of the hydraulic circuit, and it must be a constant volume flow control valve. Since a single compensator 35 is employed, it must be on the same side of the hydraulic system as the valve 44, as shown in Figure 14. In the embodiment of Figure 14, F_1 and A_1 will vary since a compensator is not employed on the pressure side of the system and it is assumed the pump 42 is not a constant pressure pump; the valve 44 being of the type shown in Figure 5 will vary F_2 in proportion to the variations of F_1 and A_1 and keep the torque output in accordance with my governing expression. The application of a valve 44 to the pressure side of the system or making pump 42 a constant pressure pump would merely make F_1 constant but would not affect A_1.

The embodiment of Figure 15 is likewise applicable to conditions where it is desirable not to have a flow control valve between the pump 42 and the motor. The flow control valve 44 is positioned on the discharge side of the system and a compensator 35 is employed on both the pressure and discharge sides of the system. Assuming pump 42 is not a constant pressure pump, F_1 will vary and valve 44 must be a constant volume control valve, so that F_2 will vary to balance the variation of F_1. If F_1 is made constant as by making pump 42 a constant pressure pump, valve 44 may be either a constant volume flow control valve or a constant pressure flow control valve.

In the illustrations of Figures 13 and 15, motor 300 may be made reversible by employing conventional four-way control valves in the manner indicated in Figure 4.

A significant feature of the embodiments of Figures 4, 13 and 15 is that in effect the pressure difference between the pressure and discharge side of the motor is made constant, which means that the torque output varies only as the pressure differential between the sides of the system varies.

Where either a pressure regulating or control valve or a volume flow control valve may be employed to satisfy my governing expression, the designer may choose between pressure or volume hydraulic liquid control conditions. Constant volume liquid control conditions will provide a uniform angular torque output, with constant r.p.m. regardless of the load imposed on the shaft, while constant pressure control will provide uniform angular torque at a uniform angular velocity that steadily increases, remains constant, or decreases depending upon the load applied to the shaft. Changes in volume flow under volume control conditions will, of course, change r.p.m., while changes in pressure under pressure control conditions will create, speed up, or slow the rate of change in velocity occurring under an assumed load condition.

The valve 80 shown in Figure 5 for completeness of disclosure only is a conventional form of volume flow regulating valve. As illustrated, it includes a valve body 82 in which is formed an inlet 84. The inlet 84 leads to an eccentrically mounted flow control shaft 86 and thence to a flow outlet 88. The flow passing from outlet 88 is rendered constant by a hydrostatic compensating device generally indicated at 90, which includes a double headed piston 92 biased toward the right hand side of the figure by a spring 94.

The function of the various structural elements illustrated in Figure 5 is well known in the art, so no further description thereof is considered necessary, though it may be mentioned that the primary function of these valves is to provide accurate volume control in hydraulic circuits regardless of any variation in the imposed liquid pressure. Rotation of the eccentrically mounted shaft 86 increases or decreases the clearance through which the hydraulic liquid must pass, and this in turn varies the volume of liquid passing through the valve.

The valve diagrammatically illustrated in Figure 7 is likewise shown only for completeness of disclosure, and is a conventional valve designed to provide a constant pressure regardless of the volumes of fluid flow imposed upon it. The valve 180 generally comprises a housing 182, including an outlet 184 and positioned about a valve seat 185 seating a ball 186. A screw threaded adjusting member 188 is employed to press spring 190 against the ball 186. The valve 180 is interposed in the hydraulic system when constant pressure conditions are required. By adjusting the member 188 to apply the appropriate amount of pressure against the ball 186, the pressure of the hydraulic liquid leaving the outlet of the valve is made constant.

It may be added that valve 80 will also provide a constant pressure where compensators are employed on both sides of the system.

Design of compensator cams for pump

The hydraulic mechanim of Figures 1 and 6 may be operated as a pump (10*a* of Figure 8) by removing drum 20 and connecting shaft 50 of Figure 8 to crank shaft 14, as by any suitable form of coupling. The pressure and return lines 22 and 24 are connected to conduits 201 and 203 extending to the source of liquid 54 and the liquid receiving receptacle 56, respectively. The pump 10*a* may be driven in either a forward (the direction of the arrow of Figure 1) or reverse direction, depending on the direction of operation of motor 52. Compensators 35 (or compensator 35*a* if three double acting cylinders or six single acting cylinders are employed) are employed in pump 10*a* without any change whatsoever in the design of their controlling cams.

In operation, the compensator on the intake side of the line operates against a suction, and consequently, makes little or no contribution to the operation of the pump. The compensator on the exhaust side operates to control the liquid discharge in the manner described above to provide a constant or uniform liquid output, as distinguished from a pulsation output. On change of the direction of rotation, the previously inactive compensator becomes the discharge controlling compensator, and performs in the same manner that the first compensator did. Compensator 35*a* operates in a similar manner.

The compensator on the discharge side of the pump

makes constant the mechanical advantage that the back pressure in the discharge flow acts through in applying countertorque to pump crank shaft. The basic conditions of operation are shown in diagrammatic Figure 16 wherein pump **310** employing a hydraulic mechanism such as that illustrated in Figure 1 without the compensators driven by constant torque motor **312** through shaft **314**. Pump **310** draws hydraulic liquid from source **54** into line **22** and discharges it from line **24** back into source **54**. A compensator **35** acting on a cam **38** keyed or otherwise coupled to shaft **314** must be associated with the discharge side of the system. Since F_1 is assumed to be zero, the governing expression becomes $F_2A_2=-C$, as hereinbefore mentioned, and as the pressure on the discharge side of the pump will be constant, my rule is satisfied without flow control valves.

When mechanisms herein disclosed are to be operated as pumps, the compensator on the intake side may be omitted unless reverse operation is desired. Motor **52** preferably is of the type that will apply a constant torque to shaft **50**.

The embodiment of Figure 11 may be operated as a pump in the same manner as described with respect to mechanism **10**.

Relation of cam surfaces to specific hydraulic mechanisms

It may be stated at this point that the cam surface controlling the operation of a compensator **35** (or **35***a*) will be different for each cylinder arrangement forming a hydraulic motor or pump. Thus, the addition to or removal of cylinders from a hydraulic device will require an entirely different cam surface, since the fluctuations of displacement and mechanical advantage will occur at different points in the cycle of operation. Moreover, the type of positive displacement device employed will have a bearing on the design of the cam surface required.

Advantages of invention

It will therefore be seen that I have provided a mechanical advantage compensating method and device, which, when applied to positive displacement hydraulic devices together with my governing expression for controlling operation, provide a hydraulic device or mechanism of constant cyclical work output that is adaptable to a wide variety of applications. When employed for motor operation, it is possible to have a very high torque at low r.p.m. with stepless speed control. When employed for pump operation, a smooth and non-pulsating discharge is provided.

One of the most important features of the invention is that the hydraulic liquid that is drawn from the hydraulic system is not lost or thrown out of it, but remains in the system. Thus, the torque increments that are withdrawn from and added to the crank shaft do not represent subtractions from the total torque input of the system, since the torque increments that are subtracted from the shaft to remove the peaks of the torque input curve are not lost to the system but are added to fill in the valleys of the torque input curve. Moreover, the liquid removed at the exhaust side of the system to level discharge peaks is replaced when it is needed to keep the discharge at the desired median volume.

A further important feature of the invention is that no valve is required to change the direction of operation of the compensators, other than valves serving the function of four-way control valve **46**. And, of course, the direction of rotation of the mechanism, whether operating as a motor or as a pump, may be changed by merely appropriately positioning valve **46** if compensators are employed on both sides of the hydraulic system.

It may also be added that the elements making up the invention are few and simple and it is contemplated that standard cylinders and pistons may be employed. These piston type devices run for extended periods without leakage and are easily repaired. While the compensators are preferably controlled by cams, a linkage controlling arrangement that provides the desired piston movement is considered possible. The cams may be coupled with the crank shaft as by being fixed or keyed directly thereto, or through gearing or otherwise. Of course, the cams and compensators need not be located within the mechanism housing, as demonstrated by the embodiments of Figures 12–16.

The number of cylinders employed in a particular hydraulic device to which my invention is applied is not critical, and they may be of either the single or double acting type, though more than one and probably at least three single acting cylinders will be required when only single acting cylinders are to be employed. My invention is applicable whether an even or an odd number of cylinders are employed. Moreover, my invention may be applied to any type of hydraulic mechanism of the positive displacement type wherein there is a pulsating output. This includes mechanisms having a crank shaft, swash plate or eccentric or other means of converting reciprocating motion to rotary motion, or vice versa, which actuate pistons or other positive displacement structures such as vanes, diaphragms, gears, and any other means of producing positive hydraulic displacement.

In the appended claims, the following terms are to be construed as having the meanings indicated:

The term "crank shaft" means crank shafts of or similar to the type illustrated, as well as all other structures generally associated with that term, such as swash plates, eccentrics, or any other means for converting reciprocating motion into rotary motion or vice versa.

The term "piston" means pistons of or similar to the type illustrated as well as vanes, diaphragms, gears or any other means of producing positive hydraulic displacement when subject to movement.

The term "cylinder" means both the single and the double acting type unless specifically stated otherwise, and is further intended to mean the housing or other structure defining the chamber in which any "piston" as defined immediately above operates.

The term "plurality of cylinders" in so far as it refers to single acting cylinders means more than two single acting cylinders.

The term "constant pressure providing means" is intended to include both volume and pressure flow control valves and only under conditions wherein these valves are interchangeable at the choice of the designer.

The term "constant mechanical advantage cam surface" as used in the appended claims means cam surfaces of the types herein disclosed and their equivalents that are shaped to control the reciprocations of the hydraulic device compensator piston or pistons as required over the cycle of operation of the device to render the effective mechanical advantage of the device constant.

The term "compensating curve" is intended to include the various cam curvatures necessary to drive the compensator piston through its proper compensating cycle.

The foregoing description and the drawings are given merely to explain and illustrate my invention and the manner in which it may be performed, and the invention is not to be limited thereto, except in so far as the appended claims are so limited, since those skilled in the art who have my disclosure before them will be able to make modifications and variations therein without departing from the scope of the invention.

I claim:

1. Apparatus for rendering substantially constant the torque output of a hydraulic motor of the type having a crank shaft driven by pistons and cylinders adapted to be supplied with hydraulic liquid through conduit means under pressure through liquid flow orienting valve means interposed in said conduit means, and means interposed in said conduit means for maintaining the hydraulic liquid supplied to said cylinders under a constant superatmospheric pressure, said apparatus comprising a compensator

cylinder and piston, the compensator cylinder being in free communication with said hydraulic liquid between said third mentioned means and the cylinders through conduit means, a cam coupled to said crank shaft and having a constant mechanical advantage cam surface for driving said compensator piston through a predetermined cycle conforming to the compensator piston movement required to make constant over the cycle of operation of the motor the mechanical advantage through which the hydraulic liquid supplied to the cylinders acts in applying torque to the shaft, and a torque transmitting connection between said cam surface and said compensator piston.

2. In a hydraulic motor including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said shaft and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders and including hydraulic liquid flow orienting valve means controlling the flow of liquid to and from the respective cylinders, intake and exhaust conduit means extending between said valve means and the cylinders, a hydraulic liquid supply under pressure, conduit means extending between said hydraulic liquid supply and said valve means, and means for maintaining said hydraulic liquid supply at a constant pressure that is greater than atmospheric pressure, the improvement wherein a mechanical advantage compensating device is associated with the motor for rendering constant the mechanical advantage that the hydraulic liquid supply acts through in applying torque to the crank shaft, said device comprising a cylinder in open communication with the hydraulic liquid supply between the last mentioned means and the motor cylinders through conduit means, a piston reciprocably mounted in said device cylinder, cam means including a cam surface coupled to said crank shaft for rotation therewith for reciprocating said device piston to permit the hydraulic liquid to displace said device piston within its cylinder whereby hydraulic liquid is withdrawn from the hydraulic liquid supply and to force hydraulic liquid from within said device cylinder into the hydraulic liquid supply, and a torque transmitting connection between said device piston and said cam surface of said cam means, said surface of said cam means being a constant mechanical advantage cam surface, whereby, on rotation of the shaft said compensator device piston is reciprocated to displace same within said device cylinder over the cycle of operation required to make constant the mechanical advantage that the hydraulic liquid supply acts through in applying torque to the crank shaft.

3. In a hydraulic motor including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said shaft, and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders including hydraulic liquid supply and exhaust conduit means extending between said hydraulic liquid supply and said cylinders, and hydraulic liquid flow orienting means interposed in said conduit means for controlling the flow of liquid to and from the respective cylinders, the improvement wherein the composite mechanical advantage between the crank shaft and the pistons is made substantially constant over the cycle of operation of the motor by compensator means acting between the hydraulic liquid passing through said cylinders and the crank shaft, said compensator means comprising a cylinder in free communication with the hydraulic liquid passing through the cylinders of the motor through conduit means, a piston reciprocably mounted in said compensator means cylinder, cam means including a constant mechanical advantage cam surface coupled to said crank shaft for reciprocating said compensator means piston to permit the hydraulic liquid to apply torque to the shaft during dips in the composite mechanical advantage between the motor pistons and the crank shaft and to permit the crank shaft to apply pressure on the liquid during peaks in said composite mechanical advantage over the cycle of operation of the motor, and a torque transmitting connection between said compensator means piston and said cam means cam surface.

4. The improvement set forth in claim 3 wherein said plurality of cylinders comprises three double acting cylinders and wherein said compensator means cylinder is a double acting cylinder, one end of said compensator means cylinder being in communication with the hydraulic liquid supply of all of the motor cylinders and the other end of said compensator means cylinder being in communication with the hydraulic liquid discharge of all of the motor cylinders.

5. In a positive displacement rotary hydraulic device including a plurality of reciprocable hydraulically operated pistons and a rotatable crankshaft in which the pistons are operably connected by torque transmitting connections, the method of making constant the composite mechanical advantage between the pistons and the rotatable crank shaft, said method including the steps of applying torque and countertorque to the crank shaft as required to compensate for the dips and peaks of said composite mechanical advantage over the cycle of operation of the pistons and crank shaft.

6. Apparatus for rendering substantially constant the torque output of a hydraulic motor of the type including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said crank shaft and a hydraulic system for supplying hydraulic pressure liquid through conduit means to said cylinders, said conduit means including hydraulic liquid flow orienting means, and conduit means for returning the hydraulic liquid from the cylinders to said system, said apparatus comprising a compensator cylinder and piston, a constant volume liquid flow control valve interposed in the last mentioned conduit means and controlling the exhaust liquid flow from said motor cylinders, said compensator cylinder being in free communication through conduit means with the exhaust liquid flow of all said motor cylinders between said motor cylinders and said control valve, a cam coupled to said crank shaft and including a constant mechanical advantage cam surface for driving said compensator piston through a predetermined cycle conforming to the compensator piston movement required to make constant over the cycle of operation of the motor the mechanical advantage through which the exhaust liquid flow acts in applying torque to the shaft, and a torque transmitting connection between said cam surface and said compensator piston.

7. In a hydraulic motor including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said crank shaft, and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders including hydraulic liquid flow orienting valve means controlling the flow of liquid to and from the respective cylinders, intake and exhaust conduit means extending between said valve means and the cylinders, and a constant volume liquid flow control valve interposed in said system downstream of said valve means, the improvement wherein a mechanical advantage compensating device is associated with the motor for rendering constant over the cycle of operation of the motor the mechanical advantage that the hydraulic liquid discharge of the motor pistons acts through in applying torque to the crank shaft, said device comprising a cylinder in open communication with the hydraulic liquid discharge through conduit means connecting the two upstream of said constant volume liquid flow control valve, a piston reciprocably mounted in said device cylinder, cam means including a constant mechanical advantage cam surface coupled to said crank shaft for

rotation therewith for reciprocating said device piston to permit the hydraulic liquid to displace said device piston within said device cylinder whereby hydraulic liquid is withdrawn from the hydraulic liquid discharge and to force hydraulic liquid from within the cylinder into the hydraulic liquid discharge, and a torque transmitting connection between said device piston and said cam means cam surface, said cam means cam surface on rotation of the shaft reciprocating said compensating device piston to displace same within said device cylinder over the cycle of operation required to make constant over the cycle of operation of the motor the mechanical advantage that the hydraulic liquid discharge acts through in applying torque to the crank shaft.

8. In a hydraulic motor including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said crank shaft, and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders including hydraulic liquid flow orienting valve means controlling the flow of liquid to and from the respective cylinders, intake and exhaust conduit means extending between said valve means and the cylinders, a hydraulic liquid supply under pressure, conduit means extending between said supply and said valve means, and means for maintaining said hydraulic liquid supply at a constant pressure, the improvement wherein a mechanical advantage compensating device is associated with the motor in free communication with the hydraulic liquid supply to all of the motor cylinders between the last mentioned means and the motor cylinders through conduit means and wherein a second mechanical advantage compensating device is associated with the motor in free communication with the hydraulic liquid exhaust of all of said cylinders through conduit means, said compensating devices each comprising a cylinder and a piston reciprocably mounted therein, cam means including constant mechanical advantage cam surfaces coupled to said crank shaft for rotation therewith for respectively reciprocating said pistons of the respective compensating devices to permit the hydraulic liquid to displace said pistons of said compensating devices within their respective cylinders whereby hydraulic liquid is withdrawn from the hydraulic liquid supply and exhaust of the motor cylinders, respectively, and to force hydraulic liquid from within the respective cylinders of the respective compensating devices into the hydraulic liquid supply and exhaust of the motor cylinders respectively, and torque transmitting connections between said cam means cam surface and the respective pistons of said compensating devices, said cam means on rotation of the shaft reciprocating the respective pistons of said compensating devices to displace same within their respective cylinders over the cycles of operation required to make constant over the cycle of operation of the motor the mechanical advantage that the hydraulic liquid of the hydraulic system acts through in applying torque to the crank shaft.

9. In a hydraulic motor including a plurality of cylinders each having a piston movably mounted therein, a crank shaft driven by said pistons, torque transmitting connections between said pistons and said crank shaft, and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders including a hydraulic liquid flow orienting valve means controlling the flow of liquid to and from the respective cylinders, intake and exhaust conduit means extending between said valve means and the cylinders, a hydraulic liquid supply under pressure, supply and return conduit means extending between said hydraulic liquid supply and said valve means, and constant volume liquid flow control valve means interposed in said return conduit means for controlling the flow of the hydraulic liquid discharge of the motor cylinders, the improvement wherein a mechanical advantage compensating device is associated with the motor in free communication through conduit means with the hydraulic liquid supply to all of the motor cylinders, and wherein a second mechanical advantage compensating device is associated with the motor in free communication through conduit means with the hydraulic liquid discharge of all of the motor cylinders between said constant volume liquid flow control valve means and the motor cylinders, said devices each comprising a cylinder and piston reciprocably mounted therein, cam means including constant mechanical advantage cam surfaces coupled to said crank shaft for rotation therewith for respectively reciprocating said pistons of the respective compensating devices to permit the hydraulic liquid to displace said pistons of said compensating devices within their respective cylinders whereby hydraulic liquid is withdrawn from the hydraulic liquid supply and the hydraulic liquid discharge, respectively, and to force hydraulic liquid from within the respective cylinders of said compensating devices into the hydraulic liquid supply and the hydraulic liquid discharge, respectively, and torque transmitting connections between the respective pistons of said compensating devices and said cam means cam surfaces, said cam means on rotation of the shaft reciprocating the respective pistons of said compensating devices to displace same within their respective cylinders over the cycle of operation required to make constant over the cycle of operation of the motor the mechanical advantage that the hydraulic liquid of the hydraulic system acts through in applying torque to the crank shaft.

10. In a hydraulic motor of the type utilizing a rotatable shaft, a source of hydraulic liquid under pressure, a plurality of cylinders having pistons movably mounted therein and coupled to the shaft by torque transmitting means for driving same under the motivating action of the hydraulic liquid when supplied to said cylinders through conduit means extending therebetween and the hydraulic liquid source, and regulating means for controlling the flow of the hydraulic liquid through the motor, the improvement wherein the liquid flow through the motor and the mechanical advantage between the shaft and said hydraulic liquid is made to conform to the formula

$$F_1A_1-F_2A_2=C$$

wherein F_1 is the force provided by the pressure of the hydraulic liquid on the intake sides of the cylinders, A_1 is the mechanical advantage through which F_1 acts on the shaft, F_2 is the counterforce provided by the pressure of the hydraulic liquid on the exhaust sides of the cylinders, A_2 is the mechanical advantage through which F_2 acts on the shaft, and C is cyclical torque output of the shaft, and wherein at least one of said mechanical advantage factors and C is held constant.

11. A hydraulic motor comprising more than two single acting hydraulic cylinders, a piston movably mounted in each cylinder, a crank shaft driven by said pistons, pressure and return conduit means connected to a source of hydraulic liquid under pressure, said pressure and return conduit means communicating with hydraulic liquid flow orienting valve means for controlling the flow of liquid to and from the respective cylinders, constant volume liquid flow control means connected to said valve means, a single liquid conduit extending between said valve means and each cylinder, said liquid conduits serving both as the intake and the exhaust conduits for the respective cylinders, a hydraulic compensator associated with the motor to provide a constant cyclical torque output, said compensator comprising a cylinder in free communication through conduit means with said constant volume liquid control means between said constant volume liquid control means and the motor cylinders, a piston reciprocably mounted in said compensator cylinder, cam means including a constant mechanical advantage cam surface coupled to said crank shaft for rotation therewith for reciprocating said compensator

piston to draw hydraulic liquid into said compensator cylinder and force hydraulic liquid from said compensator cylinder, and a torque transmitting connection between said cam means cam surface and said compensator piston, said cam means on rotation of the shaft reciprocating said compensator piston over the cycle of operation of the motor in conformity to the displacement required to level the peaks and fill in the valleys of the curve developed by plotting increments of total piston displacement of the motor against corresponding increments of crank angle revolution of the motor.

12. In a hydraulic motor including a plurality of hydraulic cylinders operably connected with the motor crank shaft for actuating same, the method of providing a uniform instantaneous torque output over the cycle of operation of the motor which includes varying the hydraulic acceptance and discharge capacities of the motor and the torque input to the crank shaft as required to fill in the valleys and level off the peaks of the curves developed by plotting increments of total displacement and torque input against corresponding increments of rotation of the crank shaft.

13. In a positive displacement rotary hydraulic motor device including a rotatable crankshaft and a plurality of hydraulic cylinders operably connected with the crank shaft for actuating same, the method of providing a constant mechanical advantage between the hydraulic liquid passing through the cylinders and the crank shaft over the cycle of operation of the device which includes providing an expanadable chamber in communication with the hydraulic liquid passing through the cylinders of the device, and employing the rotation of the crank shaft to expand and contract the chamber against the pressure of the hydraulic liquid to thereby extract and add hydraulic liquid increments to and from the hydraulic liquid passing through the cylinders of the device in proportion to the increments of work required to make the said mechanical advantage constant.

14. In a hydraulic motor including a plurality of hydraulic cylinders containing pistons operably connected with a crank shaft for rotating the latter, the method of providing a constant torque output which includes varying the summation of forces that are applied to the crank shaft in accordance with the variations in mechanical advantage between the pistons and the crank shaft over the cycle of operation of the motor as required to achieve a constant torque output over the cycle of operation of the motor.

15. A hydraulic motor comprising more than two single acting hydraulic cylinders, a piston movably mounted in each cylinder, a crank shaft operably connected to each piston, pressure and return conduit means, said pressure and return conduit means communicating with hydraulic liquid flow orienting valve means for controlling the flow of liquid to and from the respective cylinders, hydraulic liquid pressure regulating means connected to said valve means, a single liquid conduit extending between said valve means and each cylinder, said liquid conduits serving both as the intake and the exhaust conduits for the respective cylinders, a hydraulic compensator associated with the motor to provide a constant cyclical torque output, said compensator comprising a cylinder in free communication with said pressure regulating means between said pressure regulating means and the motor cylinders, a piston reciprocably mounted in said compensator cylinder, cam means coupled to said crank shaft for rotation therewith for reciprocating said compensator piston to draw hydraulic liquid into said compensator cylinder and force hydraulic liquid from said compensator cylinder, and torque transmitting means between said cam means and said compensator piston, said cam means on rotation of the shaft reciprocating said compensator piston over the cycle of operation of the motor in conformity to the displacement required to level the peaks and fill in the valleys of the curve developed by plotting increments of total piston displacement of the motor against corresponding increments of crank angle revolution of the motor.

16. In a hydraulic motor including a plurality of cylinders each having a piston displaceably mounted therein, a crank shaft operably connected to said pistons by torque transmitting connections, and a hydraulic system for supplying hydraulic liquid under pressure to said cylinders including hydraulic liquid flow orienting means controlling the flow of liquid to and from the respective cylinders and a source of hydraulic liquid under pressure, the improvement wherein the composite mechanical advantage between the crank shaft and the pressure side of the hydraulic system is made substantially constant over the cycle of operation of the motor by compensator means acting between the pressure side of the hydraulic system and the crank shaft, and wherein the composite mechanical advantage between the crank shaft and the exhaust side of the hydraulic system is made substantially constant over the cycle of operation of the motor by a second compensator means acting between the exhaust side of the hydraulic system and the crank shaft, each of said compensator means comprising a cylinder and a piston reciprocably mounted in the respective compensator means cylinder, the cylinder of the first mentioned compensator means being in free communication through conduit means with the pressure side of the hydraulic system and the cylinder of the second compensator means being in free communication conduit means with the exhaust side of the hydraulic system, said pistons of the respective compensator means being reciprocated by cam means coupled to the crank shaft, said cam means including constant mechanical advantage cam surfaces reciprocating the pistons of the respective compensator means as required to apply torque to the shaft during dips in the respective composite mechanical advantages between the motor pistons and the crank shaft and to permit the crank shaft to do work on the hydraulic liquid during peaks in said composite mechanical advantages over the cycle of operation of the hydraulic motor, said flow orienting means comprising directional valve means for changing the direction of hydraulic liquid flow through the motor cylinders whereby the motor may operate in forward and reverse directions, said compensator means cylinders communicating with the hydraulic system between said directional valve means and the motor cylinders, said first mentioned compensator means and said second compensator means automatically reversing their operations on reversal of hydraulic liquid flow through the motor cylinders.

17. In a hydraulic motor including a plurality of hydraulic piston and cylinder assemblies operatively mounted in torque applying relation with a member to be rotated and a source of hydraulic liquid under pressure for driving the member through torque transmitting means under the motivating action of the hydraulic liquid when supplied to said assemblies, said member being the driven component of the motor, and means for controlling the condition of flow of the hydraulic liquid through the motor, the improvement wherein the hydraulic liquid flow through the motor and the mechanical advantage between the member and said hydraulic liquid conforms to the formula

$$F_1A_1-F_2A_2=C$$

wherein F_1 is the force provided by the pressure of the hydraulic liquid on the intake sides of said assemblies, A_1 is the mechanical advantage through which F_1 acts on the member to be rotated, F_2 is the counterforce provided by the pressure of the hydraulic liquid on the exhaust sides of said assemblies, A_2 is the mechanical advantage through which F_2 acts on the member to be rotated, and C is cyclical torque output of the member to be rotated,

and wherein at least one of said mechanical advantage factors and C are held constant.

18. Hydraulic motor apparatus comprising a constant mechanical advantage drive arrangement including a plurality of piston and cylinder assemblies, a rotatably mounted member forming the driven component of the apparatus, said piston and cylinder assemblies being mounted in torque applying relation to said rotatably mounted member and each being connected thereto by a torque transmitting connection, a hydraulic system means for supplying hydraulic liquid under pressure to said assemblies for actuating same, under the motivating action of the hydraulic liquid when supplied to said assemblies, to rotate said member, and means for making constant the mechanical advantage through which the hydraulic liquid acts on said member to rotate same, said system means including a source of hydraulic liquid under pressure, pressure and exhaust conduit means connecting the respective assemblies and the hydraulic liquid

source, and a hydraulic liquid movement control means for controlling the condition of hydraulic liquid flow through said apparatus as required to, together with said constant mechanical advantage drive arrangement, provide a uniform angular torque output at said rotatably mounted member, over the cycle of operation of the apparatus, when the hydraulic liquid is supplied to said assemblies.

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