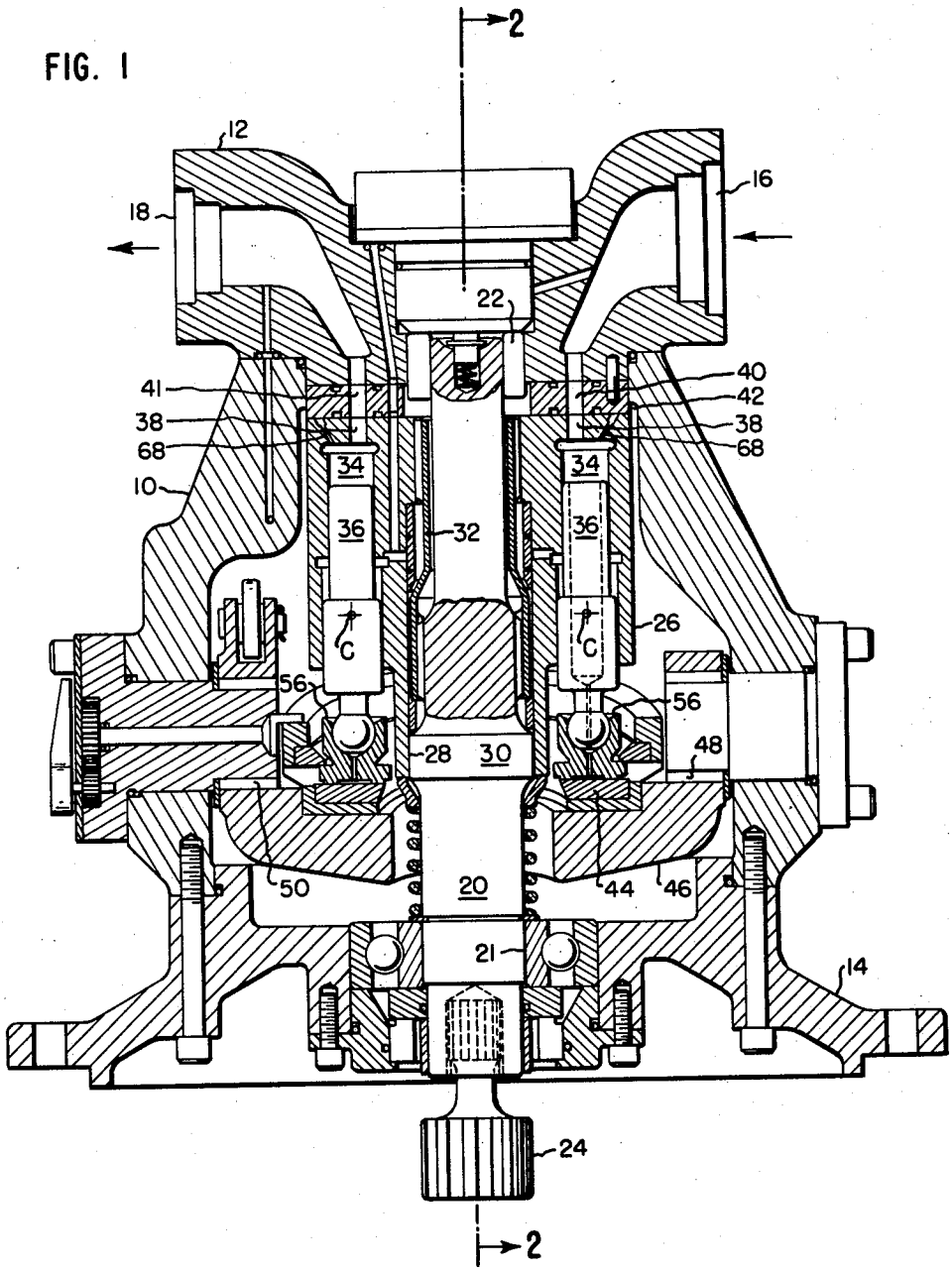
March 31, 1964 W. L. KLINE 3,126,835

FLUID PUMP

Filed Dec. 4, 1959 3 Sheets-Sheet 1

FIG. 1



INVENTOR.
WILLIAM L. KLINE

BY

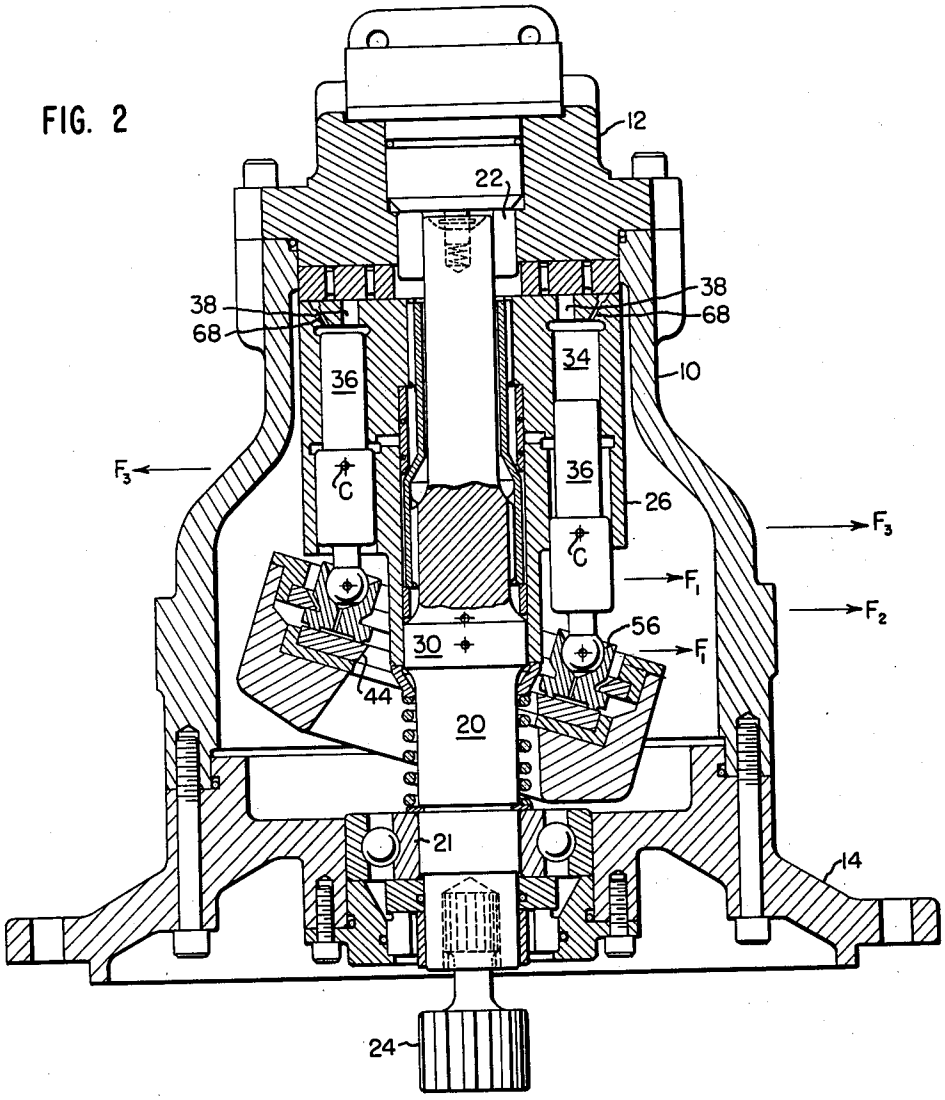
Kenway, Jenney, Witter & Hildreth
ATTORNEYS

March 31, 1964 W. L. KLINE 3,126,835

FLUID PUMP

Filed Dec. 4, 1959 3 Sheets-Sheet 2

FIG. 2



INVENTOR.
WILLIAM L. KLINE

BY

Kenway, Jenney, Witter & Hildreth
ATTORNEYS

March 31, 1964 W. L. KLINE 3,126,835

FLUID PUMP

Filed Dec. 4, 1959 3 Sheets-Sheet 3

FIG. 3

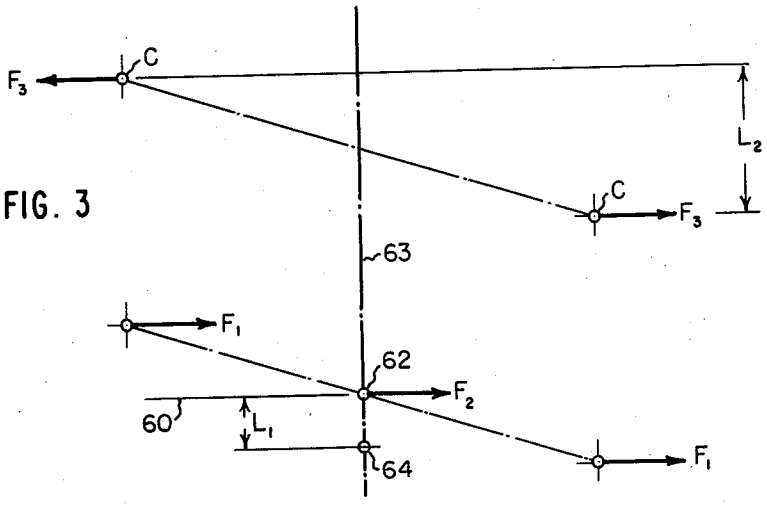
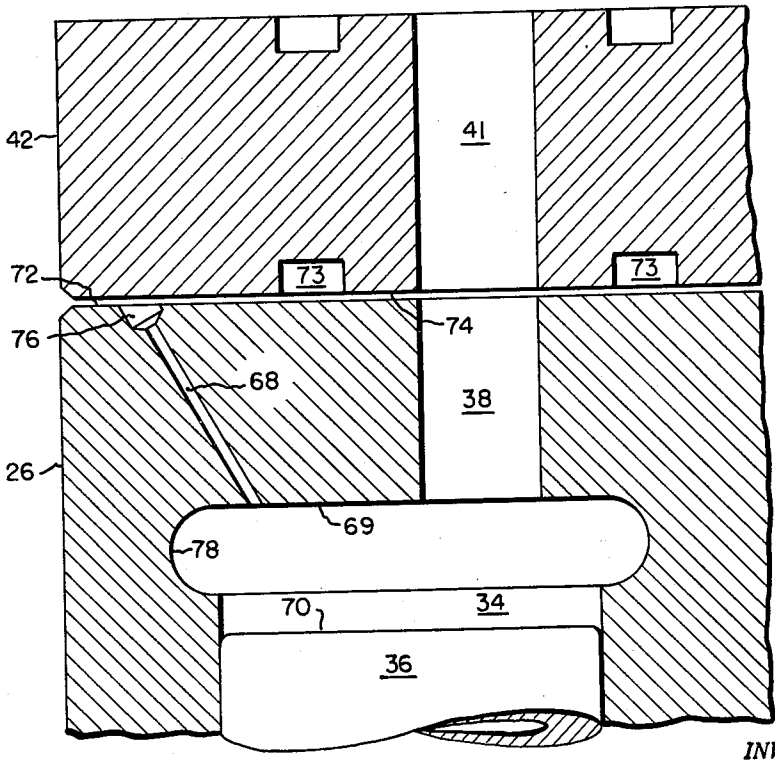


FIG. 4



INVENTOR.
WILLIAM L. KLINE

BY

Kenway, Jenney, Witter & Hildreth

ATTORNEYS

United States Patent Office 3,126,835 Patented Mar. 31, 1964

3,126,835

FLUID PUMP

William L. Kline, 10425 Redbank Road, Galena, Ohio

Filed Dec. 4, 1959, Ser. No. 857,284
2 Claims. (Cl. 103—162)

This invention relates to a fluid pressure energy translating device of a multi-piston and cylinder type wherein the device comprises a cylindrical block or barrel carrying a plurality of pistons circularly arranged concentrically of the barrel axis and slidably mounted in the barrel for reciprocation parallel with the barrel axis with an inclinable wobble or cam plate being disposed coaxially of the barrel and operatively connected to the pistons to effect reciprocation of the pistons in response to relative rotation between the cam or swash plate and the barrel.

A fluid pressure energy translating device of an exemplary type with which this invention is concerned further comprises means supporting the barrel for rotation about its axis and a valve plate or the like engaged or mated with the end of the barrel opposite the cam plate and having a pair of ports or passages for connection respectively to a source of fluid and to a discharge line. The ports in the valve plate are registerable with ports or passages in the barrel which are individual connected to the piston chambers or cylinders in the barrel so that fluid will alternately be introduced into and discharged from each cylinder as the barrel is rotated relative to the cam plate. During rotation of the barrel of a device of the general type described, there is a tendency for the barrel to tilt or laterally upset, which may adversely affect the efficiency of the device inasmuch as the end surface of the barrel mating with the valve plate tends to be urged into a cocked or inclined position relative to the mating surface of the valve plate, resulting in increased wear of the mating surfaces of the valve plate and barrel as well as possible loss of fluid pressure due to leakage between the mating surfaces of the barrel and valve plate. Also, increased wear of the mating surfaces of the valve plate and barrel may occur through insufficient lubrication between these parts of the device, permitting metal-to-metal contact.

Accordingly, it is an object of this invention to provide in a fluid pressure energy translating device of the type described a novel and improved arrangement for laterally suppoorting the barrel carrying the pistons, whereby the tendency of the barrel to tip or tilt during operation of the device is minimized or substantially eliminated.

It is another object of this invention to provide in a fluid pressure energy translating device of the type described novel and improved means for improving the lubrication of the mating surfaces of the barrel and valve plate as well as having other attendant advantages.

In one aspect of the invention, the rotatable barrel is laterally supported by bearing means disposed coaxially within the end of the barrel opposite the valve plate and next adjacent the cam plate. In accordance with this invention, the lateral center line of the support bearing, which center line extends at right angles to the barrel axis, is located at a predetermined distance below the point of intersection of the barrel axis and the general plane of the locus of the points of operative engagement between the pistons and inclined cam plate. The resultant radial component of force between the pistons and cam plate will thus act on the moment arm corresponding to the displacement of said lateral center line of the bearing from said point of intersection to provide a barrel upsetting moment in one direction about the center of the barrel lateral support bearing. Further in accordance with the invention, the value of this barrel upsetting moment in said one direction is selected to correspond generally to a second barrel upsetting moment, acting in a direction opposite said one direction, and caused generally by centrifugal force acting on the centers of gravity of diametrically opposed pistons in the barrel. This second upsetting moment on the barrel occurs, during rotation of the barrel, because of the offsetting, longitudinally of the barrel axis, of the position of the centers of gravity of opposed pistons. Thus, by spacing the lateral support for the barrel from the intersecting point of the barrel axis and the general plane of the locus of the points of operative engagement of the pistons and cam plate on the opposite side of said intersecting point from said valve plate, the tendency of the barrel to tilt is minimized or even substantially eliminated.

Another aspect of this invention contemplates the provision of bleed passages leading from the piston cylinders to the end face of the barrel in a region where it mates with a smooth uninterrupted surface on the underside of the valve plate mating with the barrel. These bleed passages provide forced lubrication of the mating surfaces of the valve plate and barrel by means of a pressure which varies with the pressures in the cylinders. Another distinct advantage of these bleed passages is that the fluid pressure of the flow through said bleed passages will vary with abnormally high transient pressures in the cylinders so as to minimize the effects of unequal hydraulic clamp forces between the barrel and valve plate, thus reducing the possibility of barrel seizure. A more detailed understanding of these aspects as well as other aspects, objects and advantages of the invention may be had by reference to the following detailed description of the invention when this description is taken in connection with the accompanying drawings, in which:

FIG. 1 is a longitudinal cross sectional view of an exemplary fluid pressure energy translating device of the type described incorporating the present invention;

FIG. 2 is a longitudinal cross sectional view of the device of FIG. 1 substantially along the line 2—2 of FIG. 1;

FIG. 3 is a diagrammatic representation of the upsetting moments acting upon the barrel of the device of FIG. 2 during rotation of the barrel; and

FIG. 4 is a fragmentary enlarged sectional view of a portion of the device of FIG. 2.

With reference to the drawings, and particularly FIGS. 1 and 2, an exemplary fluid pressure energy translating device of a type with which this invention is concerned generally comprises a hollow housing or body 10 on one end of which is mounted a port cap 12 and on the other end of which is mounted a flange mount or base 14. The port cap 12 is provided with a fluid intake port or passage 16 and a discharge port or passage 18 leading into the housing 10. A shaft 20 is disposed generally coaxially within the housing 10 and is rotatably supported at one end by a suitable bearing 22 on the port cap 12. A shaft coupling 24 is engaged with the end of the shaft adjacent the base 14 for drivingly connecting the shaft to suitable means for rotating the same.

Also disposed within the housing 10 is a generally cylindrical barrel or cylinder block 26 having a coaxial throughbore coaxially receiving the shaft 20. The barrel is provided at its lower end with a coaxial sleeve-like extension or skirt 28, the inner diameter of which provides a bearing surface engaged on a support bearing on the shaft 20 provided by an increased diameter land 30 on the shaft. A torque tube 32 is fitted coaxially over the upper portion of the shaft 20 and is coaxially received within the barrel bore and drivingly connected at its lower end to the shaft by a spline connection. The upper end of the torque tube 32 is drivingly connected to the barrel 26 by a spline connection, whereby upon rotation

of the shaft 20 the barrel will be correspondingly rotated and the support bearing 30 will laterally support the barrel coaxially of the shaft. It should be noted that the barrel 26 is radially spaced from the housing 10 and does not rotatably bear on any member in a radial direction. For a reason which will be hereinafter apparent, it is preferred that the spline connection between the torque tube 32 and shaft 20 and barrel 26 be constructed to permit slight radial displacement between connected members while still maintaining a firm rotative drive connection.

The barrel 26 is further provided with a plurality of elongated cylinder chambers 34 circularly arranged concentrically about the barrel axis and extending parallel to the barrel axis. A plurality of pistons 36 are slidably received respectively within the chambers 34 for reciprocation parallel to the barrel axis. The barrel 26 is further provided with an intake and discharge passage 38 leading from each of the chambers 34 at the end thereof next adjacent the upper end of the barrel. Each of the passages 38 is directed to register at its outer end with a pair of passages 40 and 41 extending through a valve plate 42 disposed between the upper end of the barrel and the port cap 12. As is usual in this type of device, each of the passages 40 and 41 in the valve plate arcuately extends only partially about the barrel axis, with these passages respectively communicating with the intake port 16 and discharge port 18 of the port cap 12.

Disposed adjacent to but axially spaced from the lower end of the barrel 26 is an annular or ring-like cam plate 44 disposed coaxially of the barrel and mounted on an inclinable hanger 46 which is journalled as at 48 and 50 for tilting relative to the barrel axis about an axis extending at right angles to the axis of the shaft 20. Further, as can be seen from FIGS. 1 and 2, the lower ends of the pistons 36 project outwardly of the corresponding end of the barrel 26 and are each provided with a generally spherical end engaged in a generally spherical seat in a shoe member 56 slidably engageable with the cam plate 44 to provide a universal operative connection between the cam plate and pistons.

In the operation of the device thus far described as a pump, rotation of the shaft 20 will correspondingly result in rotation of the barrel 26 relative to the inclined cam plate 44 to effect reciprocation of the pistons 36 relative to the barrel. As the barrel is rotated, fluid will be drawn in through the intake port 16, intake passage 40 in the valve plate 42, and the intake and discharge passages 38 in registry with the intake passage 40 in the valve plate. Further rotation of the barrel will cause these passages 38 to be moved out of registry with the intake passage 40 and be moved into registry with the discharge passages 41 in the valve plate, whereupon movement of the piston 36 toward the valve plate will cause pressurized fluid to be delivered outwardly through the discharge port 18 in the port cap. The angle of inclination of the cam plate will determine the displacement of the pump and thus the quantity of fluid delivered per revolution. The displacement of the pump may, of course, be varied by varying the inclination of the cam plate.

As most clearly seen from FIG. 2, the pressurized fluid in the chambers 34 exerts an axial thrust on the pistons 36, which thrust is transferred to the inclined cam plate 44 through the spherical piston ends and piston shoes 56. Thus, at the centers of each of the spherical piston ends there will be a radial component of force F1 tending to move the piston ends to the right as viewed in FIG. 2. The forces F1 may be resolved into a resultant force F2 acting on the barrel 26 in the same direction as the forces F1 and extending from the point of intersection of the axis of the shaft 20 and the general plane of the locus of the centers of the spherical piston ends about the shaft axis.

Further, as can be seen from FIG. 2, during rotation of the barrel 26 the offsetting of the centers of gravity of the combined masses of diametrically opposed pistons 36 and volumes of fluid being acted upon by the pistons will result in unequal centrifugal forces being applied to the barrel at varying distances longitudinally of the barrel. Within an accuracy satisfactory for the purposes of this invention, the mass of the fluid being acted upon by the pistons may be ignored and only the masses of the pistons considered. The centrifugal force on each piston acts through the center of gravity C of the piston in a direction radially of the barrel. The centrifugal force of diametrically opposed longitudinally offset pistons will thus apply a dynamic couple to the barrel which is a product of the centrifugal force acting on one of the pistons times the longitudinal offsetting of the centers of gravity of the opposed pistons. The value of this couple on the barrel will vary from zero in the case of pistons in which the centers of gravity are aligned at right angles to the shaft 20, as in the case of the pistons shown in FIG. 1, to a maximum value in the case of diametrically opposed pistons in their maximum offset position, as shown in FIG. 2. The gradient of the couple about the shaft axis may be resolved as shown in FIGS. 2 and 3 into resultant forces F3 extending at right angles to the shaft axis and the axis of inclination of the cam plate and spaced apart longitudinally of the shaft axis by an amount equal to the stroke L2 of the pistons. From FIG. 3 it can be seen that this centrifugal force caused couple F3L2 tends to tilt the barrel in a counterclockwise direction.

In accordance with the invention, and with reference to FIGS. 2 and 3, the lateral center line 60 of the barrel support 30 is offset below the point 62 on the shaft axis 63 at which the general plane of the locus of the centers of the spherical ends of the pistons intersects the shaft axis. Accordingly, the resultant force F2 directed radially of the shaft axis and at right angles to the axis of inclination of the cam plate 44 will exert a moment F2L1 on the barrel 26 and about the center 64 of the barrel support 30, where L1 is the distance along the shaft axis 63 from the intersection point 62 to the center 64 of the support 30. This moment F1L1 will, as is apparent from FIGS. 2 and 3, tend to tilt the barrel about the center 64 of the support 30 in a clockwise direction as viewed in FIG. 3 and in opposition to the dynamic couple F2L1, by reason of the location of the support 30 a predetermined distance below the intersection point 62 or, in other words, on the opposite side of said point 62 from the upper end of the barrel. This predetermined distance L1 between the center of the lateral support land 30 and intersection point 62 is selected so that for a given speed of rotation of the barrel 26 and inclination of the cam plate 44 the moment F2L1 will correspond to the couple F3L2 and thus substantially eliminate any tendency of the barrel to tilt relative to the shaft axis. Therefore, within a reasonable range of pump operation, the location of the support 30 below the median point 62, as taught by this invention, will minimize the tendency of the barrel to tilt, even though the opposing moments F2L1 and F3L2 are not exactly equal.

The counterbalancing of the moment F2L1 against the dynamic couple F3L2 assures more uniform contact of the end of the piston barrel 26 and valve plate 42, inasmuch as the tendency of the barrel to tip and separate or "blow off" from the valve plate is minimized or substantially eliminated. The tendency of the barrel to tilt even where a side or lateral loading occurs on the supporting land 30 is minimized by constructing the shaft 20 so that its diameter is not constant between its bearing supported ends and so that the point of maximum shaft deflection will occur at the midpoint of the support land 30. Accordingly, lateral forces on the shaft will be reflected at the barrel support land 30 by a displacement of the land in a direction laterally of the shaft axis to tend to cause a lateral displacement rather than a tilting of the barrel axis. The torque tube 32 connecting the shaft 20 and barrel

26 permits the barrel to shift radially while still maintaining the barrel end and valve plate in intimate uncocked contact. Thus, the location of the barrel support land 30 below the median point of operative engagement between the pistons 36 and cam plate 44 substantially eliminates tilting of the barrel within a reasonable predetermined range of operation, while the preferred construction of the shaft 20 in cooperation with the torque tube 32 tends to convert any tendency of the barrel to tilt to a tendency to merely shift laterally or radially of the shaft axis so as to maintain the mating surfaces of the barrel and valve plate in proper uncocked relation.

The pressurized fluid within the cylinders 34 will provide a force tending to urge the barrel 26 axially toward the valve plate 42. Also, as will be apparent from FIG. 4, there will be an oppositely directed hydraulic force acting on the end of the barrel in the area thereof bounded by the bleed passages 73 in the valve plate 42. This latter force tends to urge the barrel away from the valve plate. The net hydraulic force tending to urge the barrel toward the valve plate is referred to as the hydraulic clamp force. It is, of course, undesirable for the mating surfaces of the valve plate and barrel to be spaced apart any substantial distance; however, it is desirable to provide an oil film between these surfaces to minimize wear of the parts. The hydraulic clamp forces urging the valve plate and barrel mating surfaces together tend to act in opposition to the provision of an oil film between the valve plate and barrel, and thus there is provided forced lubrication of the mating surfaces of the barrel and valve plate with the pressure of the lubricating fluid being automatically varied in response to changes in the hydraulic clamp forces. To this end, as most clearly shown in FIG. 4, the barrel 26 is provided with a bleed passage 68 respectively associated with each of the cylinders 34. Each of the bleed passages 68 extends from the cylinder end wall 69 facing the active end 70 of the piston 36 to the barrel end surface 72 mating with the underside 74 of the valve plate 42. The outer end of the passage 68 terminates in an enlarged portion 76 registering with a smooth uninterrupted portion of the underside of the valve plate. As will be apparent, during movement of the piston 36 toward the end wall 69, a portion of the high pressure fluid in the cylinder 64 will be forced through the passage 68 to lubricate the mating surfaces 72 and 74 and preclude metal-to-metal contact of these parts. The pressure of the fluid in the passage 60 will vary with the fluid pressure acting on the cylinder end wall 69, and in the case of transient overpressures in the cylinder a correspondingly higher lubricating fluid pressure will simultaneously be provided to maintain the necessary lubrication.

Further, as will be apparent, when a transient overpressure occurs in a cylinder 34, the resulting transitory increase in the hydraulic clamp force in the area of that cylinder may tend to cause the barrel 26 and valve plate 42 to seize. However, the corresponding occurring high pressure in the respective bleed passage 68 will provide a separating force between the valve plate and barrel at the same localized area as the abnormally high hydraulic clamp force, with the result that there will be no seizure of the valve plate and barrel. In this connection, and with reference to FIG. 4, it is preferred that the diameter of the bleed passage be selected with respect to the normal running clearance between the surfaces 72 and 74 so that the pressure drop through the passage 68 is somewhat less than the pressure drop from the outer end of the passage 68 to the interior of the housing at the periphery of the surfaces 72 and 74. Thus, under normal conditions the clearance between the surfaces 72 and 74 will be the "control valve" for the bleed fluid. However, if the clearance between the valve plate and barrel should materially increase to the point that the pressure drop through the bleed passage is greater than that from the outer end of the bleed passage to the interior of the housing, then, of course, the bleed passage will become the limiting restriction. Correspondingly, the fluid pressure at the outer end of the bleed passage will drop rapidly, thus preventing undesirable separation or "blow off" of the barrel. Further, if the clearance between the valve plate and barrel decreases below normal, the pressure at the outer end of the bleed passage will rise to prevent seizure of the parts. Thus it can be seen that the bleed passages 68 not only provide for improved lubrication of the mating surfaces of the barrel and valve plate but also provide a fluid counteraction for overpressures or underpressures in the cylinders, with an inherent positive feedback being provided in the system to prevent "blow off" or seizure.

In order to reduce the circulation of particles of dirt or other foreign matter in the device, which particles may damage the close fitting surfaces or clog small passages such as the bleed passages 68, there is included in the device of FIG. 1 a dirt trap to collect these particles. With reference to FIG. 4, it can be clearly seen that at the upper end of each cylinder 34 there is provided an annular cavity 78 in the side wall of the cylinder. During rotation of the barrel, any dirt or wear particles will tend to be contrifuged out into the cavity 78, where they are entrapped and prevented from further circulating in the fluid system.

While this invention has been described in terms of preferred specific embodiments thereof, it will, of course, be understood that various changes and substitutions could be made in the structure shown and described without departing from the invention. Accordingly, the foregoing description and accompanying drawings are to be taken only in an illustrative sense, and the invention is to be considered to be limited only by the appended claims, which shall include within their scope all structure which logically falls within the language of these claims.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. In a fluid pressure energy translating device of the type described, a housing, a rotatable barrel disposed in the housing in radially spaced relation thereto, a plurality of pistons slidably received in cylinders in the barrel and circularly arranged concentrically of the barrel for reciprocation parallel to the barrel axis, one end of each of the pistons projecting from the adjacent end of the barrel, a cam plate disposed adjacent one end of the barrel and mounted on the housing for inclination about an axis at right angles to the barrel axis, spherical ball and socket joints between the projecting ends of the pistons and said cam plate to provide for reciprocation of the pistons in said cylinders relative to the barrel in response to rotation of the barrel relative to the cam plate, said joints including shoes engaging said cam plate, a valve plate engaging the opposite end of the barrel for controlling the flow of fluid into and out of said cylinders, a coaxial cylindrical sleeve-like extension on said one end of the barrel, a drive shaft for the barrel extending coaxially through the barrel and said sleeve-like extension and rotatably supported by axially spaced bearings carried by said housing, a coaxial annular land intermediate the said axially spaced bearings of the shaft and coaxially engaged within said sleeve-like extension to laterally support the barrel coaxially of the shaft, said land having a lateral center line spaced axially outwardly from the point of intersection of the shaft axis and general plane of the locus of the centers of the spherical balls at said projecting ends of the pistons and located in a plane between said plane of said centers and said cam plate, and a torque tube disposed coaxially over the shaft and within the barrel, said tube being splined at axially spaced positions axially inwardly of said shaft bearings to the shaft and barrel respectively and being splined at one end to the shaft adjacent said land and axially inwardly of said land and its opposite end to said barrel at the end of the barrel remote from the cam plate.

2. In a fluid pressure energy translating device of the type described, a housing, a rotatable barrel disposed

within the housing in concentric relationship thereto, a plurality of pistons slidably received in cylinders in the barrel and circularly arranged concentrically of the barrel for reciprocation parallel to the barrel axis, one end of each of the pistons projecting from the adjacent end of the barrel, a cam plate disposed adjacent one end of the barrel, spherical ball and socket joints between the projecting ends of the pistons and said cam plate to provide for reciprocation of the pistons in said cylinders relative to the barrel in response to relative rotation of the barrel and cam plate, said joints including shoes engaging said cam plate, a valve plate engaging the opposite end of the barrel for controlling the flow of fluid into and out of said cylinders, means providing lateral support for the barrel, said lateral support means including a shaft which supports said barrel and which in turn is supported by axially spaced bearings carried by the housing and an annular intermediate bearing surrounding said shaft, said annular bearing having a lateral center line spaced axially outwardly from the point of intersection of the shaft axis and general plane of the locus of the centers of the spherical balls at said projecting ends of the pistons and located in a plane between said plane of said centers and said cam plate, and a torque tube disposed axially over the

shaft and within the barrel, said tube being splined at axially spaced positions axially inwardly of said shaft bearings to the shaft and barrel respectively and being splined at one end to the shaft adjacent said annular bearing and axially inwardly of said bearing and being splined at its opposite end to said barrel at the end of the barrel remote from the cam plate.

References Cited in the file of this patent

UNITED STATES PATENTS

1,908,612	Johnson	May 9, 1933
1,924,629	Thoma	Aug. 29, 1933
2,298,850	Vickers	Oct. 13, 1942
2,480,069	Wright	Aug. 23, 1949
2,608,159	Born	Aug. 26, 1952
2,642,810	Robinson	June 23, 1953
2,826,922	Wiggermann	Mar. 18, 1958
2,845,941	Wagner	Aug. 5, 1958
2,915,985	Budzich	Dec. 8, 1959
2,925,046	Budzich	Feb. 16, 1960

FOREIGN PATENTS

164,725	Australia	Aug. 22, 1955
455,931	Canada	Apr. 12, 1949