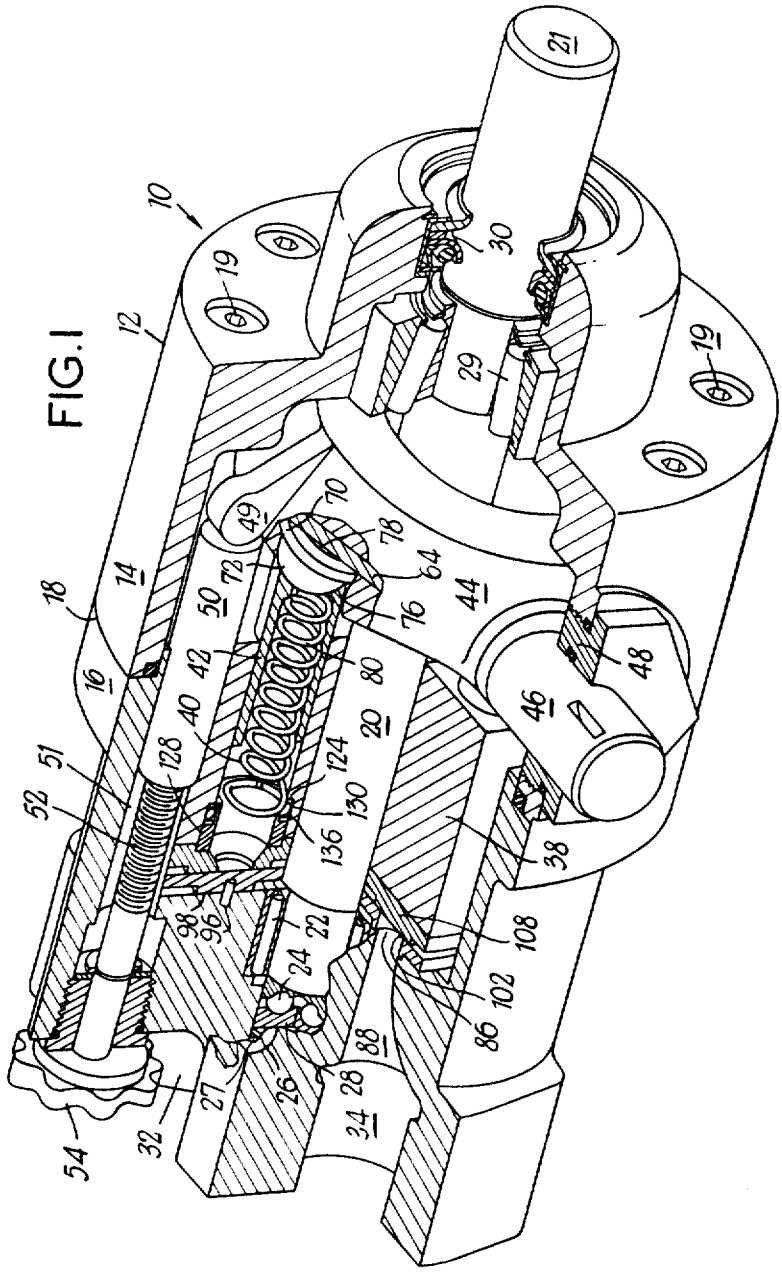
Jan. 14, 1969 M. A. D'AMATO Re. 26,519

VARIABLE DISPLACEMENT PUMP

Original Filed Oct. 16, 1962 Sheet 1 of 4

FIG. 1



INVENTOR
MICHAEL A. D'AMATO

BY James H Littlepage

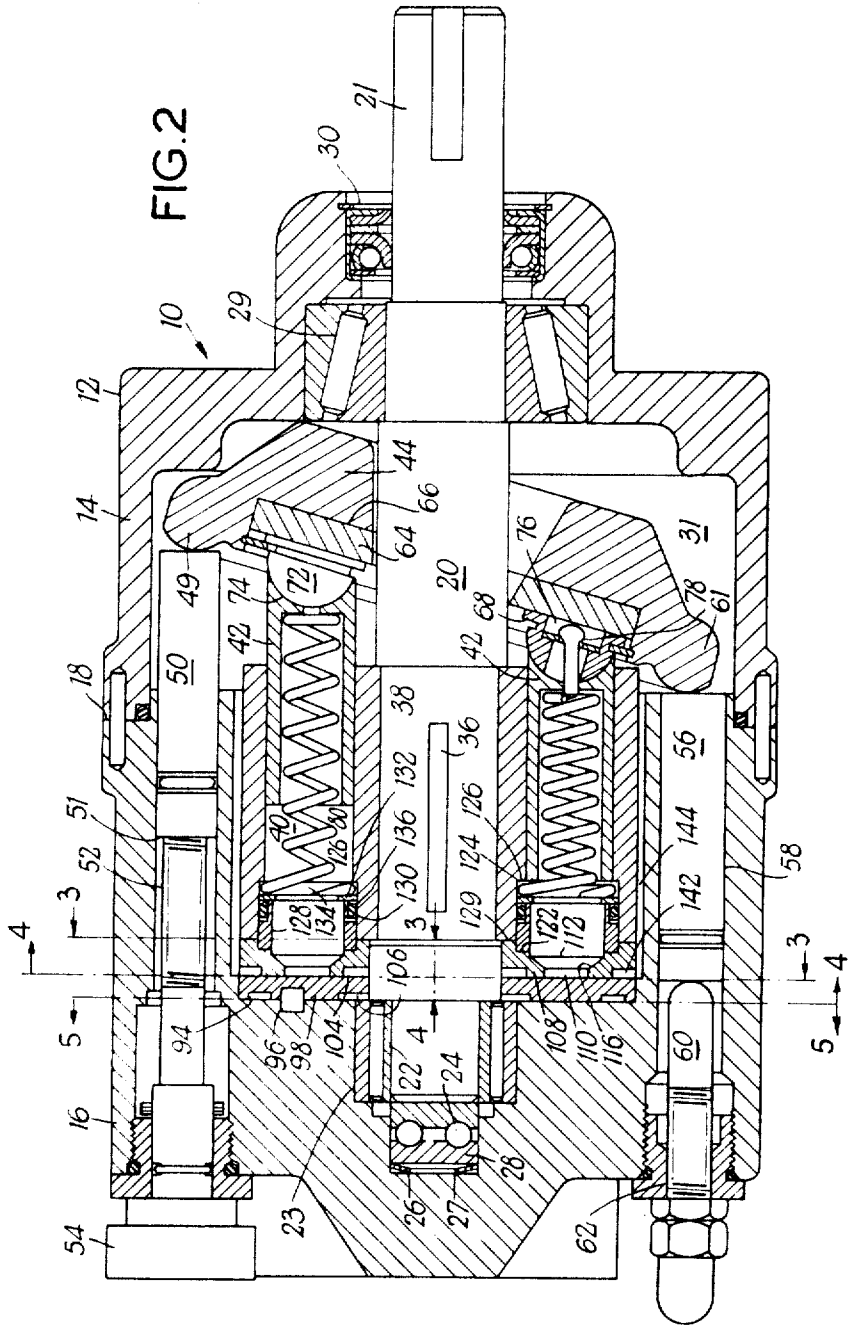
ATTORNEY

Jan. 14, 1969 M. A. D'AMATO Re. 26,519

VARIABLE DISPLACEMENT PUMP

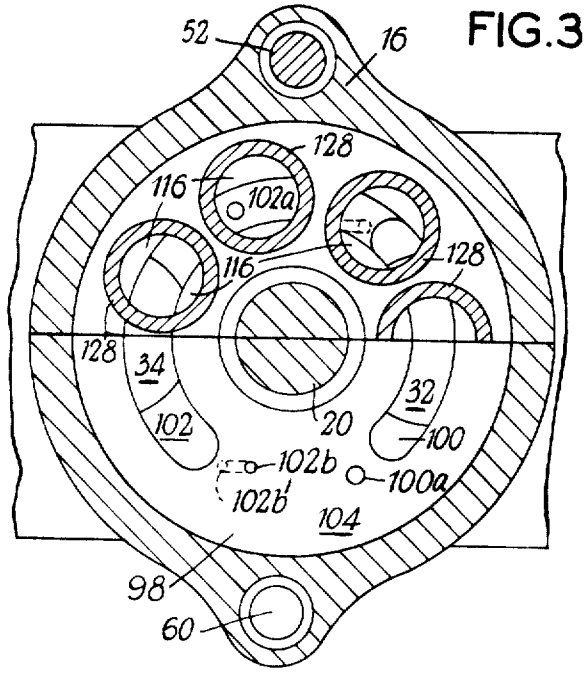
Original Filed Oct. 16, 1962 Sheet 2 of 4

FIG. 2



INVENTOR
MICHAEL A. D'AMATO
BY James H. Littlefield
ATTORNEY

FIG.3



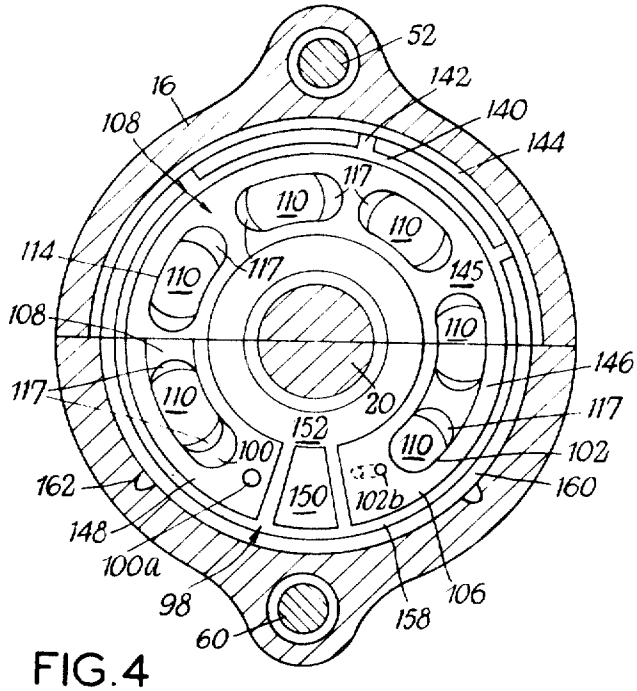


FIG.4

INVENTOR
MICHAEL A. D'AMATO
BY James Littlefield
ATTORNEY

FIG.5

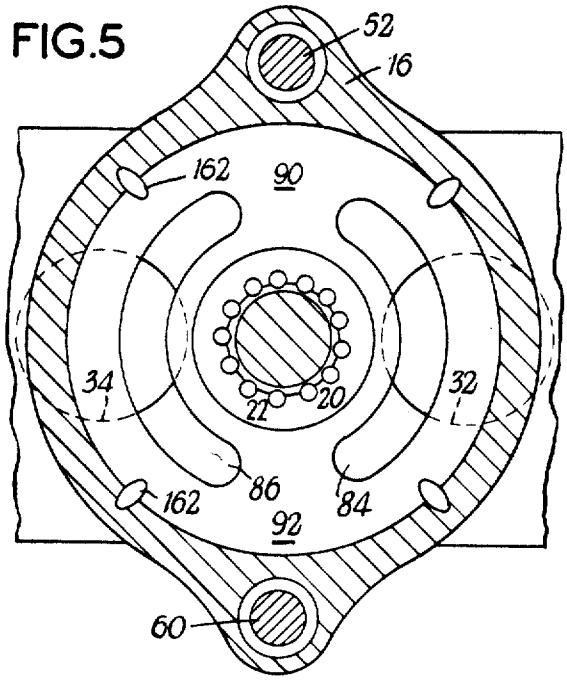


FIG.6

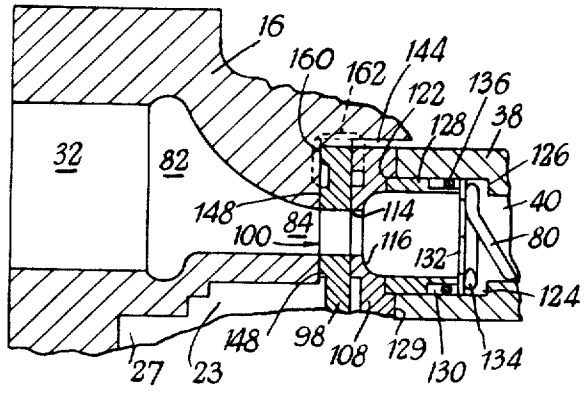
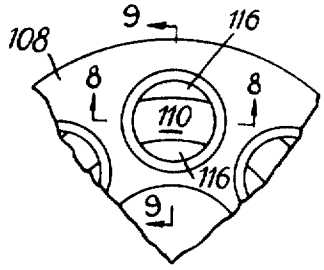


FIG.7



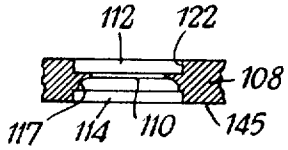


FIG.8

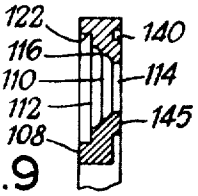


FIG.9

INVENTOR
MICHAEL A. D'AMATO

BY James H. Littlefield

ATTORNEY

United States Patent Office

Re. 26,519
Reissued Jan. 14, 1969

26,519

VARIABLE DISPLACEMENT PUMP

Michael A. D'Amato, 2905 Hillsdale Drive, Urbandale, Iowa 50322

Original No. 3,175,510, dated Mar. 30, 1965, Ser. No. 230,940, Oct. 16, 1962. Application for reissue Mar. 29, 1967, Ser. No. 637,016

U.S. Cl. 103—162 6 Claims

Int. Cl. F04b *1/02*

Matter enclosed in heavy brackets [ ] appears in the original patent but forms no part of this reissue specification; matter printed in italics indicates the additions made by reissue.

ABSTRACT OF THE DISCLOSURE

A variable displacement pump including a rotor mounted for rotation on a drive shaft which is journalled in the pump casing. Cylinders axially aligned in the rotor carry pistons which are spring loaded against canted swash plate. Arcuate intake and outlet ports in the pump casing communicate with the cylinders through cylindrical sleeves which are slideably received in counterbored recesses in the cylinders, the sleeves bearing against a wear plate which in turn bears against that casing surface which defines the arcuate ports.

This invention relates to variable displacement fluid pumps and motors and, more particularly, to multi-cylinder pumps of the type in which a plurality of pistons move axially in a rotor with variable stroke lengths, depending upon the angle of disposition of a swash plate.

In swash plate pumps, wherein the ends of cylinders in a rotor rotate successively over an arcuate intake port, a blank area, an outlet port, another blank area, and then again the intake port, there are fundamental problems, one of which is to obtain and maintain good sealing relationship between the blank surfaces at the end of the rotor, in which the cylinders are formed, and the blank areas or sealing surfaces of the ported member or plate against which the end of the rotor slides. Taking into account the requirement that the rotor must be mounted, usually on a drive shaft, on bearings in the pump casing, the tolerances, including bearing play and vibration under load necessary for economic manufacture of the parts generally result in relative separation or canting of the sliding surfaces, thereby opening the seal between them. For maximum efficiency, the sealing surfaces should be absolutely flat against one another, and should be no farther apart than to accommodate a very thin film of oil on which the rotor end slides. If the sealing surfaces separate, leakage and short-circuiting of the pumped fluid, usually oil, occurs, thereby reducing efficiency. Canting of the rotor's end surface tends to set up localized areas of wear, either on the rotor end or on the ported member against which it slides, and before long either the rotor or the ported member must be replaced. Another important object is provision for use of a material for the rotor with wear and heat expansion characteristics which are compatible with the axial pistons, and a bearing material for the rotating wear plate which has quite different characteristics. The object now is to provide a wear plate mounted on the end of a pump rotor for rotation therewith, and having significant, however limited freedom of axial movement relative to the rotor. A further object is to provide torque transmitting sleeves slidably fitting in the ends of the cylinders and the ends of cylinder ports in the wear plate, and arranged to provide seals between the cylinder walls and the wear plate. By this arrangement it is intended to provide for the manufacture of the member having the inlet and outlet ports of large cross section, and also the rotor, of durable metal such as steel, and the rotating port plate of metal having excellent slide-bearing characteristics against steel, such as bronze.

Still another object relates to the formation of the ports so as to reduce cavitation to the minimum. In order that the pistons shall undergo intake strokes and suck in the pumped fluid during nearly half of one rotation of the rotor, and then exhaust the pump fluid during nearly all of the other half rotation, it is desirable that the inlet and outlet ports occupy elongate arcuate zones, and that they be long enough so that the cylinder ends register therewith during substantially their entire intake and exhaust strokes. It is also desirable to maintain, to the extent possible, a one-to-one relationship between the cross-sectional area of the inlet and outlet ports and the inlet and outlet openings, as well as the transitional fluid connections therebetween. To accomplish this within the practical limits of design and manufacture, the inlet and outlet ports assume the form of elongate arcuate slots. The cylinders, however, are round so as to be easily bored, honed, and fitted by the pistons, and because of other design considerations, the diameters of the cylinders are greater than the radial widths of the port slots. An additional object is to utilize the rotating wear plate as a member for providing transitional port shapes and surfaces between the round cylinder ends and the relatively narrow elongate arcuate inlet and outlet ports. Furthermore, because of the separable sealing sleeves and rotating wear plate, it is possible to bore and hone the cylinders by straight-through machining whereas otherwise it would be necessary to work the tools into blind inner ends of the cylinders.

Still another object is to provide, at the inner end of a rotary multi-cylinder pump, fixed and rotating wear plates having sealing lands and bleeder grooves for bleeding a predetermined minor amount of leakage to the bearings at the inner end of the rotor shaft and to the low-pressure side of the pump. Another object is to provide a tapered roller bearing for the outer end of the pump shaft, and to pre-load the pump shaft by light spring pressure on its inner end, thereby holding the tapered bearing members firmly against one another when the pump is idling and therefore not producing the reactive forces which prevent the tapered bearing members from chattering when the pump is under load.

These and other objects will be apparent from the following specification and drawings, in which:

FIG. 1 is an isometric view of the pump, partly broken away;

FIG. 2 is a longitudinal cross section through the pump, the section being somewhat sector-shape through the rotor in order to show the extreme positions of two of the seven pistons which are not exactly diametrically opposed;

FIG. 3 is a transverse cross section along the line **3—3** of FIG. 2, looking in the direction of the arrows;

FIG. 4 is a transverse cross section along the line **4—4** of FIG. 2, looking in the direction of the arrows;

FIG. 5 is a transverse cross section along the line **5—5** of FIG. 4, looking in the direction of the arrows;

FIG. 6 is an enlarged fragmentary cross section, taken longitudinally through the inlet port, illustrating the sealing action of the parts;

FIG. 7 is a fragmentary plan view showing a cylinder port in the rotating wear plate; and

FIGS. 8 and 9 are fragmentary cross sections along the lines **8—8** and **9—9** of FIG. 7, showing the shape of the cylinder port.

Referring now to the drawings, in which like reference numerals denote similar elements, variable capacity pump **10** is enclosed within a casing **12** formed of two separable

parts **14** and **16**, respectively, connected by a joint **18** and held together by bolts **19**. Casing **12** is generally cylindrical and centrally mounted therein is a drive shaft **20** having an outwardly projecting free end **21** arranged to be connected to a suitable torque source. The inner end of the drive shaft is rotatably supported by a roller bearing assembly **22** engaging in an annular recess **23** in casing part **16**, the inner end of drive shaft **20** also engaging against an end thrust ball bearing assemby **24**. A compression spring **26** engaging between the inner end of a bearing recess **27** and ball bearing race **28** provides an axially outward thrust against shaft **20** so as to hold it firmly against tapered roller bearings **29**, thereby preventing chattering of the shaft and bearings when the pump is idling. A conventional seal **30** is provided for preventing leakage of the pump fluid outwardly from pump chamber **31**. As seen best in FIGS. 1 and 5, casing part **16** is provided with an inlet **32** on one side and a diametrically opposed outlet **34**.

Keyed, as at **36**, to shaft **20** is a cylinder barrel **38** having seven axial cylinders **40** equally spaced therearound and extending from end to end therethrough. Sliding in outer end portions of cylinders **40** are hollow pistons **42** which are reciprocated, when shaft **20** is rotated, by a swash plate **44**. As shown in FIGS. 1 and 2, plate **44** is adjustably supported by stub shafts **46** which project outwardly through bearings **48** on opposite sides of casing part **14**. At 90° from stub shafts **46** on swash plate **44** are a pair of ears **49** and **61**, ear **49** being engaged by plunger **50** which is slidable in a bore **51** adjacent one side of the casing, and which may be axially adjusted by a lead screw **52** turned by a hand wheel **54**. On the opposite side of the casing, a back-up piston **56** is axially slidable in a bore **58**, the back-up piston **56** engaging against a back-up adjustment rod **60** which is axially adjustable by a screw assembly **62**. Bore **58** is connected by a duct (not shown) to the high-pressure side of the pump so that, when the pump is under load, back-up piston **56** presses firmly against ear **61**, thereby exerting a force tending to hold the opposite ear **49** tightly against plunger **50**.

Swash plate **44** is faced with a reaction plate **64** engaging in an annular recess **66** on its inner face, the reaction plate **64** being slidably engaged by piston shoes **68** whose flat surfaces **70** slide thereagainst. Piston shoes **68** have semi-cylindrical convex surfaces **72** slidably engaging, like knuckles, against semi-cylindrical concave surfaces **74** on the outer ends of pistons **42**. It is noteworthy that piston shoes **68** are hollow, as at **76**, and communicate with the hollow interiors of pistons **42** so as to achieve a hydraulic near-balance between the outer ends of the piston shoes **68** and reaction plate **64**, the piston shoes thereby engaging against the reaction plate with but slightly more than enough pressure to effect a seal therebetween. Piston shoes **68** are loosely connected to the outer ends of pistons **42** by cotter pins **78** so that a free swiveling motion can occur between the piston shoes and the pi tons. Compression springs **80** within hollow pistons **42** provide the outward bias for the pistons.

When barrel **38** is turned by shaft **20**, those of the pistons **42** which pass around the side of the pump in which the inlet **32** is disposed move outwardly against the surface of reaction plate **64** on swash plate **44**, thereby drawing pump fluid into cylinders **40**, and those of the pistons which are then passing around the side of the pump in which outlet **34** is disposed are cammed inwardly, thereby expelling the pump fluid through the outlet. The volume of fluid drawn into and expelled from the pump, with each rotation of shaft **20**, depends upon the angle at which the swash plate **44** is disposed, it being obvious that when the surface of reaction plate **64** is normal to the axes of cylinders **40**, no pumping action occurs, and when swash plate **44** is tipped to an extreme position as in FIGS. 1 and 2, maximum delivery occurs. Since the pump fluid is drawn inwardly and expelled via openings through surfaces at the inner ends of cylinders **40**, and since these surfaces must slide against one another to achieve the necessary valving and porting, it is vital to avoid substantially leakage, since leakage, or poor sealing, may be due to necessary tolerances entailed in economic manufacture, or it may result from wear of the parts sliding against one another, or from axial movement of the rotor, or shaft vibration, it is evident that basic problems arise in the design of these parts which can be inexpensively machined but which nevertheless will maintain effective sealing against excessive leakage during the life of the pump. It is with these features that the subject invention is primarily concerned.

Referring particularly to the details illustrated in FIGS. 3 to 6, inclusive, inlet **32**, which is circular at its outer end, passes inwardly through a transition **82** to an arcuate inlet port **84**. It is desirable that the cross sectional area of transition **82** and inlet port **84** be similar, or nearly similar to the cross sectional area of inlet **32**. On the opposite side of the pump is a similar outlet port **86**, which likewise connects with outlet **34** via a transition **88**. Between the ends of the arcuate inlet and outlet ports **84** and **86** are blank areas **90**, **92**, the center of the area **90** representing the dead center inward position of the pistons and the center of the area **92** representing the dead center outward position of the pistons. Fitting within a counterbore **94** in part **16** of the casing, and maintained against rotation by a pin **96** is a fixed wear plate **98**, preferably of steel and having therethrough arcuate inlet and outlet slots **100**, **102** respectively registering over inlet and outlet ports **84** and **86**. The side **104** of fixed wear plate **98** disposed towards the cylinders is flat planar, whereas the side **106** disposed towards the bottom of counterbore **94** has lands and grooves for purposes described below.

Mounted upon and rotating with the inner end of barrel **38** is a rotating wear plate **108**, preferably of bronze or the like material having good bearing properties for sliding against steel, and having cylinder ports **110** registering with the inner ends of cylinders **40**. The ends **112** of cylinder ports **110**, which are disposed towards the cylinders **40** are circular, whereas the ends **114** disposed towards fixed wear plate **98** are arcuate, the cylindrical and arcuate ends being connected by transitional bevels **116**, **117**. The arcuate ends **114** of the cylinder ports register with the arcuate inlet and outlet slots **100**, **102** in the fixed wear plate but are much shorter, being but slightly less in angular extent than the blank areas between the ends of the inlet and outlet slots **100**, **102**, through the fixed wear plate **98**.

As shown in large detail in FIG. 6, rotating wear plate **108** is counterbored at the circular end **112** of each cylinder port **110** to provide a seat **122**. The inner ends of cylinders **40** are also counterbored to provide enlargements **124**, each terminating in a shoulder **126**, and a sleeve **128** slidably engages in both seat **122** and enlargement **124** and bridges the butt joint **129** between the inner end of barrel **38** and the adjacent side of wear plate **108**. The outer side of each sleeve **128** is turned down, near its outer end, to provide a recess **130**, and engaging over the outer end of each sleeve is a flat snap washer **132**. The inner ends **134** of piston springs **80** abut against snap washers **132**, and O-rings **136** are accommodated in recesses **130** to establish a fluid seal between the outer sides of sleeves **128** and the inner sides of cylinders **40**. It should be noted that the maximum diameters of sleeves **128** are somewhat greater than the ends of the pistons **42** which they face so that a differential sealing pressure is exerted against the outer ends of the sleeves.

Sleeves **128** serve several basic functions, the first being to transmit torque from barrel **38** to the rotating wear plate **108**. By forming the rotating wear plate and rotor separately, it is possible to utilize different metals, such as bronze for the rotating wear plate and steel for sleeves **128**, barrel **38** and fixed wear plate **98**. This also makes it possible to replace, if necessary, a single wearing member,

rather than one of the larger components, such as the barrel, after extensive use of the pump. Also, by forming the rotating wear plate **108** and barrel **38** separately, it is possible to create the desired complex shape for ports **110**, and it is also possible to hone cylinders **40** and enlargements **124** by straight through operations, rather than by working against blind inner ends of the cylinders. Most importantly, however, the slidability of sleeves **128** in seat **122** and enlargements **124** permit a slight freedom of motion between rotating wear plate **108** and barrel **38** so that the sealing surfaces on the inner side of the rotating wear plate can be maintained in flat sliding engagement against the flat planar surface **104** of fixed wear plate **98**, even though the barrel **38**, which is keyed to shaft **20**, may undergo slight axial or tipping movements as a result of shaft deflection or vibration resulting from the bearings. Rotating wear plate **108** has a slight but nevertheless significant freedom of universal movement with respect to shaft **20**, resulting from the minute play required for the sliding fit of sleeves **128** in enlargements **124** and seats **122**.

Referring now to the lower half of FIG. 3 and the upper half of FIG. 4, it will be observed that whereas the ends of ports **110** which face towards cylinders **40** are circular, the ends which face towards the flat side **104** of fixed wear plate **98** are arcuate and are chamfered at their ends to provide bevels **117** which form smooth transitions, in the annular direction of the port array, between the circular side and the arcuate side of the ports. As noted hereinbefore, the cylinder sides of ports **110** have beveled surfaces **116** which provide smooth transitions in the radial directions of the ports between their circular and arcuate sides. Surrounding and spaced radially outward from the inner sides of ports **110** is an annular groove **140** communicating with the periphery of rotating wear plate **108** via radial channels **142** for bleeding leakage to the portion **144** of the chamber in which barrel **38** is mounted, the chamber being bled to the low pressure side or intake port **34** via a duct (not shown). Surrounding the inner arcuate sides of ports **110** are flat sealing surfaces **145** which slide against the flat side **104** of fixed wear plate **98**.

Referring to the lower half of FIG. 4, the land and groove side **106** of fixed wear plate **98**, which engages against the flat plane surfaces of counterbore **94** (FIG. 5) is formed with flat segmental lands **146** and **148** which surround inlet and outlet slots **100**, **102**, respectively, and short segmental lands **150**, the lands **146**, **148** and **150** providing flat sealing surfaces. An inner annular groove **152** communicates with the recess **23** in which bearing **22** is housed, the inner annular groove communicating through radial grooves **156** with an outer annular groove **158** which surrounds lands **146**, **148** and **150**. A flat annular bearing land **160** surrounds outer annular groove **158**, the latter being vented to the pump chamber space **144** via bleeder notches **162** formed in the outer side and periphery of counterbore **94**. The bearings at the inner end of shaft **20** are thus lubricated by leakage fluid which eventually bleeds to the low pressure side of the pump.

A vitally important concept of the pump as a whole is the axial balance of forces while providing oil passageways to and from the cylinders the size of which are in excess of that which could be obtained in the conventional or common design and still maintain a hydraulically balanced or pressure loaded seal between the rotating and stationary plates at the port section of the pump. The above concept also provides two other very desirable auxiliary features: (1) the opportunity to use a ferrous metal in the cylinder rotor and a bearing metal in the rotating wear plate; and (2) the use of a tapered roller bearing for radial loading and axial orientation. The internal radial loading of this bearing is a result of the combined radial reaction of the pistons produced by the cam plate and the magnitude of this radial force is a function of the pressure being pumped. Because the permissable bearing radial load is proportional to the thrust load and the thrust load is also a function of the pressure being pumped, there results an ideal situation. The thrust or force against this bearing, neglecting that designed into it with the spring **26** for idling, is caused by the cylinder seal **128** being of greater cross sectional area than that of piston **42**. This differential area multiplied by the fluid pressure equals this thrust force.

The common design for this type of pump is to float the cylinder rotor on a shaft with axial freedom. The pressure of oil behind a piston then forces the piston to the cam plate on one end and by reducing the opening from the piston chamber to the outlet port on the other end of the cylinder rotor is hydraulically forced to the port end of the pump. The size of the passageways leading to and from the piston chamber contacting the port openings are very critical to pump operation. If they are larger in cross sectional area than the pistons are, then the integral type cylinder rotor and port plate would be forced forward breaking the seal between these faces and the pump becomes inoperative. If they are small to maintain a good seal, the oil is choked from entering and leaving the piston, which limits the amount of oil the pump can pump using atmospheric pressure to fill the piston chambers.

The subject pump cylinder rotor is rigid with the drive shaft. By using rotating wear plate **108** which is hydraulically sealed to a mechanically oriented, not rigid with the cylinder rotor by cylinder seals **128**, the disadvantages listed above for the conventional design are overcome. Oil passageways can be increased because the sealing forces between the port plate **98** and valve plate **108** is increased by the increase in size of the cylinder seal over that of the piston **42**. The cylinder rotor now is forced via the shaft into the tapered bearing by this differential force.

Another feature of fixed wear plate **98** is an arrangement which enhances greatly the quietness in the operation of the pump. To a large extent, noise in the pump is caused by the time rate of pressure change. Referring particularly to FIGS. 3, 4 and 5, it will be noted that the land space between the ends of arcuate outlet slots **100**, **102** is considerably greater than the blank areas **90**, **92** between the ends of the inlet and outlet ports **84**, **86**. Holes **100a** and **102a** are drilled through fixed wear plate **98** so as to locate the ends of slots **100**, **102** with accuracy far beyond that obtainable with ordinary casting practice (also permitting the casting of the plates for both left and right-hand drive pumps).

When the arcuate end **114** of a cylinder port **110** is midway between the ends of slots **100**, **102**, and assuming swash plate **44** to be tilted, then, if the lands between the ends of slots **100**, **102** were the same as the blank areas **90**, **92**, the cylinder port pressure would go from high to low pressure almost instantly. By using a more extensive land area between the ends of the inlet and outlet slots **100**, **102**, about 15° of rotation is required for a cylinder port **110** to go from the end of slot **102** to the start of slot **100**. Small holes **100b**, **102b**, whose opposite ends **102b'** are denoted in dotted lines, are drilled through the fixed wear plate which will permit enough transfer of oil to match the cylinder pre-compression on one side and the decompression on the other with that of the pressures existing within ports **100**, **102**, with resultant quiet operation of the pump.

Pumps constructed in accordance with the foregoing specification have high volumetric efficiencies high in the 90's while pumping pressures in excess of 3000 p.s.i. The unique porting design permits the minimum of internal oil velocity during maximum speeds, reducing inefficiency and excessive wear due to cavitation. Due to the extreme simplicity of the unit, the working parts are held to a minimum and those components subject to wear are self compensating to assure long, trouble-free performance under the most severe operating conditions.

The invention is not limited to the details illustrated and described herein, but is intended to cover all sub-

stitutions, modifications and equivalents within the scope of the following claims.

I claim:

1. In a fluid energy translating device, a casing defining a chamber having inner and outer ends, a port member affixed at the inner end of said chamber and having flat planar surface means facing the outer end thereof with inlet and outlet ports angularly spaced therein along arcs of a circle, a drive shaft extending from end to end through said chamber, bearing means rotatably supporting said drive shaft in said casing, a swash plate mounted in said chamber and spaced towards the outer end thereof from said port member, a cylinder barrel affixed on said shaft for movement therewith and having inner and outer ends respectively facing the port member and the swash plate, a plurality of cylinders angularly spaced about the rotative axis of the barrel and having inner and outer portions respectively disposed towards the port member and swash plate, pistons reciprocating in the outer end portions of said cylinder and having force transmitting means thereon engaging said swash plate for camming said pistons in one direction towards the inner end portions of said cylinders during one phase of rotation of said shaft, return stroke means for forcing said pistons in the opposite direction during another phase of rotation of said shaft, a rotating wear plate having an outer surface adjacent the inner end of said barrel and an inner surface slidingly engaging the flat planar surface means of said port member, said rotating wear plate having a plurality of passages therethrough having circular outer ends and inner ends respectively registering with said cylinders and said ports, [cylindrical seats in the outer surface of said wear plate respectively surrounding the outer ends of said passages,] *torque transfer means for coupling said wear plate for rotation with said barrel*, said barrel having right cylindrical enlargements of said cylinders at the inner end portions thereof, cylindrical sleeves having circular inner end and outer end portions slidably engaging [respectively in] said [cylindrical seats and] enlargements and constituting *both fluid seal means and a portion of the transitional fluid flow paths to and from the working chambers of said cylinders*, [combined fluid transmitting and fluid-seal means between said barrel and said rotating wear plate,] the inner diameters of the sleeves being substantially equal to *both* the diameters of the outer ends of said passages *and the diameters of said pistons, whereby the cross-sectional areas of said transitional flow paths are maintained substantially constant throughout their lengths to insure substantially uniform velocity of the fluid flowing in each of said paths at any instant.*

2. The combination claimed in claim **1**, a terminal part of the outer side of the outer end portion of each of said sleeves being of reduced diameter with respect to the diameter of the cylinder enlargement in which the remainder of said outer end portion slidably engages and defining, between the outer side of the sleeve and the wall of the enlargement a cylindrical pocket, and an O ring seal in said pocket, said seal surrounding said reduced diameter part of the sleeve and engaging the wall of the enlargement.

3. The combination claimed in claim **1**, *wherein said torque transfer means include cylindrical seats in the outer surface of said wear plate respectively surrounding the outer ends of said passages for slidably receiving the circular inner end portions of said cylindrical sleeves,*

the combined axial lengths of each of said seats and said enlargements being greater than the axial lengths of said sleeves whereby the outer ends of said sleeves are spaced from the ends of said enlargements, said return stroke means including expansion springs operatively engaging between said pistons and the outer ends of said sleeves.

4. The combination claimed in claim **1**, said bearing means comprising a radial thrust bearing supported in said casing at the inner end of the chamber, a tapered roller bearing supported in said casing at the outer end of said chamber, said tapered roller bearing having bearing members inclined to take shaft thrust in both the radial direction and those axially resultant from thrust against pumped fluid, and spring means operatively engaging between said casing and shaft for exerting an axial thrust thereon in the same direction as said axially resultant forces, whereby to pre-load said radial bearing when said pump idles.

5. The combination claimed in claim **1**, the inner ends of said passages being elongate along arcs of the same circles as the inlet and outlet ports and of substantially equal radial extent as the inlet and outlet ports, the inner ends of said passages being substantially equal in cross-sectional area to the outer ends thereof and said passages being of substantially the same cross-sectional area from end to end.

6. The combination claimed in claim **5**, said inner ends of said passages having arcuate lengths greater than the diameter of the outer ends and of less radial width than said inner ends, there being bevels constituting transitional surface between the inner and outer ends of the passages, thereby providing gradual emergence from the shape of the passage inner ends to the passage outer ends, whereby to reduce cavitation of liquid flowing at high velocity therethrough.

References Cited

The following references, cited by the Examiner, are of record in the patented file of this patent or the original patent.

UNITED STATES PATENTS

2,161,153	6/1939	Doe et al.	103—162
2,331,694	10/1943	Jeffrey	103—162
2,649,741	8/1953	Henrichsen	103—162
2,844,104	7/1958	Wennberg	103—162
2,953,099	9/1960	Budzich	103—162
2,992,619	7/1961	Nilges	103—162
3,191,543	6/1965	Hann et al.	103—162
2,365,067	12/1944	Gauld	103—162
2,619,041	11/1952	Born	103—162
2,646,754	7/1953	Overbeke	103—161
2,661,695	12/1953	Ferris	103—162 X
2,963,983	12/1960	Wiggerman	103—162
2,972,962	2/1961	Douglas	103—162
3,124,079	3/1964	Boyer	103—162

FOREIGN PATENTS

1,233,394	5/1960	France.
822,692	10/1959	Great Britain.
348,322	9/1960	Switzerland.
504,144	7/1930	Germany.
1,020,525	12/1957	Germany.
822,014	10/1959	Great Britain.

WILLIAM L. FREEH, *Primary Examiner.*