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Schottler

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[54] **VARIABLE DISPLACEMENT FLUID TRANSDUCER**

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[52] **U.S. Cl.** **91/492**

[51] **Int. Cl.** **F01b 13/06**

[58] **Field of Search** 91/472, 492; 417/215, 486

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Attorney, Agent, or Firm—Melvin F. Jager

[57] **ABSTRACT**

A variable displacement fluid transducer including a plurality of radial pistons arranged in pairs on a rotating piston barrel. Cam tracks circumscribe the pistons and convert the reciprocation of said pistons into rotary motion of a shaft means, or vice-versa. Piston chambers provided in the barrel couple the pairs of pistons, and the cam tracks are adjustable to vary the displacement of the transducer by changing the relative reciprocating movement of the pistons comprising each pair.

5 Claims, 7 Drawing Figures

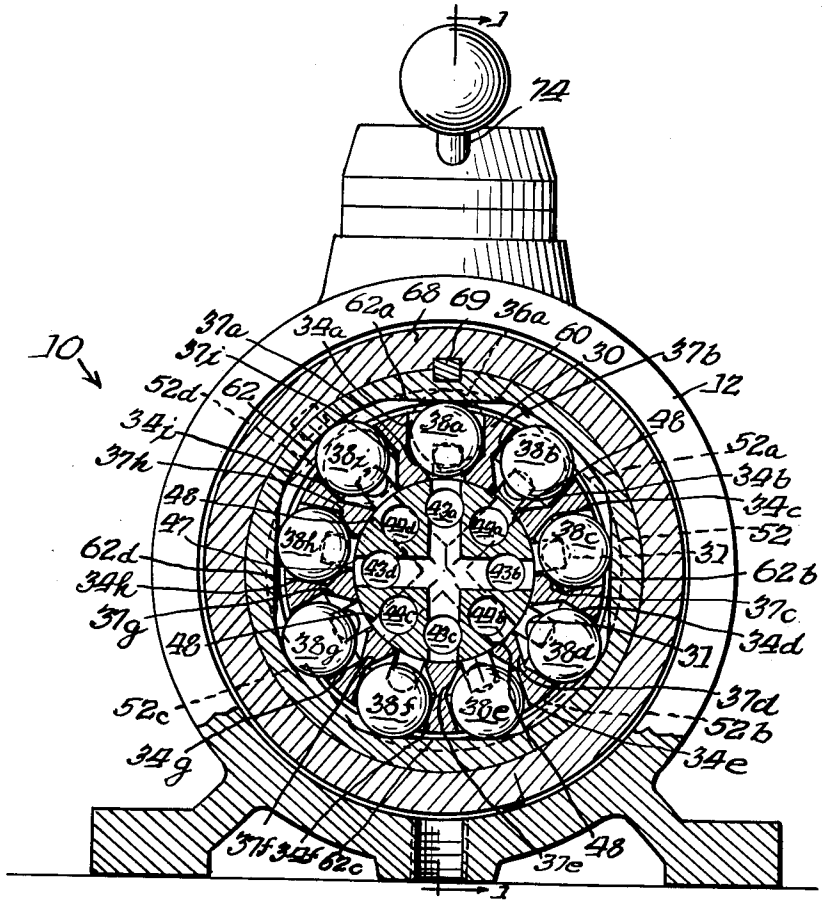

Fig. 1.

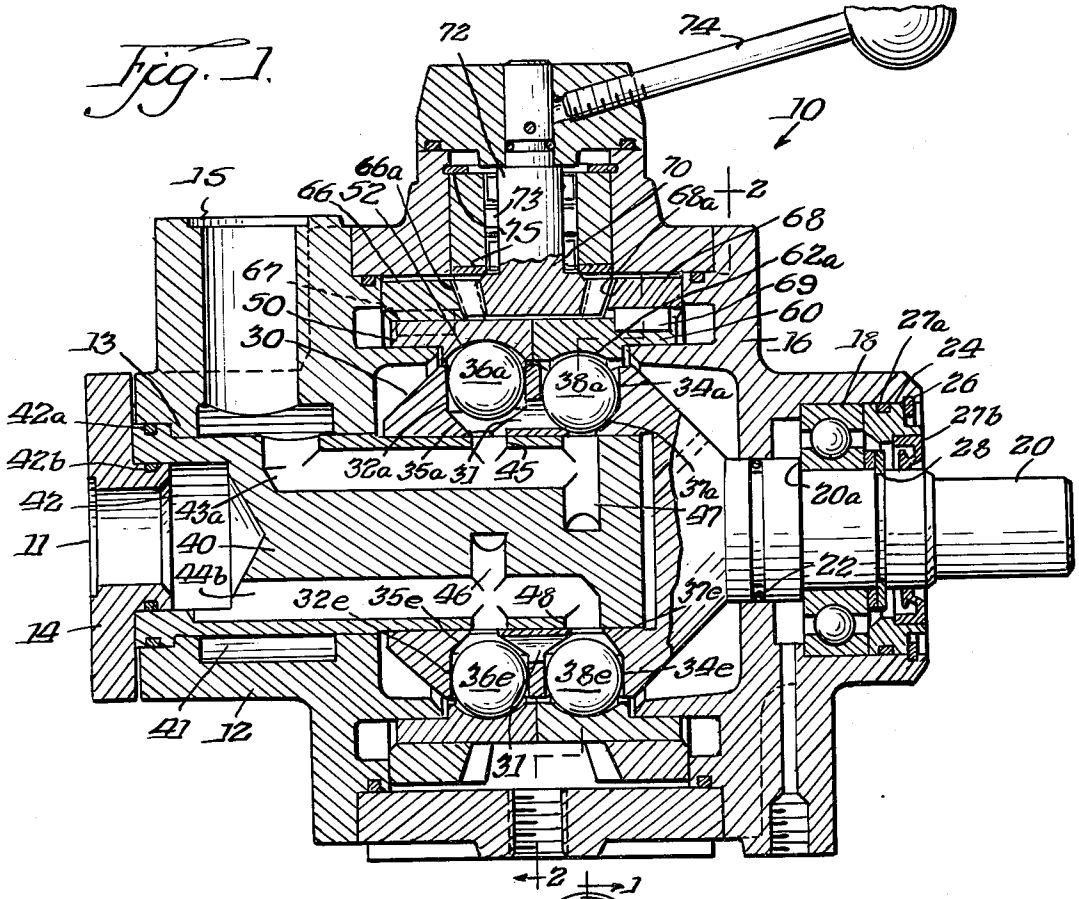
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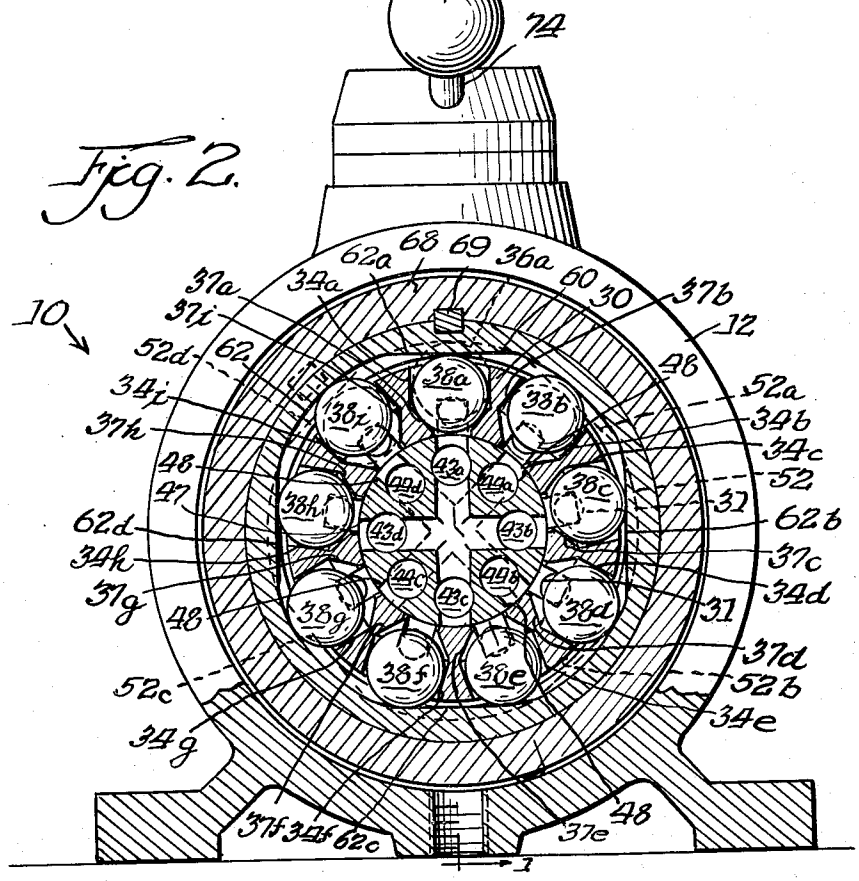
Fig. 2.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38g
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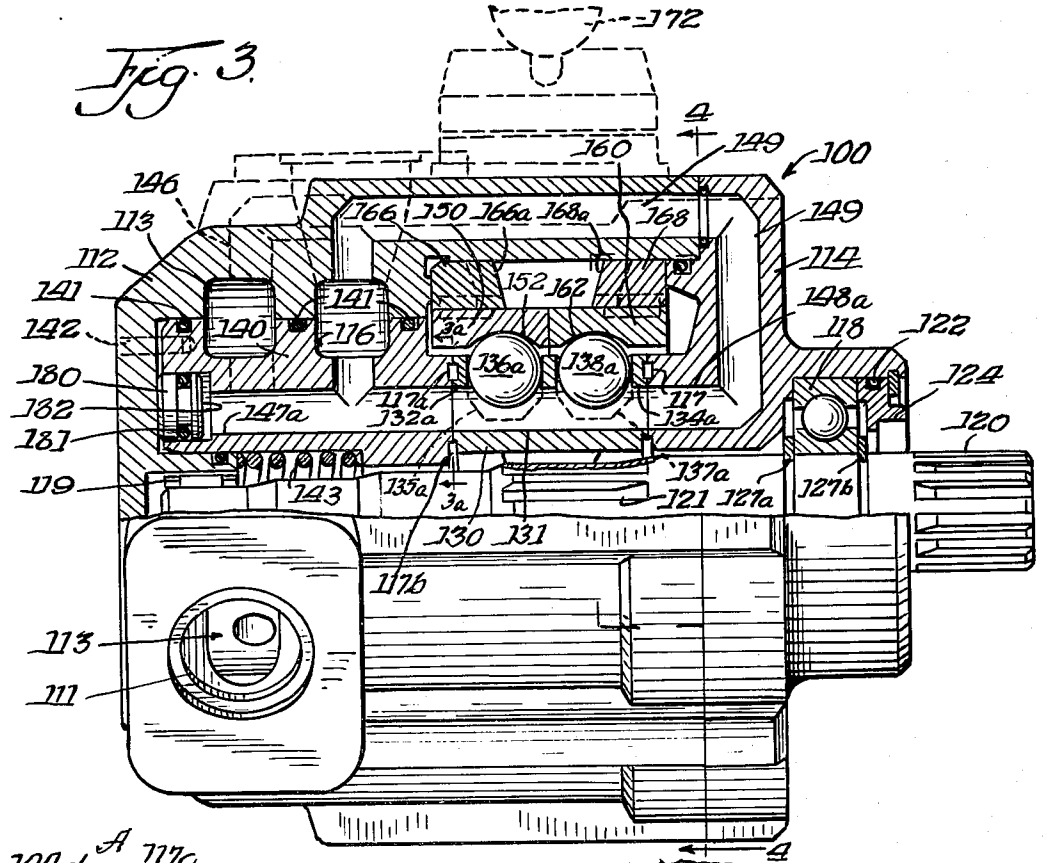
Fig. 3.



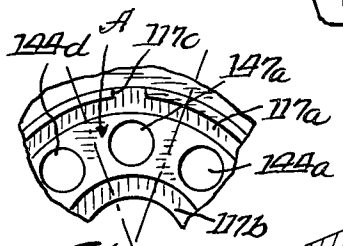
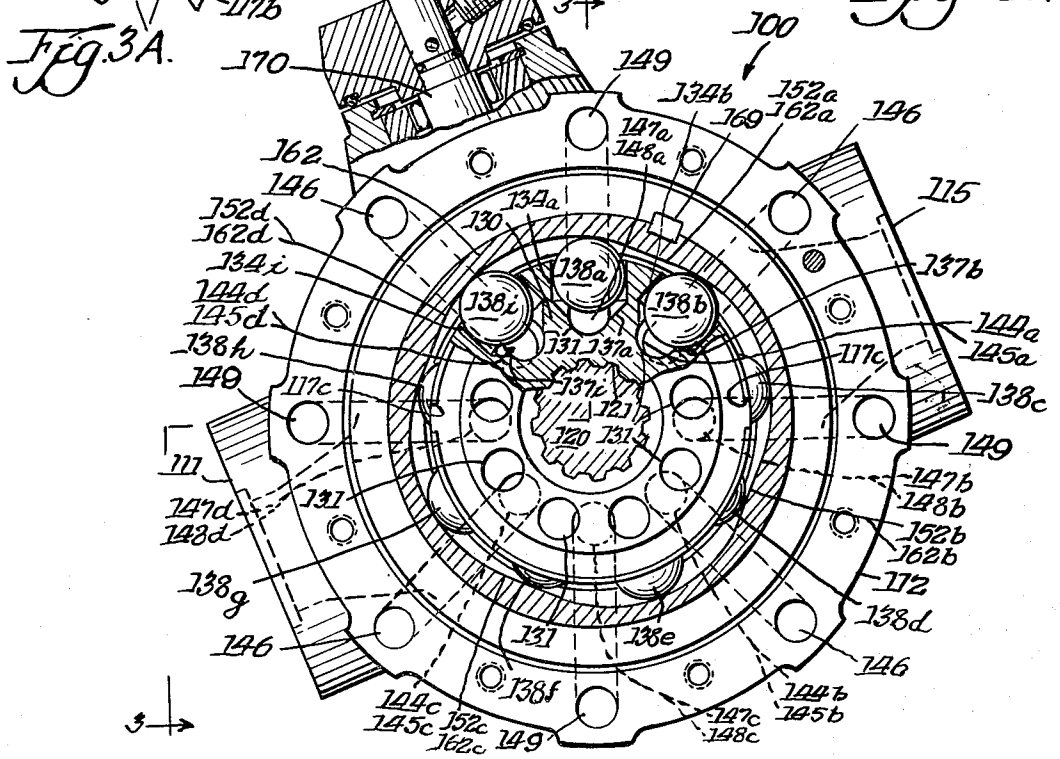


Fig. 3A.

Fig. 4.



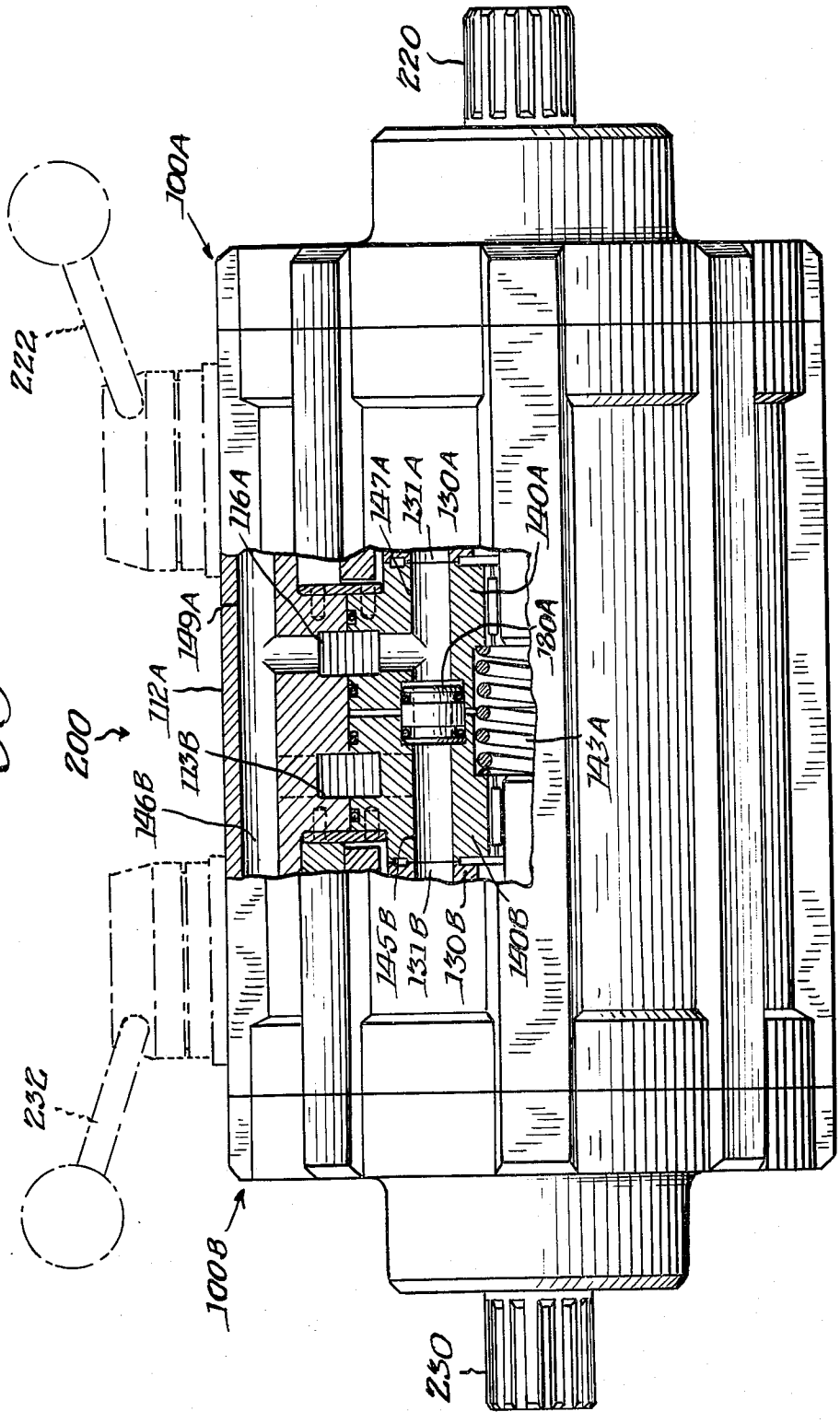

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Fig. 5.

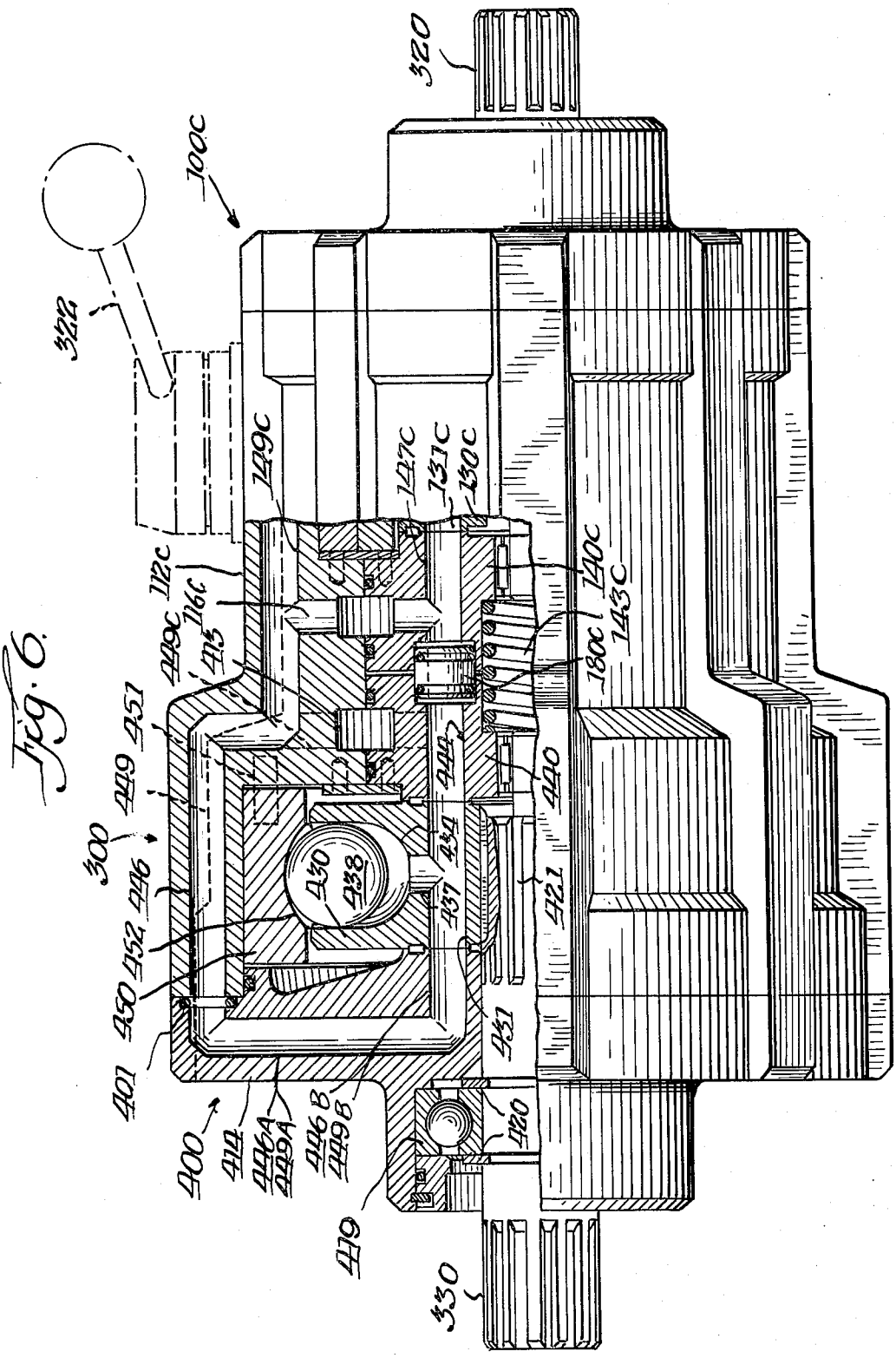
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Fig. 6.

VARIABLE DISPLACEMENT FLUID TRANSDUCER

BACKGROUND AND GENERAL DESCRIPTION

This invention relates generally to a variable displacement fluid transducer, which can be operated as a pump, motor or transmission, for imparting energy to or extracting energy from a fluid under pressure.

There is a demand for well-designed transducer units, for use as motors, pumps, transmissions and the like, which are lightweight, compact, and efficient to manufacture and operate. There is also a need for a transducer that is capable of producing an infinitely variable fluid displacement over a broad operating range. The transducer at any selected displacement within a given range also should operate with uniform and continuous fluid flow, with minimum fluid losses. Another important feature of an effective fluid transducer is the capability of being operated in a reverse direction without substantially effecting the efficiency and fluid displacement characteristics of the unit.

The purpose of this invention is to provide a fluid transducer which meets the above-described design parameters. The transducer in accordance with this invention can be utilized as a motor, pump or hydraulic transmission. The transducer is compact in design and efficient in operation. Moreover, the present transducer operates with a fluid displacement that can be infinitely varied, in either a positive or a negative direction from a zero flow condition, within a selected displacement range. The transducer therefore is sufficiently versatile to be used in applications which call for varying fluid displacement, speed and torque output requirements.

Briefly described, the transducer in accordance with the present invention includes rotatable shaft means and a cylindrical piston barrel joined for rotation with the shaft means. Sets of radial pistons comprising a plurality of piston balls are positioned uniformly spaced around the barrel. The pistons are arranged to define coupled pairs of piston balls and a piston chamber is provided in the barrel for each piston pair. In operation, the relative reciprocation of the piston balls comprising each pair displaces fluid from the associated piston chamber. Cam track means circumscribe the pistons and include a plurality of uniformly spaced lobes projecting inwardly toward the piston barrel. The cam lobes are in frictional engagement with the piston balls so that relative rotation of the barrel with respect to the cam track means orbits the balls in rolling engagement with the cam lobes. The piston balls and lobes thereby cooperate to reciprocate the pistons radially and to displace fluid from the piston chambers as a function of the relative reciprocating movement of the piston balls comprising each piston pair. Control means are provided to shift the cam track means circumferentially, to vary the relative position of the cam lobes and thereby vary the fluid displacement of the transducer by changing the relative reciprocation of the piston balls comprising each pair. Porting means are further provided for directing fluid sequentially to and from each of the piston chambers during the operation of the transducer.

EXEMPLARY EMBODIMENTS

Further objects and features of the present invention will become more apparent from a description of several preferred embodiments thereof, taken in conjunction with the accompanying drawings, in which:

FIG. 1 is a cross-sectional elevational view of one embodiment of variable displacement piston pump having the features and advantages of the present invention;

FIG. 2 is a cross-sectional end view of the pump, taken along the line 2—2 in FIG. 1;

FIG. 3 is a cross-sectional elevational view of a second embodiment of the variable displacement piston pump in accordance with this invention;

FIG. 3A is a removed cross-sectional view of the second pump embodiment, taken along the line 3*a*—3*a* in FIG. 3;

FIG. 4 is a cross-sectional end view of the second embodiment, taken along the line 4—4 in FIG. 3;

FIG. 5 is an elevational view, in partial cross-section, of a fluid transmission system embodying the features and advantages of the present invention; and

FIG. 6 is an elevational view, in partial cross-section, of a second embodiment of a fluid transmission embodying the features and advantages of the present invention.

PUMP

The first embodiment of the pump in accordance with this invention is generally indicated by the reference number **10** in FIGS. **1** and **2.** The pump **10** includes a generally cylindrical pump housing **12** having removable end cover plates **14** and **16.** A bearing **18** mounted in the end plate **16** supports an input shaft **20.** A high-pressure sealing ring **22** is provided in a groove in the shaft **20** to seal the interior of the housing **12** from the compartment containing the bearing **18.** This arrangement adapts the pump **10** to high pressure applications by preventing any oil loss from the housing **12** around the shaft **20.** The ring **22** also prevents contamination of the fluid being pumped by leakage from the lubricant applied to the bearing **18.**

The bearing **18** is held in a stationary position in the end plate **16** by means of a sealing and retaining assembly **24.** A retaining ring **26** in the assembly **24** locks the assembly in the plate **16.** Suitable sealing rings **27***a* and **27***b* seal the compartment for the bearing **18** from the atmosphere. The input shaft **20** is restrained from axial movement by a shoulder **20***a* on the shaft, which abuts against one side of the bearing **18,** and a retaining ring **28,** which abuts against the opposite side of the bearing.

As seen in FIG. **1,** the inner end of the input shaft **20** defines a hollow cylindrical piston barrel **30.** For ease in machining and assembly, the piston barrel **30** is preferably an integral part of the shaft **20.** In the illustrated embodiment, the piston barrel **30** includes two sets of radial piston cylinders **32** and **34.** Each set of cylinders **32** and **34** includes a plurality of cylinders uniformly spaced around the circumference of the barrel **30.** In the preferred arrangement, the set **32** includes nine radial cylinders **32***a–i*, and the set **34** includes nine radial cylinders **34***a–i.* The set **32** and **34** are also arranged so that the cylinders **32***a–i* and **34***a–i* are uniformly spaced in axial alignment, as seen in FIG. **2.** Moreover, a plurality of axial bores in the barrel **30** define piston chambers **31** which join the cylinders in the set **32** with the adjacent aligned cylinder in the set **34.** As seen in FIG. **1,** the adjacent cylinders, such as cylinders **32***a* and **34***a*, will thereby cooperate to impart energy to a

charge of fluid in the piston chambers **31** during the operation of the pump **10.**

In accordance with this invention, each cylinder in the sets **32** and **34** includes a ball piston which is designed to reciprocate as the input shaft **20** is rotated. Accordingly, a set of piston balls **36***a–i* is positioned within the cylinders **32***a–i*, and a second set of piston balls **38***a–i* is positioned within the cylinders **34***a–i*. In the preferred embodiment, all ball pistons **36** and **38** are the same diameter, and the diameter is selected so that the ball pistons seal the associated cylinders without the need for any piston rings or the like. In addition, the inner ends of the cylinders **32***a–i* and **34***a–i* define fluid ports **35***a–i* and **37***a–i*, respectively. The ports **35** and **37** provide openings for the flow of fluid to and from the cylinders **32** and **34** during the operation of the pump **10.** The design of the ports **35** and **37** further arranges the ball pistons **36** and **38** so that a portion of each ball piston extends radially outward from the periphery of the piston barrel **30** during the operation of the pump **10.**

The embodiment of the pump **10** illustrated in FIGS. **1** and **2** includes a stationary cylindrical pintle **40.** The pintle is press-fit within a flange **13** provided on the left end of the housing **12.** Suitable sealing rings **42***a* and **42***b* seal the juncture between the pintle and the left end cover **14.** The pintle **40** is dimensioned and machined to fit, with a close tolerance, within hollow central core of the piston barrel **30.** The barrel **30** thereby rotates around the pintle **40** during the operation of the pump **10.** In the illustrated embodiment, the fluid being pumped through the pump **10** is used to lubricate the engaged frictional contact surfaces of the barrel **30** and pintle **40** by squeeze lubrication which is induced by the rotation of the barrel **30** and the radial forces on the piston balls **36** and **38,** as described further below.

The pintle **40** includes a series of bores which define inlet and outlet passages for controlling the flow of fluid through the pump **10.** The left end cover **14** defines an inlet bore **11** which communicates with a fluid collector chamber **42** provided in the adjacent end of the pintle **40.** A plurality of axial inlet channels **44** extend through the pintle **40** to a location adjacent the cylinders **32** and **34** in the piston barrel **30.** In the arrangement illustrated in FIGS. **1** and **2,** the pintle **40** is provided with four uniformly spaced inlet channels **44***a–d* in communication with the fluid chamber **42.** As seen in FIG. **1,** four uniformly spaced radial inlet bores **46** extend through the pintle **40** and connect all four inlet channels **44***a–d* in fluid communication. The bores **46** are axially aligned with the first set of cylinders **32,** and place the cylinders **32***a–i* in successive communication with the inlet channels **44***a–d* as the barrel **30** rotates. As seen in FIGS. **1** and **2,** a second set of four radial inlet bores **48** is axially aligned with the second set of cylinders **34.** The radial bores **48** are in axial alignment with the bores **46,** and connect the cylinders **34***a–i* to the four inlet channesl **44***a–d* as the barrel **30** rotates.

The pump **10** also defines a fluid outlet bore **15** which communicates with an annular fluid collector chamber **41.** The pintle **40** also includes four uniformly spaced axial outlet channels **43***a–d* which are spaced uniformly between the inlet channels **44***a–d*. A set of four radial outlet bores **45** are aligned axially with the first set of cylinders **32.** The bores **45** are also spaced uniformly in the pintle **40,** and the cylinders **32***a–i* in successive fluid communication with the outlet channels **43***a–d* as the barrel **30** rotates. An additional set of radial outlet bores **47** is aligned with the second set of cylinders **34,** and places the cylinders **34***a–i* in successive communication with the outlet channels **43***a–d* as the barrel **30** rotates. The outlet bores **45** and **47** are axially aligned in the pintle **40.**

The pump **10** also includes a pair of annular cam members **50** and **60** which control the reciprocation of the ball pistons **36** and **38,** respectively, during the operation of the pump **10.** As seen in FIG. **1,** the cam members **50** and **60** surround the piston barrel **30** and are arranged to space the cam member **50** radially above the set of piston balls **36***a–i* and to space the cam member **60** radially above the set of piston balls **38***a–i*. The internal periphery of the cam members **50** and **60** define concave cam tracks **52** and **62,** which engage with the piston balls **36** and **38** respectively. The tracks **52** and **62** are provided with a radius of curvature slightly larger than the radius of the engaged balls **36** and **38** so that the balls will be self-seating within the cam tracks during the operation of the pump **10.** The tracks **52** and **62** further define a plurality of uniformly spaced cam lobes which cause the engaged piston balls **36** or **38** to reciprocate, as the piston barrel **30** is rotated. In the preferred arrangement the cam track **52** is provided with four uniformly spaced lobes **52***a–d*, and the track **62** is provided with four uniformly spaced lobes **62***a–d*. The lobes are designed to have a uniform amplitude and wave length, and in the preferred embodiment, are contoured to produce a cycloidal displacement of the pistons **36** and **38.** This arrangement produces a substantially uniform total fluid flow over the entire range of displacement for the pump **10.**

The cam members **50** and **60** are held in a relatively stationary position during the operation of the pump **10.** To accomplish this, the pump housing **12** includes a pair of rotatable ring gears **66** and **68** spaced around the cam members **50** and **60,** respectively. The ring gears **66** and **68** are restrained from lateral movement by the housing **12.** The inner periphery of the gears **66** and **68** include splines or keys **67** and **69,** respectively, which mate with corresponding splines or key slots provided on the adjacent outer portion of the cam members **50** and **60.** The splines or keys **67** and **69** permit the cam members **50** and **60** to move axially within the housing **12,** and further cause the cam members **50** and **60** to rotate in response to the rotation of the associated ring gears **66** and **68.**

The pump **10** also includes means to gradiently vary the displacement of the pump by adjusting the circumferential positioning of the cam members **50** and **60.** The adjustment means includes bevelled gear teeth **66***a* and **68***a* provided on the inner portion of the ring gears **66** and **68.** The teeth **66***a* and **68***a* mesh with a bevel pinion **70** mounted on the pump housing **12** by means of a shaft **72.** Needle bearings **73** support the shaft **72** in the housing **12** and are selected to have a carrying capacity sufficient to absorb the reaction torque load applied to the shaft **72** during the operation of the pump **10.** Suitable sealing rings **75** are provided around the shaft **72** to maintain the housing **12** fluid-tight. A control handle **74** projects from the housing **12** and is connected to the shaft **72.** As seen in FIG. **1,** manual rotation of the handle **74** rotates the pinion **70,** and thereby adjusts the circumferential positioning of the ring gears **66** and **68** and the connected cam members

50 and 60. In the illustrated embodiment, movement of the pinion 70 shifts the cam members 50 and 60 an equal distance in opposite directions, and varies the relative axial positioning of the cam lobes 52*a–d* with respect to the cam lobes 62*a–d*.

In accordance with this invention, the cam members 50 and 60 are positioned within the housing 12, in a selected relationship with respect to the pintle 40, to provide the pump 10 with an infinitely variable fluid displacement within a broad range in either a positive or negative direction from a zero displacement condition. Thus, the cam members 50 and 60 initially are arranged so that the lobes 52*a–d* are 45° out of phase with respect to the lobes 62*a–d*. That arrangement places the bottom center-point of the lobes 52*a–d* in axial alignment with the top center-point of the lobes 62*a–d*, and vice versa, as seen in FIG. 2. Furthermore, the cam members 50 and 60 are circumferentially arranged in the housing 12 so that maximum port opening for the pump occurs when one of the piston balls 36 or 38 is engaged with the top center-point of the cam lobes 52*a–d* or 62*a–d*, and the other axially aligned piston ball 38 or 36 is engaged with the bottom center-point of the other cam lobe 62*a–d* or 52*a–d*. This arrangement defines the zero displacement condition for the pump 10. In the arrangement illustrated in FIG. 2, this zero condition occurs when the piston ball 38*a* is engaged with the top center-point of the cam lobe 62*a*, and the axially aligned piston ball 36*a* is engaged with the bottom center-point between the cam lobes 52*a* and 52*d*. At this point in the operation of the pump 10, the balls 36*a*, 38*a* and the ports 35*a* and 37*a* are radially aligned to the maximum open port condition, with the radial outlet bores 45 and 47 in the stationary pintle 40. The remaining fluid ports 35*b–i* and 37*b–i* are at varying positions with respect to the inlet and outlet bores 45–48 in the stationary pintle 40, and correspondingly produce varying conditions of port opening.

All axially aligned piston balls, i.e., 36*a*–38*a*, 36*b*–38*b*, etc. are engaged with either the top or bottom center-point of the cam lobes 52*a–d* and 62*a–d* under the zero displacement condition described above. Since each pair of axially aligned ball pistons (e.g. 36*a* and 38*a*) are reciprocating in opposite directions under that condition, the combined pumping force transmitted by each piston pair to the charge of fluid contained within the piston chamber 31 and the associated cylinders (e.g. 32*a* and 34*a*) is zero. Upon rotation of the barrel 30, the fluid in the piston chambers 31 under those conditions will be moved only between the associated cylinders 32 and 34, and the resultant fluid displacement of the pump 10 will be zero.

The operating handle 74 is rotated to increase the pump displacement from the above described zero displacement condition to a maximum displacement condition in either direction of fluid flow. For instance, the handle 74 and the pinion 70 can be rotated to rotate the cam member 50 counter-clockwise and the cam member 60 clockwise, as viewed in FIG. 2. The cam lobes 52*a–d* are thereby axially displaced with respect to the lobes 62*a–d*. The rotation of the piston barrel 30 under such conditions will reciprocate the pistons 36 and 38 in a manner which imparts a pumping force to the fluid charge within the piston chambers 31, or imparts a suction force which draws the fluid charge into the chambers 31, depending on the arrangement of the associated porting. The maximum pumping and suction forces, and hence the maximum fluid displacement for the pump 10 for fluid flow in the positive direction, will occur when the cam lobes 52*a–d* are axially aligned with the cam lobes 62*a–d*. This maximum condition is created by rotation of the cam members 50 and 60 with respect to each other a resultant of 45 degrees. Since each member rotates, the circumferential movement of each member 50 and 60 to produce such a condition will be 22½°. In such a position, the aligned piston balls 36 and 38 (e.g. 36*a* and 38*a*) are reciprocating in unison and hence produce the maximum pressure flow or suction flow of the fluid in the cylinders 32, 34 and the piston chambers 31. The cam members 50 and 60 can be positioned between the above-described zero and maximum condition to infinitely vary the fluid displacement of the pump 10.

In accordance with this invention, the cam members 50 and 60 can be shifted beyond the zero condition counter-clockwise, in the direction opposite to that described above, to reverse the flow of fluid in the pump 10. The fluid flow is reversed because the sequential relationship between the barrel ports 35, 37 and the inlet ports 46, 48 and the outlet 45 and 47 is reversed. Hence, the ports 46, 48 become outlet ports, and the ports 45, 47 become inlet ports. The pump operates in a similar manner to provide an infinitely variable fluid displacement between the zero flow condition and the maximum flow condition. The zero flow condition again occurs with the pump arranged as shown in FIG. 2. The maximum reverse or negative flow condition would occur when the cam members 50 and 60 are shifted, a resultant 45° (22½° each), to align the cam lobes axially; with the lobe 62*a* aligned with the lobe 52*d*; 62*b* aligned with 52*c*; 62*c* aligned with 52*b*; and 62*d* aligned with 52*a*.

The symmetry of the pump 10 in accordance with this invention permits the pump to operate with equal effectiveness for either clockwise or counter-clockwise rotation of the input shaft 20. The operation of the pump 10 will therefore be described only with respect to the results obtained by rotating the shaft 20 in a clockwise direction, as viewed in FIG. 2.

The clockwise rotation of the shaft 20 causes clockwise rotation of the piston barrel 30 at the same speed as the shaft 20. The rotating barrel 30 drives the piston balls 36 and 38 in a planetary fashion around the stationary pintle member 40. The piston balls 36 and 38 also roll along the annular cam tracks 52 and 62, and the cam lobes 52*a–d* and 62*a–d* impart a radial reciprocating motion to the piston balls 36 and 38. The profile of the lobes 52*a–d* and 62*a–d* causes the balls 36 and 38 to follow a cycloidal displacement path.

The pumping effect of the reciprocation of the balls 36, 38 is a function of the relative positioning of the cam members 50 and 60. When the cam members 50 and 60 are aligned in the zero position shown in FIG. 2, the reciprocation of the piston balls 36*a–i* will oppose the reciprocation of the piston balls 38*a–i* (e.g. ball 36*a* will be moving radially outward as the associated ball 38*a* is moving radially inward.) Since the relative reciprocation of the balls comprising each pair of pistons is zero, the fluid in the common chambers 31 will merely flow between the associated cylinders 32 and 34 under such a condition, and the fluid displacement of the pump 10 will be zero.

Fluid displacement will occur in the pump 10 when the handle 74 is operated to move the cam members 50

and **60** out of the above-described zero position. Maximum positive displacement will occur, for instance, when the handle **74** is rotated to displace the cam members **50** and **60** a resultant of 45° from the aligned zero condition. Since both cam members rotate equally, but in opposite directions, upon rotation of the handle **74**, maximum positive displacement will occur in the illustrated embodiment when the cam member **60** is rotated 22½° counter-clockwise from the zero displacement condition illustrated in FIG. **2**. In such a position, the top center-point of the cam lobe **62***a* and the top center-point of the cam lobe **52***a* will be in axial alignment, at a location 22½° from vertical in FIG. **2**. The remaining cam lobes **62***b*–**52***b*, 62*c*–**52***c*, and **62***d*–**52***d* also will be in axial alignment. The rotation of the piston barrel **30** will thereby cause the piston balls **36** and **38** to reciprocate in unison under such conditions, and each pair of piston balls will provide maximum pumping displacement of the fluid in the associated piston chamber **31** and cylinders **35** and **37**.

The operation of the pump **10** under the maximum positive displacement condition is apparent when the pump **10** is visualized as being shifted from the zero displacement condition as shown in FIG. **2**, into the maximum displacement condition described above. As the piston barrel **30** is rotated at a constant speed in a clockwise direction, each piston ball **36***a–i* and **38***a–i* roll along the cam members **50** and **60** respectively. The lobes **52***a–d* cause the pistons **36***a–i* to reciprocate radially in the cylinders **32***a–i*, and the lobes **62***a–i* cause the pistons **38***a–i* to reciprocate radially in the cylinders **34***a–i*. As the barrel **30** rotates, the pistons **36***a–i* and **38***a–i* will be moving radially outward when the associated piston ports **35***a–i* and **37***a–i* are in fluid communication with the inlet channels **44***a–d* through the radial inlet bores **46** and **48**. Each outwardly moving piston **36***a–i* and **38***a–i* is thereby providing a suction stroke which draws fluid into the common chambers **31** and the associated cylinders **32***a–i* and **34***a–i* from the common fluid inlet collector chamber **42**. Similarly, the rotation of the barrel **30** causes the pistons **36***a–i* and **38***a–i* to move radially inwardly when the associated piston ports **35***a–i* and **37***a–i* are in fluid communication with the outlet channels **43***a–d* through the radial outlet bores **45** and **47**. Each inwardly moving piston thereby provides a pressure stroke which pumps the fluid from the chambers **31** and the associated cylinders **32***a–i* and **34***a–i* into the common fluid outlet collector chamber **41**. As seen in FIG. **2**, the arrangement of the pump **10** thereby causes each set of piston balls **36** and **38** (e.g. **36***a* and **38***a*) to reciprocate alternately through suction and pressure strokes so that a cycle is completed when the piston barrel **30** rotates 90°. Hence, each set of pistons will produce four complete pumping cycles for each revolution of the input shaft **20**. As also seen in FIG. **2**, the arrangement of the pump **10** causes the piston ball sets **36, 38** (e.g. **36***a* and **38***a*) to be in different phases of the pumping cycle at any given instant in the operation of the pump **10**. (e.g. one piston set is starting a pumping or suction stroke, another set is ending a pumping or suction stroke, etc.) As a result, the suction and pumping strokes will be staggered, and the output discharge of the pump **10** will be uniform and continuous.

The pump displacement can be varied infinitely between the zero position and the maximum positive displacement position by adjusting the handle **74** to change the axial alignment of the cam members **50** and **60**. The handle **74** also can be adjusted in the opposite direction to create a negative flow of fluid through the pump **10**. The negative flow results from rotating the cam member **60** in a counter-clockwise direction, and the cam member **50** in a clockwise direction, as viewed in FIG. **2**. By such an adjustment, the circumferential position of the lobes **52***a–d* and **62***a–d* is reversed with respect to the fluid channels **43***a–d* and **44***a–d* and the radial bores **45–48** on the stationary pintle **40**. The piston balls **36** and **38** are moving outwardly through a suction stroke when the associated ports **35** and **37** are in fluid communication with the bores **46, 48** on the pintle **40**, and the balls **36, 38** are moving inwardly through a pumping stroke when the ports **35, 37** are in communication with the bores **45** and **47** on the pintle **40**.

Accordingly, the pump **10** operates in a manner identical to that described above, but the fluid flow through the pump is reversed, with the bore **15** becoming the inlet bore, and the bore **11** becoming the outlet bore. The maximum negative flow will occur when the cam members **50** and **60** are axially positioned with the bottom center-point of the cam lobe **62***a* aligned with the bottom center-point of the cam lobe **52***d*. Such alignment of the lobes **62***a* and **52***d* would occur in the illustrated embodiment at 22½° to the left of the vertical centerline of the pump **10**, as viewed in FIG. **2**. The negative pump flow likewise may be infinitely varied by positioning the cam members **50** and **60** at intermediate locations between the zero position and the above-described maximum negative flow position.

FIGS. **3** and **4** illustrate a second embodiment of the variable displacement pump in accordance with this invention. The pump **100** shown in FIGS. **3** and **4** operates in essentially the same manner as the pump **10** illustrated in FIGS. **1** and **2**. The pump **100**, however, includes side porting which eliminates the need for the stationary pintle **40** incorporated in the pump **10**. The side porting design for the pump **100** permits the pump to be more compact, and to be more readily machined and assembled. The side porting also permits the fluid channels in the pump **100** to be readily enlarged, and designed to minimize fluid friction losses.

The pump **100** includes a generally cylindrical pump housing **112** having a removable end cover plate **114**. A bearing **118** in the end plate **114** supports an input shaft **120**. A sealing ring **122** is mounted in a retaining assembly **124**, and seals the pump housing **112, 114**. A retaining ring **126** locks the assembly **124** to the end plate **114**. Retainer rings **127***a* and **127***b* are mounted in grooves within the shaft **120**, and thereby restrain the input shaft **120** from axial movement. As seen in FIG. **3**, the inner end of the shaft **120** is rotatably supported in a needle bearing **119** positioned within the housing **112**.

The pump **100** also includes an annular rotatable piston barrel **130** which is positioned within the housing **112** intermediate the ends of the shaft **120**. Splines **121** on the central portion of the shaft **120** mate with corresponding splines on the piston barrel **130**. The splined connection between the barrel **130** and the shaft **120** causes the barrel and shaft to rotate in unison while allowing the barrel to shift laterally so that the pistons carried by the barrel will be self-seating.

The barrel **130** is provided with two sets of radial piston cylinders **132** and **134**. Nine cylinders **132***a–i* and

134*a–i* are included in each set in the illustrated embodiment. The cylinders **132***a–i* and **134***a–i* are uniformly spaced around the circumference of the barrel **130,** and are arranged in axial alignment. In addition, a plurality of axial bores in the piston barrel **130** define fluid piston chambers **131** which place the adjacent aligned cylinders, such as cylinders **132***a* and **134***a*, in fluid communication, and also provide the pump **100** with axial porting. Sets of piston balls **136***a–i* and **138***a–i* are positioned within the cylinders **132***a–i* and **134***a–i*, respectively. The pistons **136** and **138** in the illustrated embodiment are provided with equal diameters, and seal the associated cylinders **134** and **136.** The piston barrel **130** also includes fluid ports **135***a–i* and **137***a–i* joining the inner ends of the cylinders **132***a–i* and **134***a–i*, respectively, with the associated fluid chamber **131.**

The pump **100** illustrated in FIGS. **3** and **4** also includes a porting insert **140** which contains porting channels for directing the flow of fluid through the pump. The insert **140** is annular in configuration, and is positioned around the shaft **120** at one end of the pump housing **112** (the left end in FIG. **3**). Sealing rings **141** seal the junctions between the insert **140** and the housing **112,** and a plurality of locking pins **142** secure the insert **140** from rotation within the housing **112.** The pins **142** permit the insert **140** to move laterally a small distance so that the insert establishes a state of equilibrium when the pump **120** is in operation. A coil spring **143** is positioned around the shaft **120** and extended between the housing **112** and the insert **140** to create a preloading force which biases the insert **140** toward the rotatable piston barrel **130.**

The porting insert **140** includes a series of conduits for directing the flow of fluid through the pump **100.** An inlet bore **111** provided in the housing **112** communicates with an annular fluid inlet collector chamber **113.** The chamber **113** in turn is in fluid communication with a set of four uniformly spaced axial inlet conduits **144***a–d* extending through the porting insert **140.** A second set of axial inlet conduits **145***a–d* is also provided in the end cover plate **114** in axial alignment with the conduits **144***a–d.* A plurality of radial and axial fluid channels **146** extends through the end plate **114** and the housing **112** and place the conduits **145***a–d* in communication with the inlet collector chamber **113.** The axial conduits **144***a–d* and **145***a–d* are arranged to be in communication with the piston chambers **131** in the barrel **130** as the barrel **130** rotates.

The pump housing **112** further defines an outlet bore **115** which is in fluid communication with an annular fluid collector chamber **116.** A set of four uniformly spaced axial outlet conduits **147***a–d* extend through the insert **140** in fluid communication with the collector chamber **116** and the rotating piston chambers **131** on the pump barrel **130.** A second set of axial outlet conduits **148***a–d* is also provided in the end cover plate **114** in axial alignment with the conduits **147***a–d.* A plurality of radial and axial fluid channels **149** extend through the end plate **114** and the housing **112** to connect the conduits **148***a–d* in fluid communication with the outlet collector chamber **116.** As seen in FIG. **4,** the inlet conduits **144, 145** and outlet conduits **147, 148** define inlet and outlet ports, respectively, which are alternately arranged in a uniform pattern adjacent each side of the barrel **130.**

The above-described porting arrangement causes the piston chambers **131** to be placed in fluid communication sequentially with the inlet conduits **144***a–d*, 145*a–d*, and the outlet conduits **147***a–d*, 148*a–d* as the piston barrel **130** rotates. The reciprocation of the ball pistons **136, 138,** caused by the rotation of the barrel **130,** will thereby pump the fluid from the inlet **111** through the outlet **115.**

The pump **100** also includes means to minimize the pressure between the rotating piston barrel **130** and the stationary end plate **114** and insert **140.** A low pressure between the barrel **130** and the adjacent stationary components is desirable so that the barrel **130** can be rotated at high speed, and the pump can be operated at high fluid pressures. As illustrated in FIGS. **3** and **3A,** the interfaces between the barrel **130** and the stationary insert **140** and end plate **114** are provided with an annular pressure relief grooves **117***a* and **117***b.* A pair of radial slots **117C** (FIGS. **3A** and **4**) places the outer relief groove **117***a* in direct fluid communication with the low-pressure interior of the housing **112.** The inner relief groove **117***b* also is in direct communication with the low pressure interior of the housing **112,** as seen in FIGS. **3** and **3A.** The grooves **117***a* and **117***b* reduce the area between the barrel **130,** the insert **140** and the end plate **114** to a minimum. Accordingly, the pump **100** includes a minimum area exposed to the high-pressure fluid in the pump **100,** so that fluid friction losses are minimized.

The fluid pressure against the rotating piston barrel **130** is further minimized by a pressure balancing device incorporated within each of the conduits **144***a–d* and **147***a–d* in the insert **140.** As seen in FIG. **3,** a slidable pressure piston **180** is positioned in each of the conduits **144***a–d* and **147***a–d* adjacent the end of the housing **112.** The outer end of each pressure piston **180** engages with the housing **112,** and a sealing ring **181** maintains the fluid seal of the associated conduit. The inner end of each piston **180** defines a circular pressure area **182** which is selectively larger than the cross-sectional area of the conduits **144***a–d* and **147***a–d.* The high pressure fluid in the conduits **144***a–d* and **147***a–d* exerts an equal pressure force in both axial directions within the cross-sectional area of the conduits **144** and **147.** However, the high pressure fluid also exerts an unbalanced axial force (to the left in FIG. **3**) against the area A (FIG. **3A**) defined by the interface between the barrel **130** and the insert **140.** The pressure area **182** of each piston **180** is a selected amount larger than the cross-sectional area of the conduits **144** and **147** so that fluid pressure against the area **182** creates a reaction force (to the right in FIG. **3**) which balances the force created against the area A. The pressure relief grooves **117***a* and **117***b* reduce the effective pressure area A. The effective pressure tending to separate the barrel **130** from the insert **140** and plate **114** due to leakage is approximately one-half of the operating pressure.

Therefore, in the preferred arrangement, the balancing area **182** (in excess of the diameter of the conduits **144, 147**) is slightly larger than one-half of the opposed area A (FIG. **3A**) so that the operation of the pump **100** creates a small fluid-pressure preloading force (to the right in FIG. **3**) which assists in maintaining sealing contact between the rotating barrel **130** and the ports in the insert **140** and plate **114** under all loading conditions. Such a fluid preloading force supplements the substantially constant preloading force of the compres-

sion spring **143.** This arrangement balances the fluid pressure and reaction forces on the insert **140** and barrel **130** during the operation of the pump **100.**

A pair of annular cam members **150** and **160** are included in the pump **100** to produce the reciprocation of the ball pistons **136***a–i* and **138***a–i*, respectively. The cam members **150** and **160** are annular in configuration, and surround the piston barrel **130** in radial alignment with the ball pistons **136, 138.** The inner periphery of the cam members **150** and **160** define arcuate cam tracks **152** and **162** which have a radius of curvature slightly larger than the radius of the engaged piston balls **136** and **138.**

The cam tracks **152** and **162** define a plurality of uniformly spaced cam lobes which engage with the piston balls **136, 138** and cause the balls to reciprocate as the barrel **130** is rotated on the embodiment illustrated in FIGS. **3** and **4,** four equally spaced lobes **152***a–d* and **162***a–d* are included on the cam tracks **152** and **162,** respectively. The lobes are uniform in amplitude and wave length, and are designed to produce a cycloidal displacement of the pistons **136, 138** so that the fluid output of the pump **100** is substantially uniform and continuous over the entire pump displacement range.

The cam members **150** and **160** are held in a relatively stationary position during the operation of the pump **100** by means of a pair of rotatable ring gears **166** and **168.** The ring gears **166** and **168** are restrained from lateral movement by the housing **112** and the end plate **114.** The inner surface of each gear includes splines or keys **169** which mate with corresponding splines or key slots provided on the adjacent outer circumference of the cam members **150** and **160.** The splines or keys **169** allow the cam members **150** and **160** to move axially within the housing **112** so that the piston balls **136, 138** will be self-seating in the cam tracks **152, 162.** The splines **169** further cause the cam members **150, 160** to rotate in unison with the connected ring gears **166, 168.**

As described above with respect to the pump **10,** the fluid displacement of the pump **100** is gradiently varied by adjusting the circumferential positioning of the cam members **150** and **160.** Beveled gear teeth **166***a* and **168***a* are provided on the inner portion of the ring gears **160** and **168** to accomplish this adjustment. The teeth **166***a* and **168***a* mesh with a bevel pinion (not shown) which is connected to a control shaft **170.** A control handle **172** is coupled to the shaft **170** to permit manual adjustment of the cam members **150** and **160.** As set forth above, with respect to the pump **10,** rotation of the control handle **172** rotates the cam members **150** and **160** an equal distance in opposite directions to change the axial relationship between the cam lobes **152***a–d* and **162***a–d*, and thereby change the reciprocation of the piston balls **136** and **138.**

As also described above, the cam members **150** and **160** are initially placed in a selected relationship with respect to the porting provided in the insert **140** and end plate **114,** to provide the pump **100** with an infinitely variable fluid displacement over a broad range in either a positive or negative direction from a zero displacement condition. The cam members **150** and **160** are initially arranged so that the lobes **152***a–d* are 45° out of phase with respect to the lobes **162***a–d*. Such an arrangement initially places the bottom center-point of the lobes **152***a–d* in axial alignment with the top center-point of the lobes **62***a–d*, and vice versa. In addition, the cam members **150** and **160** are circumferentially arranged in the housing **112** so that maximum port opening for the pump **100** occurs when one of the piston balls **136** or **138** is engaged with the top center-point of one of the cam lobes **152***a–d* or **162***a–d*, and the other axially aligned piston ball **138** or **136** is engaged with the bottom center-point of the other cam lobe **162***a–d* or **152***a–d*. Such an arrangement defines the zero displacement condition for the pump **100.** At the zero point in the operation of the pump **100,** the fluid conduits **147***a* and **148***a* are axially aligned to the maximum open port condition, with one of the piston chambers **131** in the rotating piston barrel **130.** Since the axially aligned piston balls, such as **136***a* and **138***a*, are engaged with either the top or bottom center-point of the associated cam lobes **152** *a–d* or **162***a–d*, the aligned balls will be reciprocating in opposite directions. The resulting pumping displacement of the fluid contained within the piston chamber **131** therefore will be zero.

The control handle **172** is rotated to increase the pump displacement from the zero displacement condition to a maximum displacement condition in either direction of fluid flow. In FIG. **4** the pump **100** is illustrated in the maximum positive displacement condition. The cams **150** and **160** are positioned with the lobes **152***a–d* in axial alignment with the lobes **162***a–d*, and the lobes **152***a* and **162***a* are positioned 22½° to the right of the vertical centerline of the pump **100,** as viewed in FIG. **4.** Clockwise rotation of the piston barrel **130** will thereby produce maximum displacement of fluid by the pump **10** in a positive direction with the fluid entering the suction bore **111** and discharging from the outlet bore **115.**

The operation of the pump **100** under the maximum positive displacement condition is illustrated in FIG. **4.** Since lobes **152***a–d* and **162***a–d* on the cam members **150** and **160** are in axial alignment, the aligned pistons **136***a*, 138*a*, etc., will reciprocate in unison. Thus, with the piston barrel **130** rotating in a clockwise direction in FIG. **4,** the aligned pistons **136***i* and **138***i* are moving radially outward on the cam members **150** and **160** in between the cam lobes **152***d*, 162*d* and **152***a*, 162*a*. Simultaneously, the piston barrel **130** is positioned to align the piston chamber **131** associated with the piston balls **136***i*, 138*i* with the axial inlet conduits **144***d*, 145*d*. The outward motion of the piston balls **136***i*, 138*i* thereby creates a suction force which draws fluid from the inlet collector chamber **113** into the associated piston chamber **131.** This suction stroke continues as the barrel **130** rotates and continually reduces the port opening between the chamber **131** and the conduits **144***d*, 145 *d*.

The rotation of the barrel **130** next carries the pistons **136***i*, 138*i* into the position illustrated by the aligned pistons **136***a*, 138*a* in FIG. **4.** In this condition, the piston chamber **131** is axially aligned with the outlet conduits **147***a*, 148*a*, and the piston balls **136***i*, 138*i* (as shown by the balls **136***a*, 138*a* in FIG. **4**) are approaching the cam lobes **152***a*, 162*a*. The following rotation of the barrel **130** thereby engages the balls **136***i*, 138*i* with the lobes **152***a*, 162*a*, and drives the balls radially inward through a pressure or pumping stroke. The fluid in the chamber **131** is hence forced through the conduits **147***a*, 148*a* into the outlet collector chamber **116.** The pressure stroke for the piston balls **136***i*, 138*i* is completed, and a successive suction stroke is initiated,

when the balls **136***i*, 138*i* assume the position illustrated by the balls **136***b*, 138*b* in FIG. **4.**

As illustrated in FIG. **4,** the four lobes **152***a–d* and **162***a–d* will cause each set of aligned piston balls **136***i*, 138*i*, **136***a*, 138*a*, etc., to reciprocate through four complete suction and pressure strokes for each revolution of the piston barrel **130.** Moreover, the arrangement of the components of the pump **100** in accordance with this invention causes the sets of aligned piston balls to be placed in different stages of the pressure or suction stroke at any given instant during the operation of the pump **100.** The operation of the piston balls is thus staggered to produce a substantially uniform displacement of fluid through the pump **100.**

TRANSMISSION

FIG. **5** illustrates a first embodiment of a hydrostatic transmission **200** incorporating the features and advantages of the present invention. The transmission **200** is a constant horsepower unit comprised of two units, similar to the above-described pump **100,** coupled to form a variable speed transmission.

In FIG. **5** the right-hand portion of the transmission **200** embodies a variable displacement pump **100A** which is substantially identical, in construction and operation, with the pump **100** described above. The left-hand portion of the transmission **200** embodies a variable speed hydraulic motor **100B.** The motor **100B** is also substantially identical in construction to the above-described pump **100,** but operates in a reversed fashion.

Generally, the pump **100A** in the transmission **200** is driven by the rotation of an input shaft **220.** The pump **100A** then operates to displace fluid under high pressure into the motor **100B.** The motor **100B** in turn functions to drive an output shaft **230.** The fluid displacement of the pump **100A** can be gradiently varied in a positive or negative direction, within a selected range, by manipulating the adjustment handle **222.** Similarly, the fluid displacement of the motor **100B** can be gradiently varied by manipulating the adjustment handle **232.**

The arrangement of the operation components of the pump **100A** and motor **100B** are essentially identical to the components of the pump **100** shown in FIGS. **3–4,** and have not been duplicated in FIG. **5.** The fluid interface connections between the pump **100A** and the motor **100B** are schematically illustrated in FIG. **5.** Generally, the pump **100A** includes a housing **112A** and a porting insert **140A.** A plurality of uniformly spaced and axial fluid outlet channels **149A** are provided in the housing **112A** to connect the axially aligned outlet conduits **147A,** provided in the porting insert **140A,** and **148A** (not shown in FIG. **5;** see FIG. **3**). The channels **149A** and conduits **147A** and **148A** (not shown) are in fluid communication with a common annular outlet collector chamber **116A.** The chamber **116A** also functions as the inlet fluid collector chamber for the motor **100B.** As explained in detail with respect to FIGS. **3–4,** the fluid displacement of the pump **100A** is created by discharging fluid from the piston chambers **131A** provided in a rotating piston barrel **130A.** The displaced fluid is collected in the common chamber **116A** for discharging into the motor **100B.**

The pump **100A** also includes uniformly spaced inlet channels and conduits such as illustrated by the conduits **144***a–d* and **145***a–d*, and the connecting channels **146** shown in FIGS. **3** and **4.** The inlet conduits and channels, not shown in FIG. **5** are alternately spaced with respect to the above-described outlet channels and conduits, and are in fluid communication with an annular common inlet collector chamber **113B.** The chamber **113B** also functions as the outlet collector chamber for the motor **100B.** In operation, fluid therefore is recirculated into the piston chambers **131A** of the pump **100A** from the collector chamber **113B.**

The pump housing **112A** also defines the outer housing for the motor **100B.** A plurality of uniformly fluid inlet channels **146B** are provided in the motor **100B,** in axial alignment and fluid communication with the outlet channels **149A** of the pump **100A.** The motor also includes a porting insert **140B** in axial alignment with the pump insert **140A.** Motor inlet conduits **145B** are uniformly arranged in the motor insert **140B** in axial alignment and fluid communication with the outlet conduits **147A** of the pump **100A.** As illustrated in FIGS. **3** and **4,** the motor **100B** also includes inlet conduits **144B** (not shown) axially aligned with the conduits **145B** and coupled to the conduits **145B** by the channels **146B.** The fluid is displaced into the motor **100B** through the channels **146B** and conduits **144B, 145B** from the annular collector chamber **116A.** The fluid then flows into the piston chambers **131B** provided in a rotatable piston barrel **130B.** The motor **100B** further includes uniformly spaced outlet conduits and channels such as illustrated by the conduits **147***a–d* and **148***a–d*, and the connecting channels **149** shown in FIGS. **3** and **4.** These outlet passages for the motor **100B,** not shown in FIG. **5,** are in fluid communication with the collector chamber **116A,** which functions both as an outlet collector chamber for the motor **100B** and an inlet collector chamber for the pump **100A.**

The transmission **200** also includes a compression spring **143A** which functions to apply an axial preload force to the aligned porting inserts **140A** and **140B.** The spring **143A** thereby urges the inserts **140A** and **140B** toward the associated piston barrels **130A** and **130B,** respectively, to assist in maintaining a fluid seal at the interface between the rotating piston barrels and stationary porting inserts. A common pressure piston **180A** is placed in fluid alignment with the conduits **147A** and **145B,** and operates to balance the axial pressure forces applied against the inserts **140A** and **140B,** in the same manner as described above with respect to the pressure piston **180** shown in FIG. **3.** In the transmission **200** the piston **180A** includes a central bore so that fluid will flow through the piston from the pump conduits **147A** into the motor conduits **145B** during the operation of the transmission.

In other respects the construction of the pump **100A** and motor **100B** is the same as the pump **100** illustrated in FIGS. **3** and **4.** Moreover, the operation of the pumps **100** and **100A** are the same. The motor **100B** operates in the reverse manner, as compared to pumps **100** and **100A,** and responds to the introduction of fluid under pressure from the pump **100A** to drive the output shaft **230.** Both the motor **100B** and the pump **100A** are gradiently variable within a selected range, in either a positive or negative direction.

During the operation of the hydraulic transmission **200** the output speed and direction of rotation of the output shaft **230** can be varied by manipulation of the adjustment handles **222** and **232.** If the pump **100** A is adjusted to its zero displacement setting, no fluid will

be displaced by the pump **100A** to the motor **100B**, and the speed of the output shaft **230** will be zero, regardless of the adjustment of the motor **100B**. As the pump **100A** is adjusted to increase its fluid displacement, the output speed of the motor **100B** and shaft **230** increases proportionately, up to a maximum for each setting of the motor **100B**. When the levers **222** and **232** are adjusted so that the pump **100A** and motor **100B** have the same fluid capacity or displacement, then the output speed of the shaft **230** will equal the input speed of the pump shaft **220**.

When the fluid displacement of the motor **100B** is less than the selected displacement for the pump **100A**, the transmission **200** operates as a speed increaser. If the displacement of the motor **100B** is one-half of the selected displacement for the pump **100A**, the motor speed and the speed of the output shaft **230** will be twice the speed of the pump and the input shaft **220**. Similarly, if the displacement of the motor **100B** is set to be larger than the displacement of the pump **100A**, the transmission **200** will operate as a speed reducer in the same manner. If the motor displacement is twice the pump displacement, the speed of the output shaft **230** will be one-half the speed of the input shaft **220**.

Thus, the motor **100B** and the pump **100A** can be gradiently adjusted to provide speed reducing or speed increasing ratios for the transmission **200** which can be varied within a broad range. Moreover, the direction of rotation of the output shaft **230** can be selected by adjusting the setting of the pump **100A** into either a positive or negative flow condition. Also, if the speed of the motor **100B** is doubled for any particular pump setting, the torque of the motor shaft **230** will be reduced by a factor of one-half, etc. The transmission **200** is therefore very versatile, and can be adjusted to meet the requirements of many different transmission applications.

FIG. **6** illustrates a second embodiment of a hydrostatic transmission **300** incorporating the features and advantages of the present invention. In contrast to the above-described transmission **200**, the illustrated transmission **300** is designed to produce substantially constant torque output at constant fluid pressure. As seen in FIG. **6**, the transmission **300** comprises a pump **100C** and a motor **400** coupled to form a variable speed hydrostatic transmission.

The pump **100C** in the transmission **300** is identical in construction and operation with the above-described pump **100A** incorporated in the transmission **200**. The pump **100C** is driven by the rotation of an input shaft **320**, and operates to displace fluid under high pressure into the motor **400**. The motor **400** then drives the output shaft **330**. The fluid displacement of the pump **100C** can be gradiently varied, in a positive or negative direction within a selected range, by manipulating the adjustment handle **322**. In this constant torque transmission **300**, the displacement of the coupled motor **400** is fixed at a selected multiple of the displacement of the pump **100C**. In the illustrated embodiment, the motor displacement is about three times as large as the pump displacement.

The operative components of the pump **100C** are described in detail above with respect to the pumps **100** and **100A**. Generally, the pump **100C** includes a housing **112C** and a porting insert **140C**. Four uniformly spaced axial fluid outlet channels **149C** and conduits **147C** (and conduits **148A**; see FIG. **3**) are in communication with a common annular output collector chamber **116C**. The fluid displacement of the pump **100C** is created by discharging fluid under pressure from the piston chambers **131C** provided in the rotating piston barrel **130C**. This fluid under high pressure is collected in the chamber **116C** for discharge into the coupled motor **400**. The chamber **116C** also operates as the inlet fluid collector chamber for the motor **400**.

The pump **100C** also includes four uniformly spaced inlet channels and conduits such as illustrated by the conduits **144***a–d*, **145***a–d* and the connecting channels **146**, shown in FIGS. **3** and **4**. These inlet conduits and channels are alternately spaced with respect to the above-described outlet conduits **147C** and **148A** and channels **149C**, so that the rotation of the piston barrel **130C** aligns the piston chambers **131C** sequentially with the inlet and outlet conduits during the operation of the pump **100C**. The inlet channels and conduits (not shown in FIG. **6**) are in fluid communication with an annular inlet collector chamber **413**. The transmission **300** is designed to recycle the fluid from the motor to the pump. Hence, the chamber **413** also operates as a fluid outlet collector chamber for the motor **400**.

The motor **400** is contained within an enlarged housing **412**. Four fluid outlet channels **449** and four inlet channels **446** are alternately spaced around the housing **412**. The motor inlet channels **446** are coupled to the motor inlet collector chamber **116C** and the outlet channels **449** are coupled to the motor outlet collector chamber **413**. The outer end of the motor housing **412** is closed by an end plate **414**. The plate **414** supports the output shaft **330** in a suitable bearing **419**. Retaining rings **420** engage within grooves in the shaft **330** and with the bearing **419** to prevent axial movement of the shaft **330** during operation of the transmission **300**. The end plate **414** further includes a plurality of radial fluid channels **449A** and **446A** in direct communication with the axial channels **449** and **446**, respectively. Four additional axial channels **449B** and **446B** are coupled to the radial channels **449A** and **446A** to connect the channels **449**, **446** and **449A**, **446A** with the interior of the motor **400**. Suitable O-ring seals **401** are positioned between the end plate **414** and housing **412** to maintain the fluid seal for the channels **449** and **446**.

The motor **400** also includes an annular porting insert **440** in axial alignment with the pump insert **140C**. Four inlet conduits **444** are uniformly spaced in the insert **440** in axial alignment and fluid communication with the motor inlet channels **446B** and the pump outlet conduits **147C** and **148A**. The fluid displaced from the chamber interior through the pump conduits **147C** is directed to the interor of the motor **400** through these inlet conduits **444** in the insert **440**. The porting insert **440** in the motor **400** also includes four axial outlet conduits **147** uniformly spaced between the axial conduits **444** (conduits **147** are not shown in FIG. **6**, but are evident from FIG. **3**). The outlet conduits **147** are in axial alignment and fluid communication with the motor outlet channels **449B** and are connected to the motor outlet collector chamber **413** by radial passages **449C**, as shown in phantom lines in FIG. **6**.

The motor **400** further includes a rotatable piston barrel **430**. The barrel **430** is joined to the output shaft **330** by suitable splines or keyways **421** so that rotation of the barrel imparts rotation to the shaft **330**. The piston barrel **430** in the transmission **400** is an enlarged version of one side of the piston barrel **130** incorpo-

rated in the pump **100**, as shown in FIGS. **3** and **4**. The barrel **430** in the illustrated transmission includes nine radial cylinders **434**. A reciprocating piston ball **438** is located in each of the cylinders **434**. Each cylinder **434** is connected by a radial port **437** to an axial bore **431**. The bores **431** are arranged in the barrel **430** to be in axial alignment with the fluid conduits **144, 447, 446B** and **449B** as the barrel **430** rotates with respect to the stationary insert **440** and the end plate **414**.

The motor **400** is provided with an annular cam member **450** which is arranged in rolling engagement with the piston balls **438**. The cam member **450** surrounds the piston barrel **430** and defines an arcuate cam track **452** which has a radius of curvature slightly larger than the radius of the engaged piston balls **438**. As illustrated in FIG. **4** with respect to the cam tracks **152** and **162**, the cam track **452** includes a plurality of uniformly spaced cam lobes which engage with the piston balls **438**. In the preferred embodiment, the cam track **452** is designed to include four equally spaced lobes which are uniform in amplitude and wave length, and which further are machined to produce a cycloidal displacement of the piston balls **438**. The cam member **450** is held from rotation by suitable means such as a pin **451** connected to the housing **412**. However, the pin **451** permits the cam member **450** to move axially so that the balls **438** and track **452** will be self-centering.

The motor **400** operates in the opposite manner with respect to the pump **100** described above. When the pump **100C** is operated in a positive direction, fluid under high pressure from the pump **100C** is directed into four of the piston chambers **431** through the inlet conduits **444** and **446B**. The admitted high pressure fluid enters four of the cylinders **431** and urges the piston balls **438** radially outward against the cam track **452**. The lobed configuration of the cam track **452** converts the radial reciprocating motion of the piston balls **438** into rotatory motion of the piston barrel **430**. The motion of the barrel **430** in turn drives the connected output shaft **330**. As the barrel **430** rotates, the four cylinders **431** charged with fluid align with the motor outlet conduits **147** and **449B**. The lobes on the cam tracks **452** then exhaust the fluid from the piston chambers **431** into the outlet conduits **449B** and **147**, and into the collector chamber **413**.

If the pump **100C** is operated in a negative direction, the operation of the motor **400** is reversed. The fluid from the pump **100C** will flow into the motor outlet conduits **147** and **449B** and will enter the alternate four piston chambers **431**. Then the fluid will exhaust from the motor **400** through the inlet conduits **446B** and **444**. In this negative mode of operation, the functions of the fluid collector chambers **116C** and **413** also are reversed.

As seen in FIG. **6**, the transmission **300** also includes a compression spring **143C** positioned at the interface between the pump **100C** and the motor **400**. The spring **143C** functions to apply an axial preload force to the alignment porting inserts **140C** and **440**, to urge the inserts toward the associated piston barrels **130C** and **430** and thereby maintain a fluid seal between the stationary inserts and the rotating piston barrels. A common pressure piston **180C** is placed in fluid alignment with each of the aligned conduits **147C, 444** and **148A, 147** and operates to balance the axial pressure forces applied against the inserts **140C** and **440**, in the same manner as described above with respect to the pressure pistons **180** as shown in FIG. **3**. The pistons **180C** includes a central bore so that fluid will flow through the pistons from the pump conduits **147C** and **148C** into the motor conduits **444** and **147** during the operation of the transmission **300**.

The hydrostatic transmission **300** is a speed-reducing and torque-increasing unit which produces a substantially constant torque output at constant fluid-pressure. The speed reduction of the transmission **300** results from the motor **400** having a fluid displacement larger than the displacement of the pump **100C**. In the illustrated embodiment, for instance, the motor **400** has about three times the displacement of the pump **100C**. Therefore, at maximum pump displacement, the motor speed will be about one-third the speed of the pump **100C**. The torque-increasing characteristics of the transmission **300** derive from the fact that theoretically (assuming no fluid friction losses), the input horsepower of the pump **100C** equals the output horsepower of the motor **400** at any given fluid displacement. Hence, since horsepower is a function of torque multiplied by speed, the built-in lower speed for the motor **400**, compared to the speed of the pump **100C**, increases the torque output of the motor **400**.

The illustrated pumps, motors and transmissions in accordance with this invention possess many inherent advantages. For instance, the performance of the fluid transducer (pump or motor) is improved by reciprocating the piston balls in a radial rather than an axial direction. With radially reciprocating pistons, the centrifugal force on the piston balls in pumps assists the necessary piston return movement. In fact, since the centrifugal force on the piston balls increases as a square of the speed of rotation of the piston barrel, the charging pressure necessary to feed fluid into the unit, when operating as a pump, is reduced proportionately.

Moreover, the use of pairs of reciprocating piston balls coupled by a common piston chamber in the piston barrel permits the fluid displacement of the fluid transducer, in accordance with this invention, to be variable. With only one reciprocating piston ball arrangement, as seen from the motor **400** illustrated in FIG. **6**, the fluid displacement of the transducer is fixed. The design of the preferred embodiment also allows the fluid displacement of the transducer to be infinitely varied, within a selected range, in either a positive or negative direction from a zero displacement condition. This feature of the invention results from the use of uniformly spaced and symmetrical lobes on the adjustable cam tracks which engage with the piston balls.

The preferred cam profile which induces cycloidal movement of the piston balls also induces uniform fluid flow through the transducer. The cycloidal piston movement further reduces fluid flow velocity, and fluid noise, errosion and cavitation in the transducer, by controlling the piston balls so that the piston acceleration is approximately zero when the associated piston ports open. The preferred arrangement of 9 pairs of piston balls and 8 associated fluid ports (four inlet and four outlet ports) in combination with the cycloidal piston movement also improves the fluid flow characteristics of the transducer by causing the port openings to occur at a rate which increases more rapidly than the increase in the speed of the piston balls.

The arrangement of the piston chambers and fluid passages in the pump or motor in accordance with this invention also produces uniform flow conditions, low fluid flow velocity, minimum fluid friction losses. The invention provides a common fluid collector chamber for both the inlet and outlet sides of the transducer (e.g. chambers **113** and **116** in FIG. **3**) to minimize fluid surges and pressure fluctuations. The fluid passages in the transducer are designed to facilitate uniform flow of fluid to and from the common collector chambers. In the embodiment shown in FIGS. **1** and **2**, four pairs of piston balls **36** and **38** are simultaneously coupled to the common inlet collector chamber **42** by means of the passages **44, 46** and **48.** Since the suction strokes of the piston balls are sequentially staggered, this arrangement connects the inlet collector chamber **42** simultaneously to four pairs of pistons which are in different stages of operation (i.e., one piston pair is at maximum suction; one is beginning a suction stroke, and two are at intermediate stages). In the same regard, in the pump **10** shown in FIGS. **1** and **2**, four pairs of piston balls **36, 38** are simultaneously coupled to the common outlet collector chamber **41** by means of passages **43, 45** and **47.** Thsu, the fluid displaced into the collector chamber **41** by each connected pair of piston balls **36, 38** is likewise sequentially staggered, and the fluid flow into the collector chamber **41** will thereby be uniform and continuous.

The modified pump **100** illustrated in FIGS. **3** and **4** also includes a flow passage arrangement which creates uniform flow with minimum fluid friction losses. The pump **100** includes large axial inlet conduits **144, 145** and outlet conduits **147, 148** which align with the common piston chambers **131** sequentially as the piston barrel **130** rotates. The inlet conduits **144, 145** lead from the common inlet collector chamber **113** to the piston chambers **131,** and the outlet conduits **147, 148** lead from the piston chambers **131** to the common outlet collector chamber **116.** Due to this axial porting arrangement at both sides of the barrel **130,** the porting area in the pump **100** can be maximized to reduce the flow velocity and fluid friction losses in the pump. The design of the pump **100** allows the conduits area to be expanded to any desired size downstream from the piston barrel **130,** to minimize the flow velocity and fluid friction losses.

The embodiment of the invention illustrated in FIGS. **1** and **2** also readily allows adequate lubrication of the highly machined pintle **40** and piston barrel **30.** The radial stroke of each piston ball **36, 38** creates a small, unbalanced radial load on the pintle **40,** the barrel **30** and the shaft **20.** Since each of the nine pistons **36, 38** in the piston pairs will experience four strokes per revolution of the barrel **30,** due to the four lobes on the cam tracks **52, 62,** the radial direction of the resultant small, unbalanced force will change seventy-two times for each revolution of the piston barrel **30.** As a result, the stationary pintle **40** and the rotary barrel **30** will be subjected to a slight oscillating motion which creates a squeeze lubrication effect between the engaged surfaces of the barrel and pintle.

Although the invention has been described above with a certain degree of particularity with respect to several embodiments, it should be understood that this disclosure has been made only by way of example. Consequently, numerous changes in the details of construction and in the combination and arrangement of the components as well as the possible modes of utilization for the rotary engine power assembly in accordance with this invention will be apparent to those familiar with the art, and may be resorted to without departing from the scope of the invention.

What is claimed is:

1. A variable displacement fluid transducer comprising:

rotatable shaft means;

a cylindrical piston barrel joined for rotation with said shaft means;

first and second sets of radial pistons, with each set comprising nine uniformly sized piston balls positioned for reciprocation within cylinder bores spaced uniformly around said barrel, said sets of pistons arranged on said barrel to define nine pairs of piston balls;

a piston chamber defined in said barrel for each pair of piston balls so that radial reciprocation of said pair of balls displaces fluid from the associated piston chamber;

first and second cam tracks circumscribing said first and second sets of pistons, respectively; each of said tracks including four uniformly positioned cam lobes projecting inwardly toward said barrel in frictional engagement with the balls of one set of pistons and shaped to reciprocate the engaged piston balls with a cycloidal motion so that the relative rotation of said barrel with respect to said tracks orbits said balls into engagement with the cam lobes and reciprocates said balls radially to operate said transducer with a fluid displacement from said piston chambers having a substantially uniform flow through said transducer proportionate to the relative reciprocating movement of the balls comprising each pair;

control means to move said cam tracks to shift the circumferential alignment of said lobes on said first track with respect to the lobes on said second track to vary the fluid displacement of the transducer by changing the relative reciprocating movement of the balls comprising each pair; and

fluid porting means for directing fluid sequentially to and from each of said piston chambers during the operation of said transducer.

2. A transducer in accordance with claim **1** wherein said cam tracks are arcuate in cross-section with a radius of curvature larger than the radius of the engaged piston balls and further wherein the center of curvature of said cam tracks is spaced axially inwardly from the center of the engaged piston balls so that the reaction force of said balls on said tracks urges said tracks together and thereby assists in maintaining the tracks in an axially stationary position.

3. A variable displacement fluid transducer comprising:

rotatable shaft means;

first and second sets of radial pistons, with each set comprising nine uniformly sized piston balls positioned for reciprocation within cylinder bores spaced uniformly around said barrel;

said sets of pistons being arranged on said barrel to define pairs of piston balls;

a piston chamber extending through said barrel for each pair of piston balls so that radial reciprocation of said pairs of balls displaces fluid from the associated piston chamber;

first and second axially aligned cam tracks circumscribing said barrel in radial alignment with said first and second sets of pistons, respectively; each of said tracks including four uniformly shaped and positioned cam lobes projecting inwardly toward said barrel in frictional engagement with the balls of the aligned set of pistons and shaped to reciprocate the engaged piston balls with a cycloidal motion so that the relative rotation of said barrel with respect to said tracks orbits said balls into engagement with the aligned cam lobes and reciprocates said piston balls radially to thereby operate said transducer with a fluid replacement providing a substantially uniform flow through said transducer proportionate to the relative reciprocating movement of the balls comprising each pair;

control means to shift the circumferential alignment of said lobes on said first track with respect to the lobes on said second track to vary the fluid displacement of the transducer by changing the relative reciprocating movement of the balls comprising each pair;

porting means for directing fluid sequentially to and from said piston chambers, said porting means comprising a porting member axially aligned on each side of said barrel and including axial inlet and outlet ports alternately spaced in a uniform circular pattern, with said inlet and outlet ports spaced adjacent said barrel so that rotation of said barrel places each piston chamber in sequential communication with said inlet and outlet ports, said ports and piston balls being further arranged so that said cycloidal motion of said piston balls causes said inlet and outlet ports to open to said piston chambers at a rate which increases faster than the increase in speed of reciprocation of said piston balls; and

fluid collector means in fluid communication with said inlet and outlet ports for directing fluid to said inlet ports and receiving fluid displaced from said outlet ports.

4. A fluid transducer in accordance with claim **3** wherein the cross-sectional area of said piston chambers and said inlet and outlet ports is substantially the same and the inlet and outlet ports are spaced alternately so that each rotating piston chamber closes an inlet port and opens the successive outlet port substantially simultaneously.

5. A fluid transducer in accordance with claim **4** wherein said porting members include four inlet and four outlet ports.

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