United States Patent

Schottler

[15] **3,659,971**

[45] **May 2, 1972**

[54] **FLUID TRANSDUCER**

[72] Inventor: **Henry Schottler**, North Riverside, Ill.

[73] Assignee: **Roller Gear Ltd.**, Zug, Switzerland

[22] Filed: **Apr. 23, 1970**

[21] Appl. No.: **31,314**

[52] **U.S. Cl.** 417/538, 92/107

[51] **Int. Cl.** F04b 23/06

[58] **Field of Search** 92/31–33, 107, 92/52; 417/486–488, 460, 521; 74/58, 60

[56] **References Cited**

UNITED STATES PATENTS

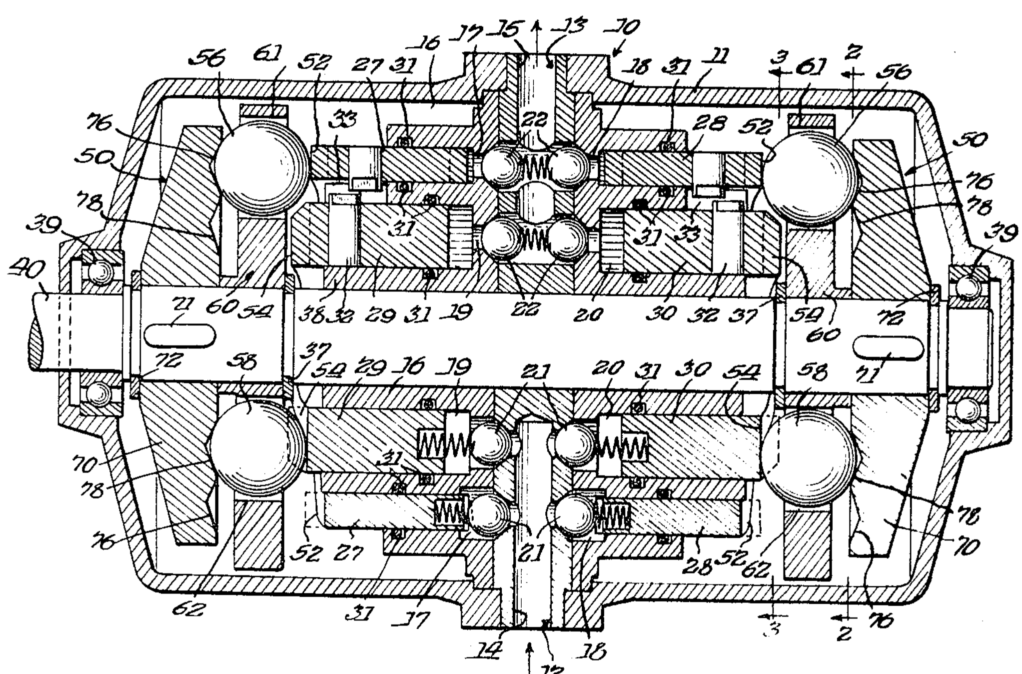
3,403,668	10/1968	Schottler	74/60
1,177,126	3/1916	Miller	92/107
3,482,525	12/1969	Bartolo et al.	74/60
2,973,137	2/1961	Jensen	92/107
1,900,284	3/1933	Herzmark	417/460
3,105,415	10/1963	De Muth	91/480

Primary Examiner—William L. Freeh
Attorney—Hume, Clement, Hume & Lee

[57] **ABSTRACT**

A cam drive mechanism for reciprocating a plurality of pistons in a fluid transducer comprising opposed annular lobed cam surfaces and flat reaction tracks for each piston; a plurality of rolling members positioned between each of said opposed cam surfaces and tracks; and a common retainer spacing said rolling means uniformly around said cam surfaces; with said retainer and tracks adapted so that said rolling means roll between each of said opposed surfaces and tracks with substantially the same angular movement and speed.

5 Claims, 8 Drawing Figures



Patented May 2, 1972 3,659,971

3 Sheets-Sheet 1

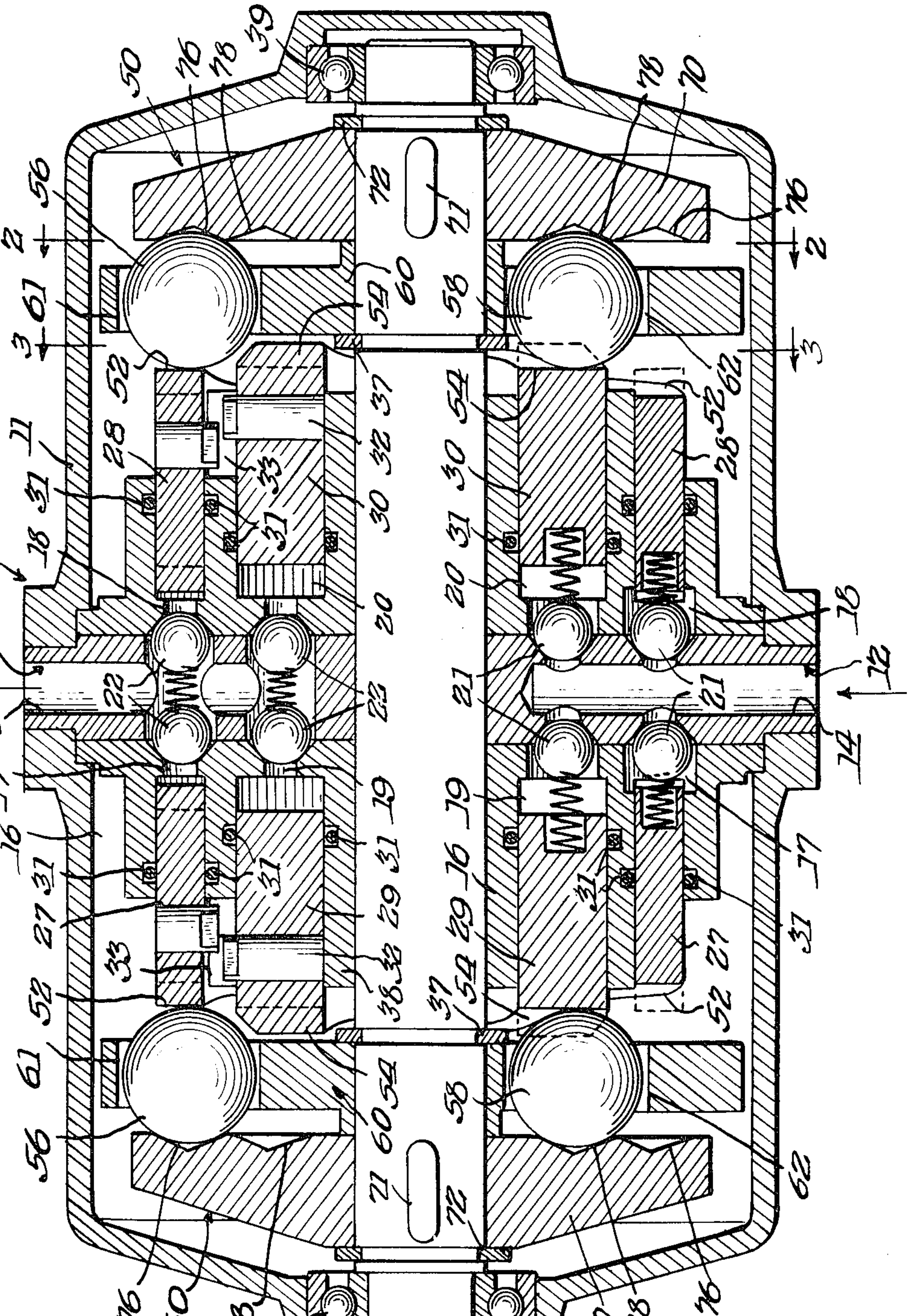


Fig. 1

Inventor
Henry Schottler

By Hume, Clement, Hume, & Lee
Attys.

Fig. 2.

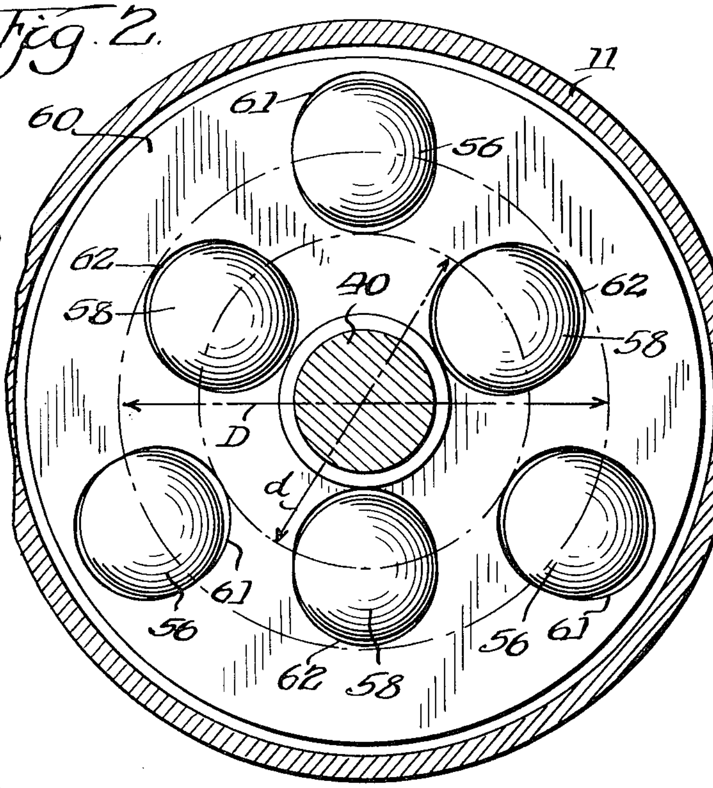


Fig. 5.

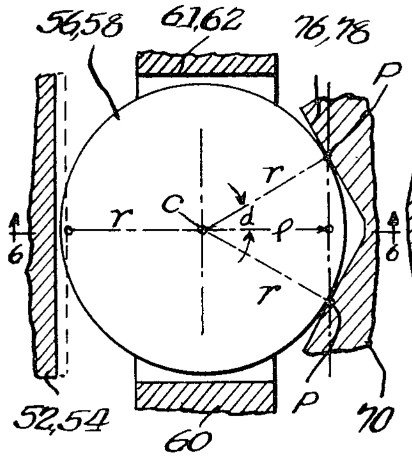


Fig. 3.

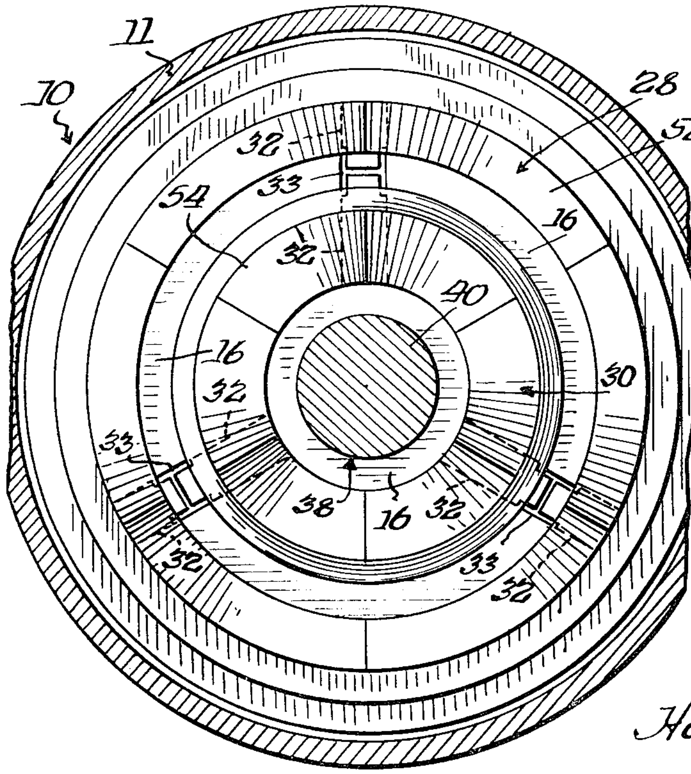
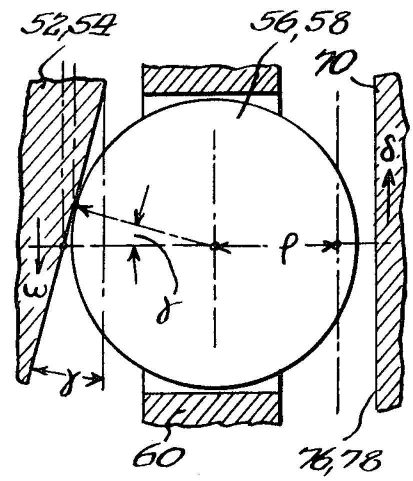
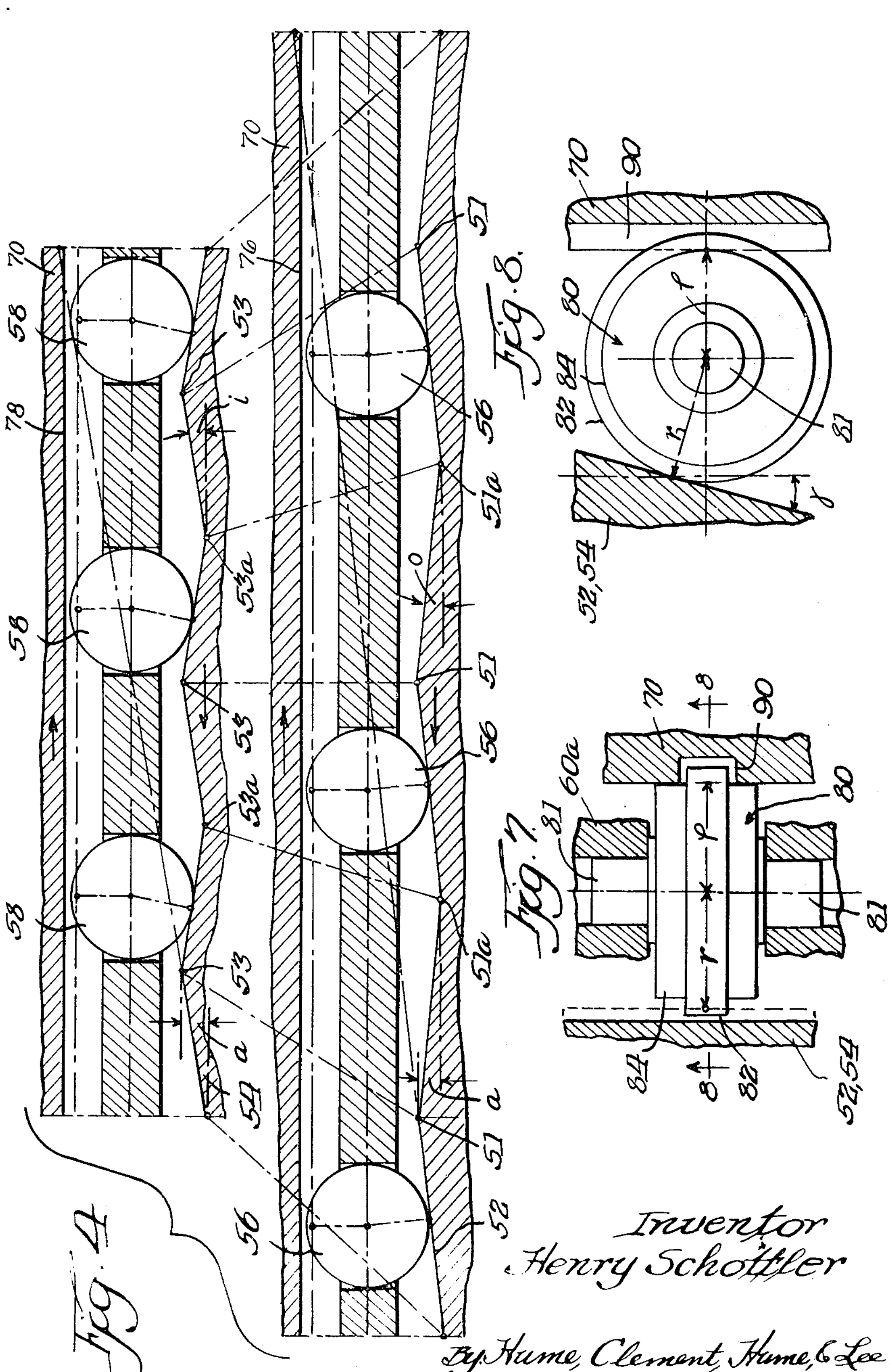


Fig. 6.



Inventor
Henry Schottler

By. Hume, Clement, Hume, & Lee
Attys.



Inventor
Henry Schottler

By Hume, Clement, Hume, & Lee.
Attys.

FLUID TRANSDUCER

BACKGROUND AND GENERAL DESCRIPTION

This invention relates generally to an improved drive mechanism for converting rotary motion into reciprocating motion, and vice versa. More particularly, this invention relates to an improved cam drive mechanism for synchronizing the movements of a plurality of pistons in a fluid transducer.

The type of fluid transducer for which this invention is particularly suited is illustrated in my previous U.S. Pat. No. 3,403,668, issued on Oct. 1, 1968, entitled "Fluid Transducer." That patent discloses and claims fluid transducers which incorporate a cam-controlled drive mechanism for converting the rotary motion of an input shaft into the reciprocating motion of a piston, such as in a pump or compressor, or for likewise converting the reciprocating motion of a piston into a rotary motion of an output shaft, such as in an engine or hydraulic motor. The cam-controlled drive mechanism set forth in my earlier patent eliminated the need for piston crank shafts, connecting rods and the like, and increased the efficiency and loading capacity of the fluid transducers. The cam drive mechanisms also permit the transducers to be readily adaptable for operation with variable output and fluid displacement characteristics.

The present invention relates to an improved cam drive mechanism that simplifies the design and synchronization of a fluid transducer having a plurality of closely arranged reciprocating components, such as concentric annular pistons. If such a transducer incorporates cam drive mechanisms formed from pairs of opposed cams with rolling elements therebetween, the planetary motion of the rolling elements must be guided and controlled by bevelled gears or other suitable devices so that the rolling elements maintain the proper position with respect to the opposed cams. Without guiding the motion of the rolling elements there is a danger that the rolling elements may slide into a dead center or stall position between the opposed cam lobes. The incorporation of devices, such as bevel gears to guide the motion of the rolling elements would result in a relatively complicated design for a transducer including a plurality of closely arranged pistons.

Furthermore, if each of the reciprocating pistons in such a transducer is provided with a separate cam-controlled drive mechanism, the transducer must include bevel gears or other suitable devices to synchronize the operation of the various drive mechanisms so as to coordinate the stroke and fluid pressure on each piston and produce a smoothly operating transducer. The devices need for synchronizing the motions of a plurality of cam drive mechanisms also are relatively complicated and cumbersome in transducers incorporating a plurality of close-fitting reciprocating pistons.

The present invention minimizes or eliminates the foregoing problems by providing a multiple-piston transducer with a cam-controlled drive mechanism that has a compact and simple design. In this improved transducer, the problems of guiding the motion of the rolling elements with respect to the opposed cam lobes in the drive mechanisms is eliminated by providing each drive mechanism with a camless reaction disc. Each reaction disc is positioned for engagement with the rolling elements of the drive mechanism and each disc defines circumferential tracks for the rolling elements associated with a plurality of reciprocating pistons. Since the tracks are camless, the rolling elements will roll uniformly along the entire track length and the motion of the rolling elements need not be guided with respect to the reaction disc.

The present cam drive mechanism also synchronizes the motion of a plurality of reciprocating pistons in a simple fashion. Each cam drive mechanism in the transducer incorporates a plurality of rolling means for simultaneously driving a plurality of closely arranged pistons by rolling between a cam surface and an opposed track on an associated reaction disc. The rolling means are mounted in a common rotatable retainer so that the angular speed of the rolling means driving a plurality of pistons are equalized. Also, the tracks in the reaction disc are adapted to provide the rolling means with effective rolling radii which synchronize the angular movement of the plurality of rolling means in each cam drive mechanism.

Additional objects 1 is features of this invention will become more readily apparent from the following description of exemplary embodiments thereof, taken in conjunction with the accompanying drawings, wherein:

FIG. is a longitudinal sectional view of a pump incorporating the features and advantages of the present invention;

FIG. 2 is a cross-sectional view, taken along the line 2—2 in FIG. 1, showing the common roller retainer incorporated in each cam drive mechanism of the present invention;

FIG. 3 is a cross-sectional view of the pump, taken along the line 3—3 in FIG. 1;

FIG. 4 is a plane developed view of the cam drive mechanisms incorporated in the pump illustrated in FIG. 1;

FIG. 5 is an enlarged sectional view of a portion of the cam drive mechanism of the present invention;

FIG. 6 is an enlarged partial sectional view of the cam drive mechanism as viewed along the line 6—6 in FIG. 5;

FIG. 7 is an enlarged partial sectional view of a modified cam drive mechanism incorporating the features and advantages of the present invention; and

FIG. 8 is an enlarged partial sectional view of the modified cam drive mechanism, as viewed along the line 8—8 in FIG. 7.

EXEMPLARY EMBODIMENTS

Referring generally to the drawings, the pump embodying the features and advantages of the present invention is generally indicated by the reference numeral **10**. The pump **10** includes a sealed pump housing **11** which is symmetrical about a vertical center line, as illustrated in FIG. **1**. An inlet port **12** in the housing **11** permits charging of fluid to flow into the pump **10**, and an outlet port **13** provides means for discharging the fluid from the pump **10**. Intake and exhaust channels **14** and **15** are arranged in fluid communication with the inlet and outlet ports **12** and **13**, respectively, to direct the flow of fluid through the pump **10** during the pumping operation.

As illustrated in FIG. **1**, the pump **10** includes an annular cylinder block **16** mounted in the central portion of the housing **11**. This block **66** is machined to define a plurality of concentric annular cylinders **17**, **18**, **19** and **20**. Each of these annular cylinders **17–20** is aligned with the longitudinal axis of the housing **11**. Each cylinder is also in fluid communication with the intake channel **14**, through spring-biased intake check valves **21**, and with the exhaust channel **15**, through spring-biased exhaust check valves **22**. As well-known to those skilled in the art, the check valves **21** and **22** respond to the pressure of the fluid in the pump **10** to permit the fluid to be drawn into the cylinders **17–20**, through the valves **21**, and then exhausted from the cylinders through the valves **22**.

The pump **10** further includes a plurality of annular pistons **27**, **28**, **29** and **30**. As illustrated in FIG. **1**, these pistons **27–30** are mounted within the annular cylinders **17–20**, respectively, and are adapted for reciprocating movement within the cylinders during the operation of the pump **10**. Elastomeric seal rings **31**, or other suitable piston ring arrangements, are provided to seal the pistons **27–30** in their respective annular cylinders **17–20**.

It is desirable to provide the working face of the annular pistons **27-30** with the same effective area to provide a uniform and continuous flow of fluid from the exhaust outlet **13** during the operation of the pump **10**. To accomplish this feature, the pistons **27–30**, and their associated cylinders **17–20**, are dimensioned to compensate for the differing piston diameters. Thus, the outer cylinders **17** and **18**, and the associated outer pistons **27** and **28**, are dimensioned to have a relatively small radial width or thickness, and the inner cylinders **19** and **20** and inner pistons **29** and **30** are dimensioned to have a portionately larger radial width or thickness. The reciprocating annular pistons **27–30** will therefore have the same effective working area during the operation of the pump

10 and will apply the same pressure to the fluid being pumped despite the fact that the diameter of the outer pistons 27 and 28 is substantially larger than the diameter of the inner pistons 29, 30.

In order to operate effectively, the above-described pistons 27–30 also must be secured from rotating within the pump housing 11. As illustrated in FIGS. 1 and 3, each of the pistons 27–30 is thus provided with a key 32, and the adjacent portions of the cylinder block 16 are provided with axial keyways 33. The keys 32 and keyways 33 prevent rotational movement of the pistons 27–30, but allow the pistons to slide freely in an axial direction within the housing 11. The keys 32 and keyways 33 are arranged in a uniform circumferential pattern on the pistons 27–30 so that the loads on the keys 32 during the operation of the pump 10, and the resulting forces on the pistons, are uniformly distributed.

The pump 10 in accordance with this invention also includes a central input shaft 40 for driving the pistons 27–30. As illustrated in FIG. 1, the shaft 40 is supported by bearings 39 in the ends of the pump housing 11. The illustrated arrangement of the opposed annular pistons 27–30 assures that the axial forces resulting from the operation of the pump 10 are self-contained. Hence, these bearings 39 will be subjected to substantially no axial load, and can be simple, non-thrust bearings. The shaft 40 extends through a central aperture 38, provided in the cylinder block 16. Snap rings 37 on the shaft 40 engage with the adjacent ends of the block 16 and prevent the block from shifting axially with respect to the shaft 40.

In accordance with this invention, the pump 10 further includes a pair of cam drive mechanisms 50 for transmitting the motion of the input shaft 40 to the pistons 27–30. Each of the cam drive mechanisms 50 are identical in construction, and operate in the same manner to convert the rotary motion of the input shaft 40 into reciprocating motion of at least two of the annular pistons 27–30. Accordingly, only one cam mechanism 50 is described in detail below.

Each of the drive mechanisms 50 includes an annular, multi-lobed cam surface connected to each of a pair of pistons. Thus, the end portions of the outer pistons 27 and 28 are machined to define multi-lobed cam surfaces 52, and the end portions of the inner pistons 29 and 30 are machined to define multi-lobed cam surfaces 54. As best seen in FIG. 4, in the illustrated pump 10, the cam surface 52 defines three uniformly spaced lobes 51, and the cam surface 54 defines three uniformly spaced lobes 53. Due to the three lobes on the cams 52 and 54, each of the pistons 27–30 will travel through three power strokes for every two complete rotations of the input shaft 40.

Also, as illustrated in FIGS. 1 and 4, the amplitudes (a) of the lobes 51 and 53 are equal in length. By this arrangement, the resultant power strokes of the pistons 27–30 will be of equal length during the operation of the pump 10. The equal strokes of the pistons 27–30 will cause uniform displacement of the fluid from the cylinders 17–20 and provide a substantially steady and uniform flow of fluid from the pump 10.

FIGS. 1 and 4 further illustrate the circumferential relationship of the cam surfaces 52 and 54. As shown clearly by the phantom lines in FIG. 4, in this embodiment the cams 52 and 54 are arranged so that the respective lobes 51 and 53 are in radial alignment. This arrangement places the lobes 51 on the cam 52 in radial alignment with the lobes 53 on the cam 54, and similarly places the cam recesses 53A on the cam 54 in radial alignment with the cam recesses 51A on the cam 52.

As illustrated in FIGS. 1 and 2, the drive mechanisms 50 in accordance with this invention incorporate a set of rolling elements in engagement with each of the multi-lobed cams 52 and 54. A first set of rolling elements, comprising a plurality of balls 56, engage with the cam surfaces 52 on the outer pistons 27 and 28, and a second set of rolling elements, comprising a plurality of balls 58, similarly engage with the cam surfaces 54 on the inner pistons 29 and 30. Since the cams 52 and 54 are provided with three lobes in this embodiment, each set of rolling elements includes three balls 56 and 58 to correspond with the number of associated cam lobes.

Both sets of balls 56 and 58 in each of the drive mechanisms 50 are mounted within a common ball retainer 60. As illustrated in FIG. 2, the outer balls 56 are positioned within apertures 61 in the retainer 60 at a large diameter D. The diameter D is selected so that the centers of the balls 56 are arranged in axial alignment with the cams 52 on the outer pistons 27 and 28. In the same manner, the inner balls 58 are arranged in apertures 62 in the retainer 60 at a small diameter *d*, which places the centers of the balls 58 in axial alignment with the cams 54 on the inner pistons 29 and 30. The apertures 61 and 62 in the retainer 60 are slightly elongated in a radial direction with respect to the diameter of the associated balls 56 and 58. With this arrangement, the apertures 61 and 62 permit the balls 56 and 58 to shift radially and be self-seating within the cam drive mechanism. The circumferential dimension of the apertures 61 and 62 is substantially the same as the diameter of the balls 56 and 58 so that the apertures hold the balls 56 and 58 in a predetermined circumferential position on the retainers 60, within a close tolerance.

The apertures 61 and 62 are uniformly spaced in the retainers 60 so that the three balls 56 are angularly spaced by 120° at the large diameter D, and the three balls 58 are angularly spaced by 120° at the small diameter *d*. Moreover, the apertures 61 and 62 are staggered in the retainers 60 so that the balls 56 are uniformly spaced between the balls 58, and vice versa. By such an arrangement, the balls 56 and 58 will cooperate to alternately drive the outer pistons 27, 28 and the inner pistons 29, 30 inwardly to produce power strokes at uniformly spaced time intervals during the operation of the pump 10. The fluid output of the pump 10 will thereby be substantially uniform and continuous.

As shown in FIG. 1, each of the ball retainers 60 is mounted on the input shaft 40 adjacent the ends of a pair of inner and outer pistons 27, 29 and 28, 30. The snap rings 37 on the shaft 40 prevent the retainers 60 from shifting axially inward along the shaft 40. The fit between the shaft 40 and the retainer 60 is sufficiently loose so that the retainers can rotate freely about the shaft during the operation of the pump 10.

Each cam drive mechanism 50 in accordance with this invention further includes a camless reaction disc 70. As illustrated in FIG. 1, a disc 70 is positioned on the shaft 40 adjacent each ball retainer 60. Suitable keys 71 lock the discs 70 from rotating on the shaft 40, and suitable snap rings 72 prevent the discs from sliding outwardly on the shaft. Further, the discs 70 abut against the hub portion of the adjacent ball retainer 60 so that inward shifting of the retainers and discs on the shaft 40 is precluded.

In accordance with this invention, the reaction discs 70 engage with the adjacent sets of rolling balls 56 and 58 maintained in the common ball retainer 60. As shown in FIGS. 1 and 5, each of the discs 70 includes concentric roller tracks 76 and 78 in axial alignment with the balls 56 and 58. By this arrangement the balls 56 and 58 will roll along the reaction disc 70 in the tracks 76 and 78, respectively, while simultaneously rolling along the surface of the opposed cam surfaces 52 and 54.

To begin the operation of the pump 10, the input shaft 40 is rotated by an external force (not shown). The rotation of the shaft 40 is transmitted to the reaction discs 70 through the keys 71 so that the discs 70 rotate at the same angular speed as the shaft. The angular movement of the discs 70 in turn causes the balls 56 and 58 to roll between the opposed tracks 76, 78, and the cams 52, 54. The outer balls 56 thereby orbit around the shaft 40, in a planetary fashion, at the large diameter D, and the inner balls 58 likewise orbit about the shaft at the diameter *d*.

Since the reaction discs 70 are keyed on the shaft 40 against axial movement, the rolling action of the balls 56 and 58 will cause the pistons 27–30 to reciprocate axially within their respective annular chambers as the balls engage with the lobes 53 and 51 on the cams 52, 54. The pistons 27–30 thereby apply a pumping force to the fluid in the pump cylinders 17–20. Any suitable biasing means (not shown) can be employed to return the pistons 27–30 to their outward positions after the pumping stroke is completed. Since each of the cams **52** and **54** include three cam lobes in the illustrated embodiment, each piston **27–30** will move through three inward pumping strokes for every two revolutions of the input shaft **40**.

The use of the reaction disc **70** and the common ball retainer **60** in each of the drive mechanisms **50** greatly simplifies the design for multiple piston transducers in accordance with this invention. Since the action between the camless tracks **76** and **78** and the balls **56** and **58** is constant, there is no need to include complicated and expensive bevel gearing or the like to control the positioning of the ball retainer **60**, and the balls **56** and **58** with respect to the lobed cams **52** and **54**. The positioning of the two sets of balls **56** and **58** in the common ball retainer **60** also simplifies the transducer design by synchronizing the motion of the balls so that the two sets of balls **56** and **58** orbit around the shaft **40**, at different diameters d and D, with the same angular speed.

The cam drive mechanisms **50** facilitate the use of the same amplitude a for the lobes **51A** and **53A** on the cams **52** and **54**. The mechanisms **50** thereby drive the associated pistons **27–30** through the same power stroke during the operation of the pump **10**. Each piston therefore applies equal pumping pressure to the fluid and will displace an equal volume of fluid.

The use of equal amplitudes a for the cam lobes **51A** and **53A** create different cam angles γ for the cams **52** and **54**, since the diameters of the cam are different. As clearly illustrated in FIG. 4, the cam angle γ for the small diameter cam **54** is greater than the corresponding cam angle γ_o for the large diameter cam **52**, when both cams have the same lobe amplitude a. This difference between the angles γ_o and γ_i is a function of the difference between the diameters d and D for the cams **54** and **52**, respectively.

The existence of the different cam angles and diameters for the cams **52** and **54** inhibits the synchronization of the motion fro the two sets of balls **56** and **58**. Since both sets of balls **56** and **58** are in the same retainer **60**, their angular speeds will be equal. However, as evident from FIG. 4, the balls **56** and **58** will roll along paths of different circumferential lengths on the associated cams **52** and **54**, respectively, for each revolution of the common ball retainer **60**, because the cam angles and diameters are different. Thus, there is a natural tendency for one set of balls to slide on the cams and resist the smooth rolling motion of the other set of balls during the operation of the pump **10**. The use of the common ball retainer **60** and the different circumferential lengths for the paths of the balls **56** and **58** on the cams **52** and **54** combine to create friction and reaction forces which tend to prevent the proper functioning of the pump **10**.

In accordance with this invention the potentially adverse effect of the different circumferential lengths of the cams **52** and **54**, and different cam angles γ_i and γ_o are offset by providing means to correspondingly vary the effective rolling radii for the balls on the adjacent reaction disc **70**. Generally, these different rolling radii are produced by recessing the roller tracks **76** and **78** on each disc **70** in a selected manner so that the balls **56** and **58** engage the tracks at selected contact points.

FIGS. 4, 5 and 6 illustrate more specifically the manner in which the tracks **76** and **78** are constructed in accordance with this invention to produce the desired effective rolling radii ρ. Each track is formed as a V-shaped recess or groove in the interior face of the disc **70**. As shown in FIG. **5**, the balls **56** and **58** will thereby engage with the sides of the grooved tracks at contact points P. The location of the contact points P with respect to the center C of the balls **56** and **58** is a direct function of the track groove angle α. Hence, the effective rolling radii ρ for the balls **56**, **58** in the grooved tracks **76**, **78** is likewise a function of the track groove angle α. As seen in FIG. 5, the effective radii are smaller than the actual radii r for the balls, and decrease in length as the groove angle α increases. The effective radius ρ of the balls **56** and **58** therefore depends upon the following geometric relationship between the balls and the track grooves:

(1) $\rho = r\cos\alpha$

Thus, one of the tracks, such as the outer track **76**, can be designed to have a particular groove angle α_o. The selection of the angle α_o thereby determines the effective rolling radius ρ_o through which the balls **56** roll along the outer track **76**. Then, the groove angle α_i for the inner track **78** can be calculated so that the effective rolling radius for the engaged balls **58** will compensate for the different diameters D and d, cam angles γ_o and γ_i, and circumferential lengths of the cams **52** and **54**.

The relationship between the cam angles γ and the effective rolling radii ρ for the cams **52**, **54**, and the balls **56**, **58** and the groove angle α for the tracks **76**, **78** can be described in geometric terms. For purposes of analysis, it is assumed that the ball retainers **60** are held stationary and the opposed cams **52**, **54** and tracks **76**, **78** are frictionally connected with each other through the rolling balls **56**, **58**, respectively. Referring to FIGS. **5** and **6**, it will be understood that, in general, the angular distance ω through which the balls **56**, **58** will roll on the cam surfaces **52**, **54** in response to an angular movement δ of the associated disc **70** will be proportionate to the relationship between the ball radium r and the effective radius ρ for the engaged track **76**, **78**. Hence, taking into account the fact that the balls **56**, **58** roll along the inclined cams **52**, **54** having a cam angle γ:

(2) $\omega/\cos\gamma r = \delta/\rho$

Also, since the cams **52**, **54** are restrained from rotation and the tracks **76**, **78** are on the same rigid disc **70**, the following relationships also prevail between the movement of the outer balls **56** at the diameter D and the inner balls **58** at the diameter d along the outer and inner cams **52**, **54** and tracks **76**, **78**:

(3) $\omega_o/\omega_i = \delta_o/\delta_i = D/d$; or

(4) $\omega_o = \omega_i\ (D/d)$; and

(5) $\delta_0 = \delta_i\ (D/d)$

Also, from equation (2) above it is evident that the angular movement δ_o of the outer grooved track **76**, and the angular movement δ_i for the inner grooved track **78** can be expressed as follows:

(6) $$\delta_0 = \frac{\omega_0 \varphi_0}{\cos \gamma_0 r_0}$$

and

(7) $$\delta_i = \frac{\omega_i \varphi_i}{\cos \gamma_i r_i}$$

In accordance with this invention, the track groove angles α (FIG. 5) are selected so that the resultant effective radii ρ_o and ρ_i offset the effect of the different cam angles γ_o and γ_i for the cams **52** and **54**, respectively, and equalize the effective angular movement of the tracks **76** and **78**. Hence, from equations (6) and (7):

(8) $$\delta_0 = \delta_i;\ \text{or}\ \frac{\omega_0 \varphi_0}{\delta_0 \cos \gamma_0 r_0} = \frac{\omega_i \varphi_i}{\delta_i \cos \gamma_i r_i}$$

Substituting equations (3) and (4) for ω_o and δ_o, and cancelling common terms in equation (8), it is apparent that with the radii r_o and r_i for the balls **56** and **58** equal:

(9) $\rho_o/\cos\gamma_o = \rho_i/\cos\gamma_i$; or

(10) $\rho_i = \rho_o(\cos\gamma_i/\cos\gamma_o)$

Further, as seen in FIGS. **4** and **6**, the relationship between the cam amplitude a and the cam angles γo and γi for the three-lobed cams **52** and **54** in the illustrated embodiment can be expressed:

(11) $$\tan \gamma_0 = \frac{a}{\pi D/6}$$

(12) $$\tan \gamma_i = \frac{a}{\pi d/6}$$

Since the cam amplitudes a for the cams **52** and **54** are equal, from equations (11) and (12):

(13) $$\frac{a}{\pi/6} = D \tan \gamma_0 = d \tan \gamma_i;$$

or

(14) $\tan\gamma_i = \tan\gamma_o(D/d)$

The method of employing the above-described relationships for designing the cam drive mechanisms **50** will be evident from the following example where the radii r_i and r_o of the balls **56** and **58** are the same; the groove angle α_o for the outer track **76** is selected as 20°; D is 84 mm; d is 56 mm; and a is 3.5 mm.

From equation (1):

$$\rho_o = r \cos 20° = 0.93969r$$

With equation (11):

$$\tan \gamma_0 = \frac{a}{\pi D/6} = \frac{3.5}{\pi \cdot 84/6} = 0.075788$$

Therefore, $\gamma_o = 4° 20'$, and $\cos \gamma_o = 0.99714$.

Then from equation (14):

$$\tan \gamma_i = \tan\gamma_0 \frac{D}{d} = 0.075788 \left(\frac{84}{56}\right)$$

$$\tan\gamma_i = 0.11368$$

Therefore $\gamma\gamma_i = 6° 29'$ and $\cos \gamma\gamma_i = 0.99360$

Next, from equation (10) above, the effective rolling radius ρ_i for the balls **58** rolling in the track **78** can be determined:

$$\varphi_i = \varphi_0 \frac{\cos \gamma_i}{\cos \gamma_0} = 0.93969r \frac{0.99360}{0.99714}$$

$$\rho_i = 0.93634r$$

Finally, from equation (1):

$\rho_i = r \cos\alpha i = 0.93634r$

$\cos\alpha_i = 0.93634$ and thus

$\alpha_i = 20° 33'$

Therefore, referring to FIG. **5**, a 40° groove **76** ($2\alpha_o$) is to be combined with a 41° 6′ groove **78** ($2\alpha i$). With such groove angles, the angular speed and movement of the two sets of balls **56** and **58** in the common retainer **60** will be equalized. The balls **56**, **58** will therefore roll smoothly, and the friction and reaction forces will be substantially reduced during operation of the pump **10**.

FIGS. **7** and **8** illustrate a modified cam drive mechanism in which the rolling balls **56**, **58** have been replaced by pressure rollers **80**, and the angular grooves **76**, **78** in the reaction disc **70** have been replaced by modified grooved tracks **90**. The rollers **80** are adapted to have stepped surfaces **82** and **84** which define different rolling radii for the rollers with respect to the cams **52**, **54** and reaction disc **70**. The larger roller surface **82** defines the radius r through which the rollers **80** engage with the cams **52**, **54**. In the same regard, the smaller rolling surface **84** defines a smaller effective rolling radius ρ through which the rollers **80** engage the reaction disc **70**. The grooved track **90** is dimensioned to receive the larger rolling surface **82** with sufficient clearance so that the surface **82** can freely roll along the disc **70**. A modified roller retainer **60A** receives the axle pins **81** for each roller **80**, and supports a set of rollers **80** in axial alignment with each cam surface **52** and **54**.

The modified cam drive mechanism incorporating the rollers **80** can be designed, in accordance with the above-described equations, in a manner similar to the procedure for designing the cam mechanisms **50** including the balls **56** and **58**. First, a desired radius (r_o and r_i) is chosen for the larger roller surfaces **82** on each of the rollers **80**. An effective rolling radius ρ for the smaller roller surface **84** on one of the rollers **80** is also selected, such as the rolling radius $\rho_o$for the outer rollers **80** positioned opposite the cams **52**. The diameters D and d, and the cam amplitude a for the cams **52** and **54** will also be known.

From this data, the effective rolling radius ρ_i for the smaller surface **84** on the set of other rollers **80** on the common retainer **60A** can be calculated. For example, the foregoing data will permit the calculation of the rolling radius ρ_i for the surface **84** on the inner rollers **80** by employing equation (**10**). Although any desired initial rolling radius ρ can be chosen to begin the calculations described above, it is expected that the selected radius ρ, such as the radius ρ_o will be chosen to equal the rolling radius r for the rollers **80**. By such an arrangement, the steps on one of the sets of rollers **80** in each common retainer **60A** will be eliminated, and the design for the cam mechanism thereby simplified. The selection of the effective radius ρ_o to equal r for one of the rollers **80** further simplifies the cam drive mechanism by eliminating the need for the annular groove **90** in the reaction disc **70** adjacent that set of rollers.

Although the invention has been described above with a certain degree of particularity, it should be understood that the present disclosure has been made only by way of example. Consequently, numerous changes in the detail of construction and in combination of components of the transducer, as well as in the possible modes of utilization, will be apparent to those skilled in the art, and may be resorted to without departing from the spirit and scope of the invention.

What is claimed is:

1. A cam drive mechanism for reciprocating a pair of pistons in a fluid transducer comprising:
   - cam means defining first and second annular cam surfaces each adapted to reciprocate one of said pistons, and having selectively different circumferential lengths;
   - first and second annular flat reaction tracks spaced in opposed positions adjacent said first and second cam surfaces, respectively;
   - rolling means positioned between said opposed cam surfaces and reaction tracks; and
   - roller retaining means engaged with said rolling means and rotatable to orbit said rolling means between said opposed cam surfaces and tracks at a common angular speed;
   - said rolling means engaging each cam surface at a selected cam rolling radius and each reaction track at a selected track rolling radius as said rolling means orbit therebetween; and
   - said opposed cam surfaces and tracks being arranged to define effective cam and track rolling radii which compensate for the different circumferential lengths of said cam surfaces and cause said rolling means to roll freely in engagement between said opposed first and second cam surfaces and tracks with substantially the same angular movement and speed, to thereby substantially reduce reaction and friction losses in said drive mechanism.

2. A cam drive mechanism in accordance with claim 1 wherein said first and second cam surfaces are concentrically positioned and include a plurality of uniformly spaced cam lobes of equal lobe amplitude.

3. A cam drive mechanism for reciprocating a plurality of pistons in a fluid transducer comprising:
   - first and second annular an concentric cam members each adopted to reciprocate one of said pistons, said cam members having different selected diameters and including a plurality of uniformly spaced cam lobes having substantially the same lobe amplitude and selectively different cam angles;
   - first and second reaction tracks comprising annular and concentric grooves spaced in opposed positions adjacent said first and second cam members, respectively;
   - a plurality of rolling balls positioned between said opposed cam members and reaction tracks and uniformly spaced with respect to said cam lobes, said rolling balls engaging each cam member at a selected cam rolling radius and each reaction track at a selected track rolling radius as said balls orbit therebetween;
   - said cam and track rolling radii being selected so that said balls roll freely along said first and second tracks with the same angular movement; and
   - roller retaining means engaged with said balls and rotatable between said opposed cam members and tracks to orbit said balls therebetween at a common angular speed.

4. A cam drive mechanism for reciprocating a plurality of pistons in a fluid transducer comprising:
   - first and second annular and concentric cam members each adopted to reciprocate one of said pistons, said cam members having different selected diameters and including a plurality of uniformly spaced cam lobes having substantially the same lobe amplitude and selectively different cam angles;

first and second reaction tracks comprising annular and concentric grooves spaced in opposed positions adjacent said first and second cam members, respectively;

a plurality of multi-step rollers positioned between said opposed cam members and reaction tracks and uniformly spaced with respect to said cam lobes, said rollers having an outer surface engaging each cam member at a selected cam rolling radius and an inner surface engaging each reaction track at a selected track rolling radius within one of said grooves as said rollers orbit between said opposed cams and tracks;

said cam and track rolling radii being selected so that said rollers roll freely along said first and second tracks with the same angular movement; and

roller retaining means engaged with said rollers and rotatable between said opposed cam members and tracks to orbit said rollers therebetween at a common angular speed.

5. A cam drive mechanism for reciprocating a plurality of pistons in a fluid transducer comprising:

first and second annular cam members each adapted to reciprocate one of said pistons, said cam members including cam lobes having selectively different cam angles γi, and $\gamma 2$, respectively;

first and second annular reaction tracks spaced in opposed positions adjacent said first and second cam members, respectively;

rolling means positioned between said opposed cam members and reaction tracks and engaging said first and second cam members at selected rolling radii r_1 and r_2, respectively and further engaging said first and second reaction tracks at selected track rolling radii ρi, and $\rho 2$, respectively, as said rolling means orbit between said opposed cam members and tracks; and

retaining means engaged with said rolling means and rotatable between said opposed cam members and tracks;

said track rolling radii for said rolling means being selected in accordance with the relationship

$$\varphi_1 = \varphi_2 \frac{\cos \gamma_1 r_1}{\cos \gamma_2 r_2}$$

so that said rolling means orbit at substantially the same angular speed and roll freely along said first and second reaction tracks with substantially the same angular movement.

* * * * *