United States Patent

Maroth

[15] **3,653,789**

[45] **Apr. 4, 1972**

[54] **POWERFUL POSITIVE DISPLACEMENT RECIPROCATING PRESSURIZING DEVICE AND METHOD AND MEANS FOR CONTINUOUSLY VARYING THE PRESSURIZING STROKE**

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[52] **U.S. Cl.** **417/534**, 92/13.5, 92/13.51

[51] **Int. Cl.** **F04b 21/02**

[58] **Field of Search** 74/56, 60, 567; 92/31, 33, 92/13.5, 13.51; 417/534–537

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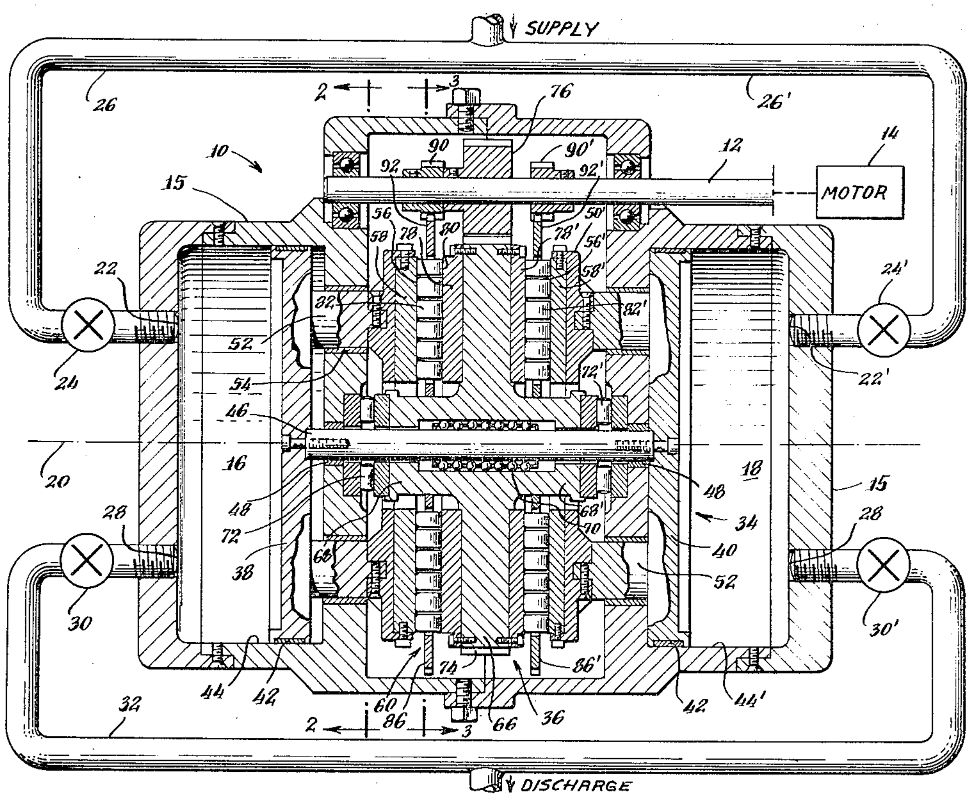
[57] **ABSTRACT**

A powerful positive displacement reciprocating pressurizing device is described for use as a liquid pump or gas compressor capable of generating immense fluid pressures with significant volume flow. A double piston pump is shown driven by a rotating input member with inclined planes operatively interposed between the rotating input member and the pistons. The inclined planes are located as undulations on cam surfaces with peaks and valleys. Antifriction elements driven by the rotating input member roll over the undulations to thereby actuate the pistons in a reciprocating manner. A powerful positive displacement reciprocating apparatus and method for pressurizing fluids with continuously variable strokes at any desired operating speed is disclosed.

14 Claims, 12 Drawing Figures



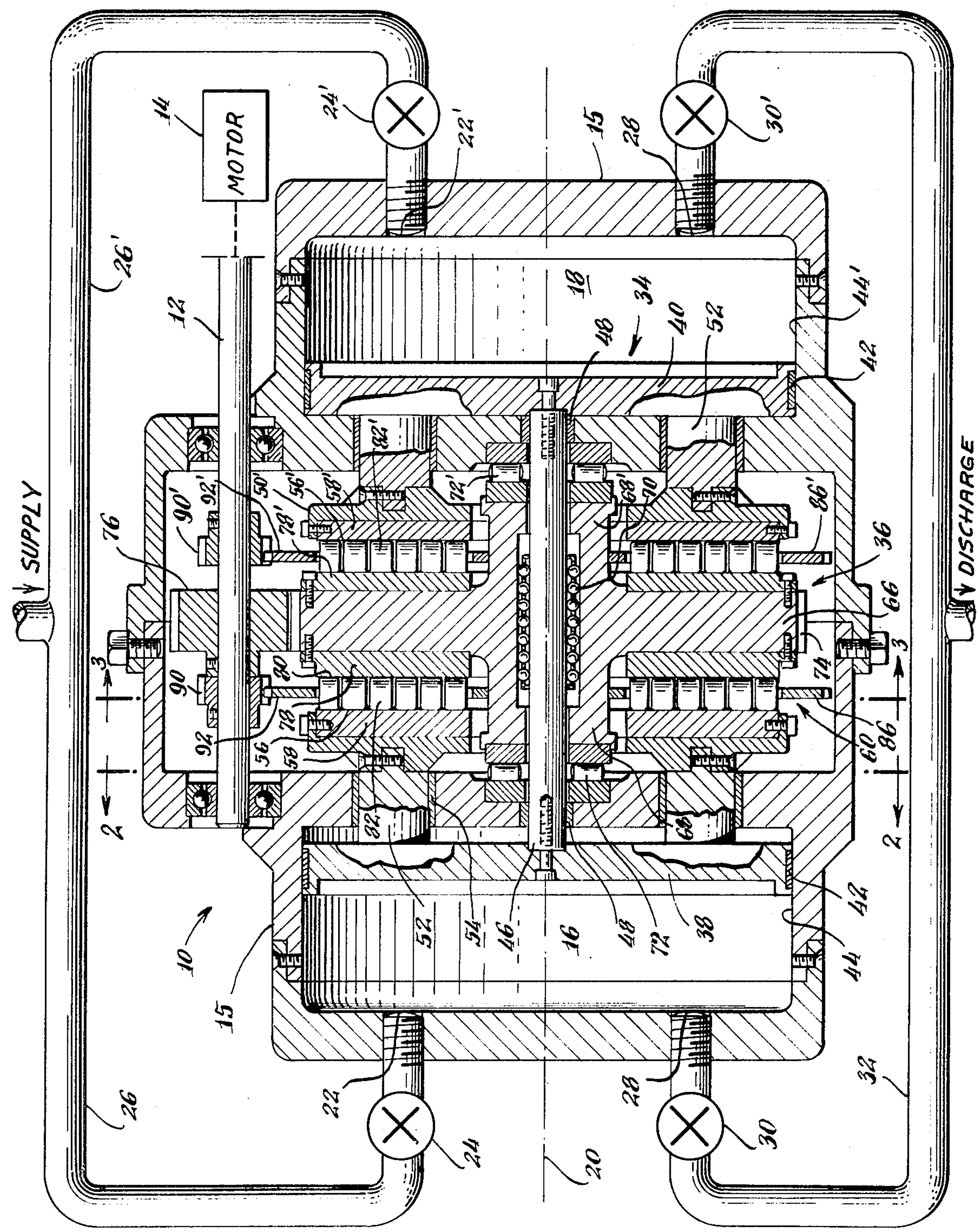


Fig. 1

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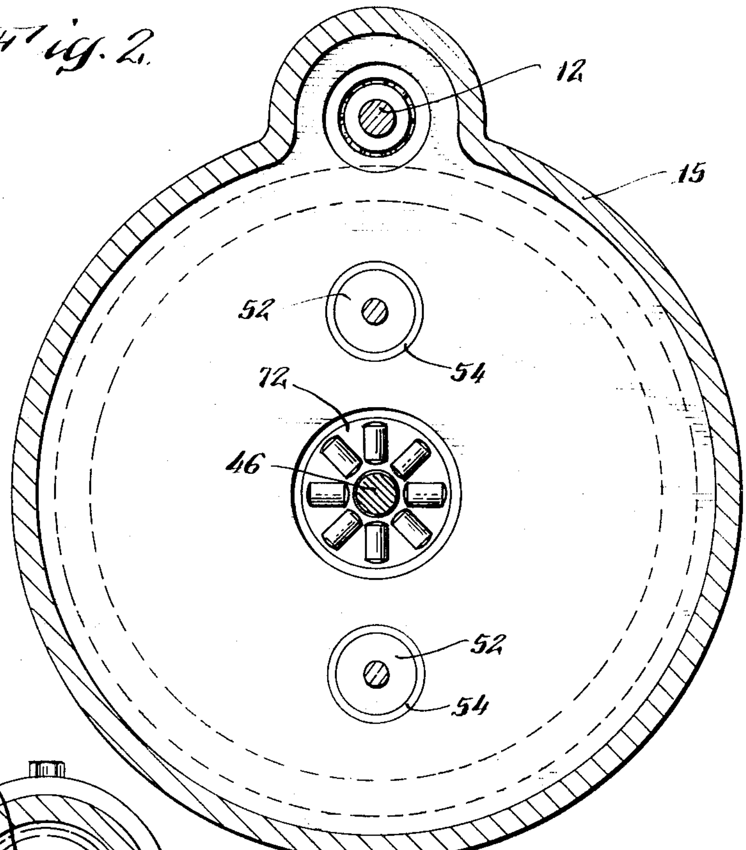
Fig. 2.



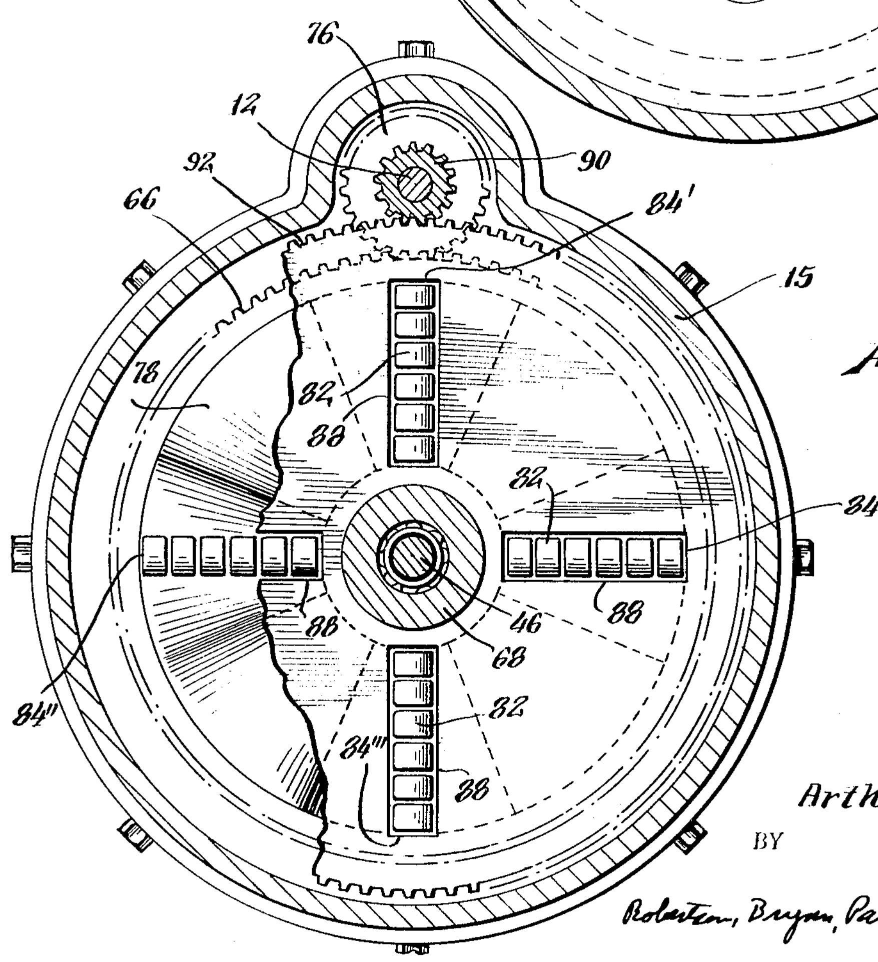


Fig. 3.

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Fig. 4.

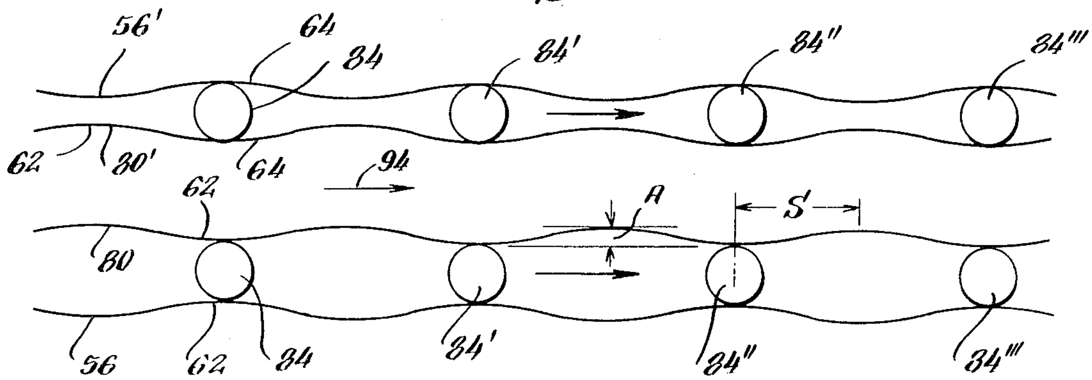

Fig. 5.

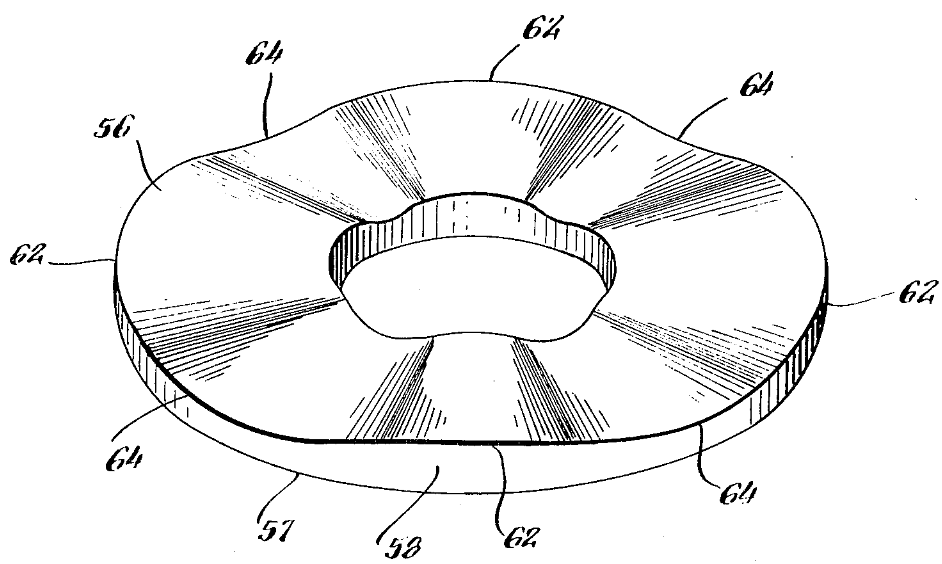

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Fig. 6.

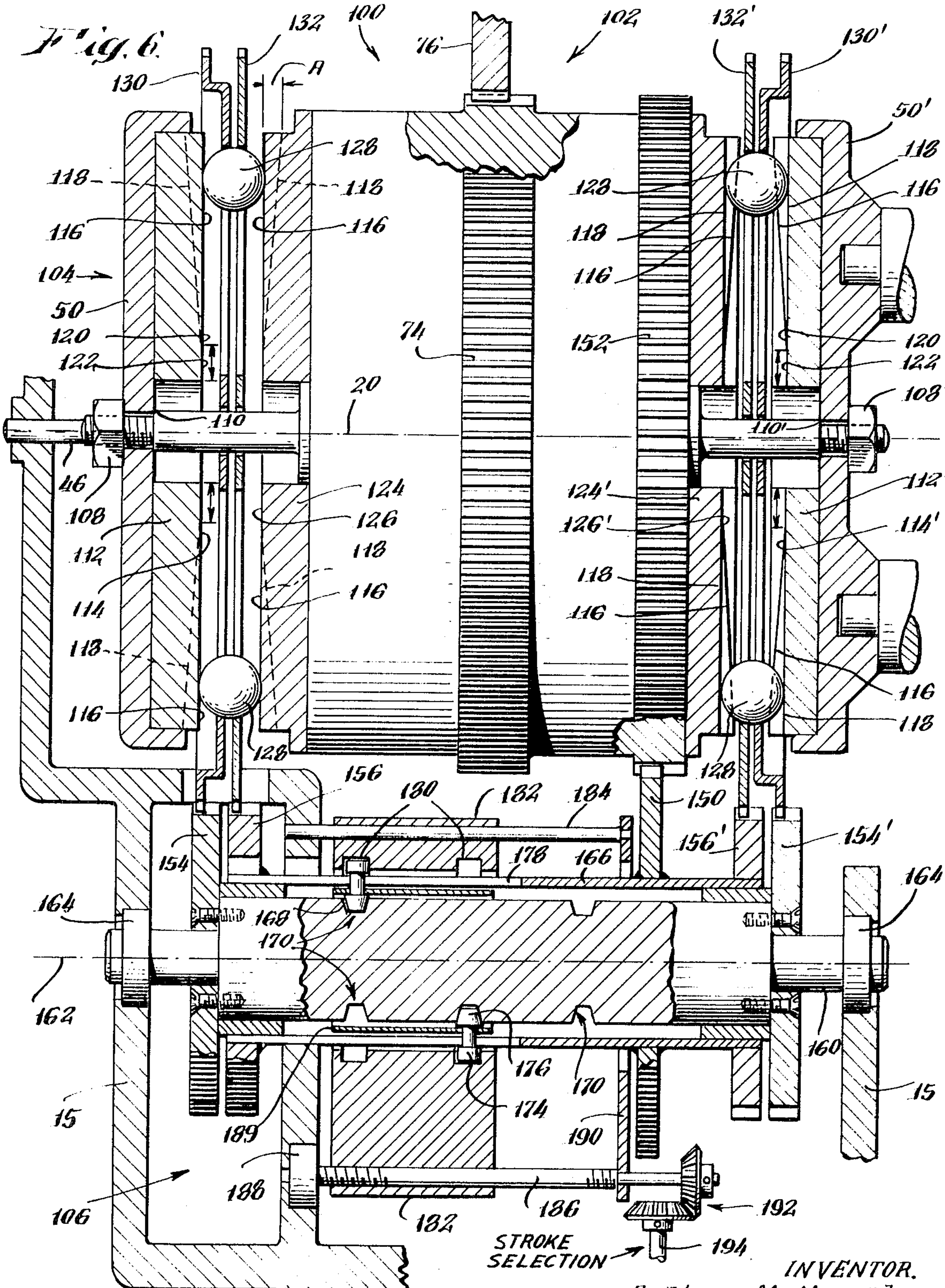

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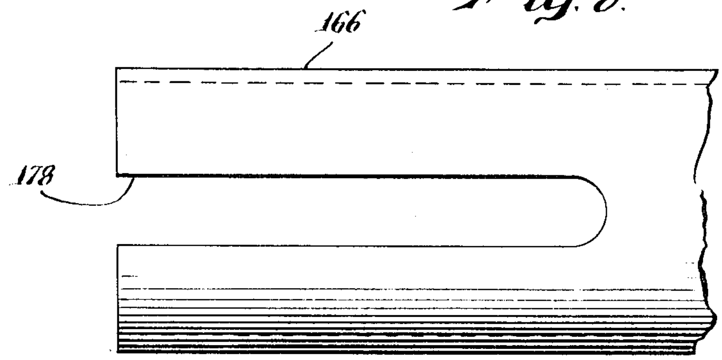
Fig. 8.



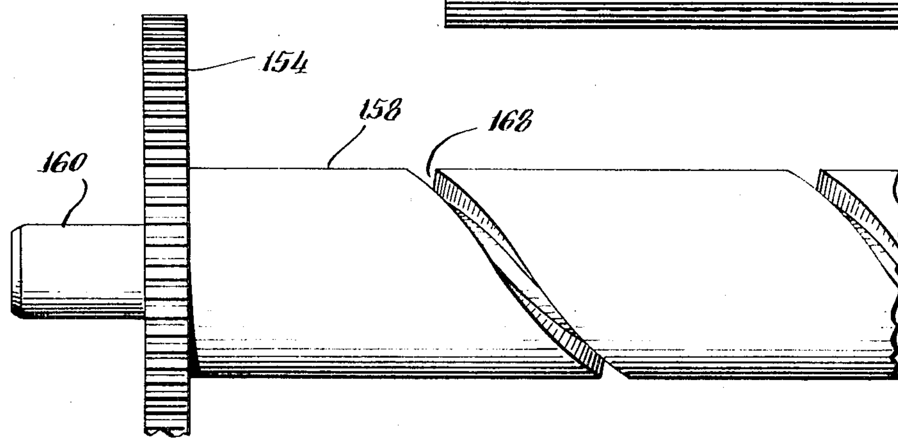


Fig. 7.

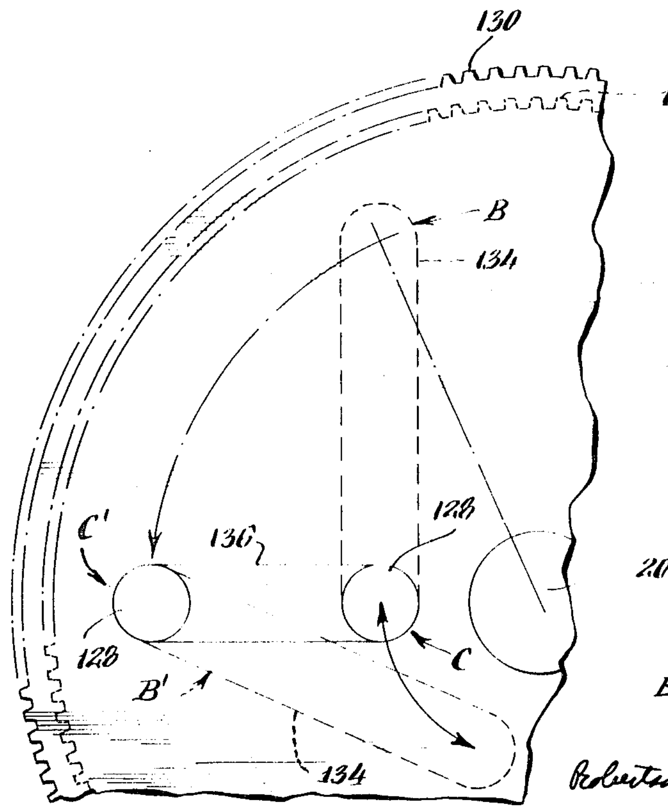


Fig. 9.

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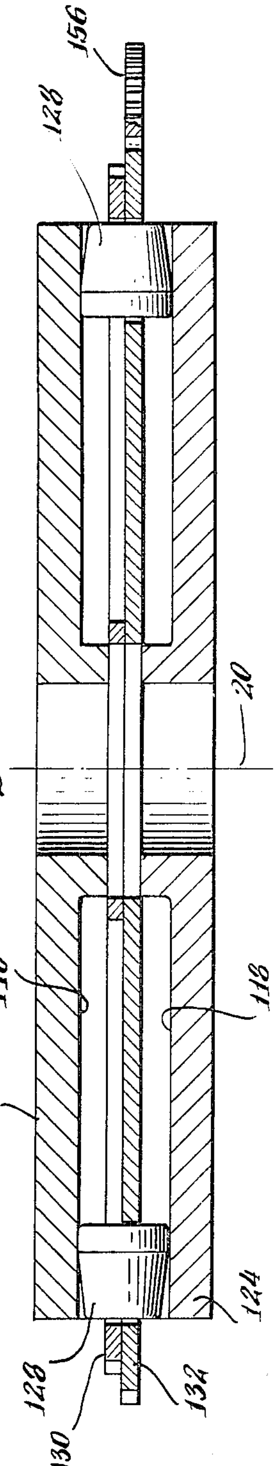

Fig. 10.

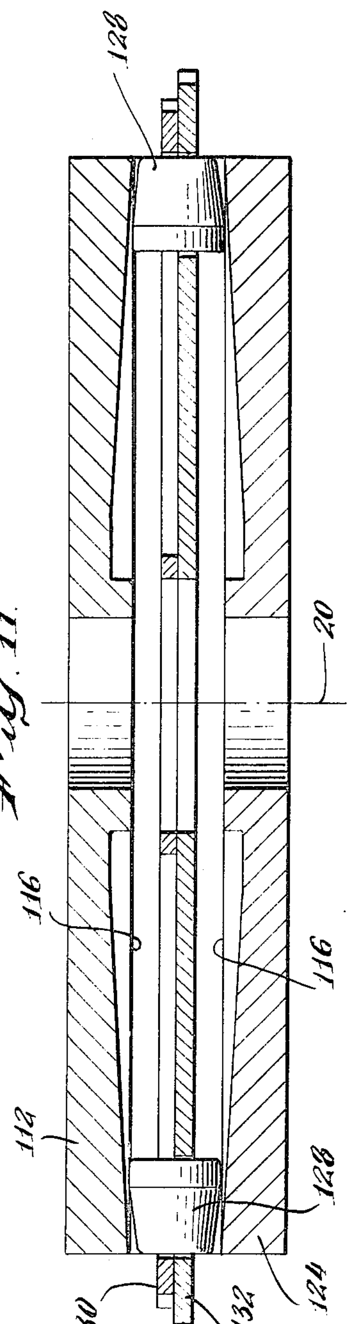

Fig. 11.

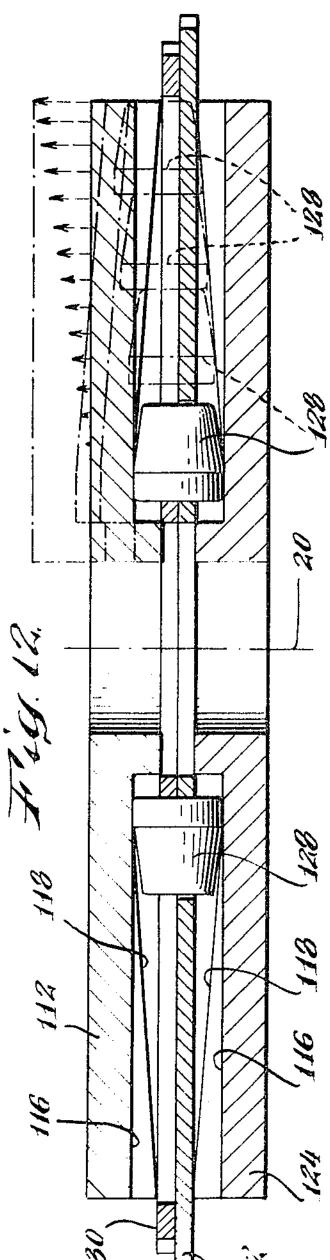

Fig. 12.

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POWERFUL POSITIVE DISPLACEMENT RECIPROCATING PRESSURIZING DEVICE AND METHOD AND MEANS FOR CONTINUOUSLY VARYING THE PRESSURIZING STROKE

This invention generally relates to a powerful positive displacement apparatus for converting mechanical kinetic energy to fluid potential energy. More specifically, this invention relates to a positive displacement reciprocating pump or compressor and method for pressurizing fluid.

This invention further relates to a positive displacement reciprocating pump or compressor having a continuously variable stroke at any desired operating speed.

A fluid pressurizing apparatus made in accordance with the invention converts a rotary mechanical input into powerful reciprocating strokes of a piston operatively located in a pressurizing chamber by employing the mechanical advantage afforded by inclined planes.

Inclined planes are operatively interposed between the rotary mechanical input and the piston with an orientation generally transverse to the reciprocating direction of the piston. Antifriction elements driven by the rotary input roll over the inclined planes to thereby actuate the piston and pressurize fluid located in the chamber. The inclined planes are shaped to provide a very high mechanical advantage to the rotary mechanical input. By suitably selecting the shape of the chamber and piston the apparatus may be used as a gas compressor or a liquid pump.

In a variable stroke pressurizing apparatus in accordance with the invention the inclined planes are effectively varied in amplitude in a preselected direction. Antifriction elements are controllably positioned along the preselected direction to correspondingly vary the piston stroke. A pump constructed with this feature is capable of providing huge pumping forces at any desired operating speed. Large input torques may be fully utilized at start up time. Input torques in fact an effectively significantly amplified by utilizing the high mechanical advantages available.

In a described preferred embodiment for a liquid pump utilizing the features of the invention powerful reciprocating strokes of a pair of mechanically connected pistons are obtained with a compact structure. Immense fluid pressures may be generated, yet with substantial fluid flow due to the excellent mechanical advantages provided by the inclined planes and the low friction high speed drive by antifriction elements.

It is, therefore, an object of this invention to provide a positive displacement powerful fluid pressurizing apparatus capable of producing high fluid flows at elevated pressures with a compact structure.

It is a further object of the invention to provide a compact light weight high pressure positive displacement reciprocating pump.

It is still further an object of this invention to provide a positive displacement powerful fluid pressurizing apparatus with a continuously variable stroke.

It is a further object of the invention to provide a method of varying the stroke of a recriprocating pressurizing apparatus.

These objects and others as well as many of the advantages of the invention may be understood from the following description of several embodiments of the invention in conjunction with the drawings wherein

FIG. 1 is a section view of a positive displacement reciprocating pump formed in accordance with the invention with the pump elements shown in an illustrative operative position for a selected instant in time;

FIG. 2 is a section view of the pump of FIG. 1 taken along the line 2—2 in FIG. 1;

FIG. 3 is a section view of the pump taken along the line 3—3 in FIG. 1;

FIG. 4 is a schematic view of a linear projection of the radial periphery of a cam employed with the pump of FIG. 1 and the position of antifriction elements at the instant of time selected for the view in FIG. 1;

FIG. 5 is a perspective view of a cam employed with the pump of FIG. 1;

FIG. 6 is a section view of a pump drive with a mechanism for continuously varying the piston stroke;

FIG. 7 is a partial side view of an element employed in the continuously variable stroke mechanism of FIG. 6;

FIG. 8 is a partial top view of a retainer employed in the apparatus shown in FIG. 6;

FIG. 9 is a partial axial plan view of a pair of retainer rings used with the variable stroke mechanism in FIG. 6.

FIGS. 10, 11 and 12 are sectional views of a pair of cams with varied inclined surfaces thereon to provide a variable stroke pressurizing operation and respectively show different stroke positions of antifriction elements located between the cams.

With reference to FIG. 1, a fluid pressurizing device 10 is shown having a rotating input member 12 driven by a motor 14. The pressurizing device is a pump formed of a generally cylindrical housing 15 having cylindrical fluid pressurizing chambers 16–18 oppositely spaced from each other symmetrically about a reciprocating axis 20. Chambers 16–18 are each provided with fluid inlet ports 22–22′ respectively connected through check valves 24–24′ and conduits 26–26′ to a supply of fluid (not shown). The supply of fluid may be a liquid or a gas depending upon whether the pressurizing device is used as a pump or as a compressor. Chambers 16–18 are further provided with fluid discharge ports 28–28′ effectively connected through check valves 30–30′ to a common discharge conduit 32.

The pump is composed of a piston structure 34 reciprocatingly mounted to housing 15 and a piston driving structure 36 rotatably mounted to housing 15. The piston structure is formed of pistons 38–40 operatively located in chambers 16–18 respectively and properly sealed with walls in the chambers to compress fluid therein. Thus, each piston is provided with an annular piston seal 42 engaging the cylindrical walls 44–44′ of the cylindrically shaped chambers. Pistons 38–40 are rigidly connected to one another by use of a reciprocating shaft 46 centrally connected at its ends to pistons 38–40. Pistons 38–40 are cylindrical bodies connected to each other with a piston reciprocating shaft 46 located concentrically with reciprocating axis 20 and the cylinder axis of the piston. Piston reciprocating shaft 46 slides with sealing relationship relative to housing 15 with sliding seals 48 located between shaft 46 and the housing to assure high pressure sealing of the chambers.

Attached to pistons 38–40 are like thrust rings 50–50′ mounted coaxially with reciprocating shaft 46. The thrust rings are each firmly attached with piston guides 52 to pistons 38–40. Piston guides 52 also slidingly engage housing 15 with slide seals 54 interposed to maintain high pressure pumping capability.

Piston structure 34 is further provided with cam surfaces 56–56′ respectively located on annular cams 58–58′. Annular cams 58–58′ are mounted on thrust rings 50–50 respectively and coaxially located with reciprocating axis 20. Cam surfaces 56–56′ are oriented to face one another across an annular space 60 sized to receive rotating piston driving structure 36. Cams 58–58′ and their cam surfaces 56–56′ move with pistons 38–40 in a reciprocating manner as will be further described.

Reciprocating cams 56–56′ and cam surfaces 58–58′ may be formed such as disclosed in my U.S. Pat. No. 2,836,985 issued on June 3, 1958. As disclosed in that patent cams are provided with undulations to form peaks and valleys. These undulations are circumferentially distributed over cam surfaces and coact with rotating cam surfaces and antifriction elements.

A cam 58, such as is preferably used in a pressurizing device in accordance with the invention, is perspectively illustrated in FIG. 5. Cam 58 is provided with a cam surface 56 on one axial side and a flat mounting surface 57 on the opposite axial side. Surface 57 is shaped to mount to thrust ring 50. Cam surface 56 has a plurality of circumferentially varying radially aligned undulations in the form of symmetrically sized like peaks 62 and like valleys 64. The number and amplitudes of these peaks and valleys are selected in view of factors such as the desired mechanical advantage, operating speed, piston stroke, pumping volume and the like. The cam shown in FIG. 5 has four peaks 62 and four valleys 64, though it should be realized that this number may vary as desired.

Both cams 58–58', shown in FIG. 1, have like undulations varying with equal amplitude along the reciprocating axis 20. Both cams have their cam surfaces generally transversely oriented with respect to the reciprocating axis 20.

Piston driving structure 36 is placed in gap 60 between cams 58–58'. Piston driving structure 36 is coaxially located with reciprocating axis 20 about which structure 36 is also mounted for rotation under action by input drive member 12. Piston driving structure 36 is formed of a disc 66 having centrally located hub sections 68–68'. Disc 66 is coaxially mounted for rotation about piston reciprocating shaft 46 by use of a universal bearing 70 accommodating, with low friction, rotation of disc 66 about shaft 46 as well as reciprocating axial motion by shaft 46.

Hubs 68–68' extend axially outwardly from disc 66 to effectively contact housing 15 through thrust bearings 71–72'. Hence, disc 66 is firmly axially locked between pistons 38–40, being only capable of rotation about axis 20. Disc 66 effectively serves as a second thrust ring for both pistons 38–40. Disc 66 is provided at its outer periphery with a ring gear 74 which operatively engages a gear 76 coaxially connected with rotating input driving member 12.

Disc 66 is further provided on its axial sides with a pair of annular cams 78–78' having cam surfaces 80–80' like cam surfaces 56–56'. Cam surfaces 80–80' are located opposite to and spaced from reciprocating cam surfaces 56–56' respectively to form pairs of reciprocating and rotating cam surfaces 56–80 and 56'–80'. Antifriction elements in the forms of sets of roller bearings 82–82' are operatively located between pairs of reciprocating and rotating cam surfaces to provide rolling contact between piston structure 34 and piston driving structure 36.

As illustrated in FIG. 3, the antifriction elements between each pair of cam surfaces are arranged in four angularly distributed rows 84, 84', 84'', and 84'''. The angular distribution of rows 84 is commensurate with the peaks or valleys, i.e., all rows of rollers are at the same time opposite either a peak 62 or a valley 64 or some other corresponding angular intermediate location. Hence, the axial reciprocation of a piston effected by any one set of rollers will be the same as that by the other sets of rollers. All the antifriction elements between each pair of reciprocating and rotating cam surfaces act jointly upon a piston in the same manner with an advantageous distribution of loads. Preferably as many sets 84 of antifriction elements are employed as there are pairs of peaks and valleys, although fewer sets may be used.

The angular position of each set 84 of rollers is maintained with the aid of flat retainer rings 86–86' provided with radially extending rectangular guide slots 88. The retainers 86–86' are located between pairs of reciprocating and rotating cam surfaces and their slots are shaped to freely enclose the antifriction elements.

During assembly of cams, antifriction elements and the retainers, the position of the antifriction elements between each pair of reciprocating and rotating surfaces 56–80 and 56'–80α⅔ is selected so that they will operatively contact the cam surfaces at points where opposing cam surfaces are continuously parallel to each other. As shown in FIG. 3 this is obtained in the embodiment by placing the sets of rollers at angular intervals of 90° between each pair of rotating and reciprocating surfaces 56–80 and 56'–80'. Furthermore, the effective cam surface contact of antifriction elements when compared between the pairs of opposing cam surfaces is so angularly shifted during assembly that all antifriction elements have equal axial motions in the same direction. Hence, antifriction elements between one pair of opposing cam surfaces 56–80 may be ascending a peak 62 while the elements between the other pair of cam surfaces 56'–80' will be descending into a valley 64. With equal axial motions of the antifriction elements, pistons 38–40 are moved in unison in a reciprocating manner.

One may appreciate that once cam surface contact relationships of antifriction elements have been established variations are to be avoided to prevent binding of the apparatus. For this reason, retainer rings 86–86' are both rotationally driven, with little power, by input member 12 at speed which is equal to the rotational speed of rollers 82 about axis 20, i.e., one half the rotational speed of disc 66. Rotation of retainer rings 86–86' is obtained by providing input member 12 with pinions 90–90' which operatively engage gear teeth 92 located around the outer periphery of retainer rings 86–86'. Pinion gear teeth and retainer ring gear teeth are selected to provide the desired retainer ring rotation at one half the speed of disc 66.

The sectional view of FIG. 1 illustrates the roller bearings 82 for the left pair of opposing cam surfaces 56–80 located between peaks whereas the roller bearings 82' in the right pair of opposing cam surfaces 56'–80' are located between valleys. In this manner like reciprocating axial motions of the roller bearings and thus pistons 38–40 is obtained.

In the operation of the pump 10 assume that the relationship of the several sets 84 of antifriction elements between the pairs of cam surfaces 56–80 and 56'–80' are as illustrated in FIG. 4. Input member 12 is rotating counterclockwise, thus causing cam surfaces 80–80' to rotate clockwise as shown by arrow 94 in FIG. 4. Clockwise rotation of cam surfaces 80–80' produces a like rotation of roller sets 84 about axis 20 but at half the speed. Thus between cam surfaces 56–80, all the roller sets move from a peak towards a valley while the roller sets between cam surfaces 56'–80' move from a valley to a peak.

When the described cam surface motions are viewed in relation to the operation of the pump 20 as shown in FIG. 1, piston 40 commences a pressurizing stroke of fluid in chamber 18 to discharge fluid through check valve 30' while piston 88 commences a fluid stroke to take in fluid through check valve 24.

The direct interconnection of the pistons provides a powerful and accurate piston stroke in both reciprocating directions. This, a double acting piston structure may be advantageously employed. For instance, each piston 38–40, or one of them alone may have a power stroke in both reciprocating directions. If a single piston is employed it may be operated in a double acting manner in its cylinder.

The mechanical advantage provided by relatively low ratios of peak to valley amplitude undulations A (See FIG. 4) to peak to valley angular spacing S (as measured around the radially outer perimeter, See FIG. 4) imparts immense pumping powers to the apparatus. Typically, an annular cam having a 4 inch inner cam surface diameter and an 8 inch outer cam surface diameter may be employed with four uniformly angularly distributed circumferentially varying undulations composed of pairs of peaks and valleys having an amplitude A of about an eighth of an inch or even less. The large mechanical advantage obtained with such low angled inclined surfaces (angles of the order of one degree being usable) enables the use of large piston surface areas to yield significant pumping volumes at practically realizable speeds with large pumping pressures.

The total piston stroke is determined by twice the amplitude A between peaks 62 and valleys 64. As mentioned, large working areas of pistons may be employed. For instance, a piston of 12 inch diameter may be conveniently used with, for example, like diameter cams having peaks and valleys of an amplitude of one-sixteenth of an inch. The total stroke of such piston would be twice this amplitude, since both rotational and reciprocating cam surfaces 56–80 contribute to the axial motion. Hence, the ratio of piston working area to piston stroke length (a convenient figure of merit) would be about 3,600 to one.

Smaller pistons with working areas of 2 inch diameters (12.5 square inches) and a stroke of ⅛ inch (peak to valley amplitude A of 1/16 inch) may be used in which case the piston working area to stroke ratio would be approximately

100 to one. These ratios are selected in view of factors such as desired operating speed, pumping volume and valve cycling.

Cam plates such as 58–58' are not limited in size to the diameter of the piston working surface. For instance, it may be desirable to employ large diameter cams with smaller diameter pistons to obtain a very powerful high mechanical advantage operation with a substantial stroke. In such case, a convenient figure of merit may be the cam circumference divided by the stroke. Generally figure of merit ratios greater than about 50 to 1 are within operating ranges for a pressurizing device in accordance with the invention.

FIG. 6 illustrates a pressurizing drive 100 for a single ended continuously variable stroke liquid pump. The pump however is not completely shown with its piston and compression chamber deleted for clarity. Pump drive 100 is shown for operation with a single pumping piston, though it should be realized that a double acting pump as depicted in FIG. 1 may be used. Also deleted for clarity from the view in FIG. 6 is the drive coupling from a rotating input member such as 12 in FIG. 1. A portion of the driving gear 76 is shown to indicate that pump drive 100 is driven by a rotating input.

The continuously variable stroke reciprocating pump of FIG. 6 utilizes a rotating piston driving structure 102, a reciprocating piston structure 104 and a stroke selecting device 106.

Piston structure 106 is similar to piston structure 34 in that a reciprocating shaft 46 rigidly connects to and spaces thrust rings 50–50' with nuts 108–108' engaging threaded ends of shaft 46 and clamping thrust rings 50–50' against annular shoulders 110–110' on shaft 46. Annular reciprocating cams 112–112' are mounted on thrust rings 50–50' and have annular cam surfaces 114–114' of a selected shape different from the cams employed in FIG. 1.

Cam surfaces 114–114' are each provided with a plurality of circumferentially varying radially aligned undulations in the form of symmeterically located like peaks 116 and like valleys 118.

The amplitude A between a peak 116 and an adjacent valley 118, however, varies in a radial direction. In the embodiment shown in FIG. 6 the peak to valley amplitude is a maximum at the radially outer edge of cam surfaces 114–114' and gradually reduces to zero along an annular radially inner line 120. For a small radial distance 122 the peak to valley amplitude is zero and corresponds as will be explained to a zero stroke of the pump piston.

The peak to valley amplitude in FIG. 6 is shown to vary linearly as a function of the radial distance from axis 20. Other functional relationships, however, can be established as desired to suit different pump or pressurizing stroke characteristics.

Piston driving structure 102 fits between reciprocating cams 112–112' and is rotationally mounted over reciprocating shaft 46 as in the embodiment of FIG. 1. Driving structure 102 is driven into rotation by gear 76 connected to a rotating input member (not shown).

Piston driving structure is provided with cams 124–124' spaced opposite cams 112–112' respectively. Cams 124–124' are provided with cam surfaces 126–126' like those on cam 112–112' i.e. with radially varying amplitudes between peaks 116 and valleys 118.

Between pairs of cam surfaces 114–126 and 114'–126' are antifriction elements 128 which operatively roll over the latter pairs of surfaces. Antifriction elements 128 are angularly distributed about axis 20 with a single element 128 provided for each set of a peak and a valley. The elements 128 are further so angularly spaced that they are operatively located between corresponding portions of a set of a peak and valley for each set of opposing cam surfaces. Thus, as shown in FIG. 6, the antifriction elements between cam surfaces 114–126 are each contacting opposing peaks while all antifriction elements 128 between cam surfaces 114'–126' are each contacting opposing valleys. This arrangement of the antifriction elements is similar to that described for the embodiment of FIG. 1.

As may be understood from the embodiment of FIG. 1 each of the antifriction elements 128 have an angular freedom of movement about axis 20. In addition each element 128 is provided with a controlled radial freedom of movement. Radial movement of elements 128 brings them into contact with those portions of the cam surfaces having different amplitudes between peaks and valleys.

One may appreciate that when all elements 128 are located as shown in FIG. 6 the piston stroke obtained equals twice the amplitude of that portion of peaks and valleys in effective contact with elements 128. As elements 128 are all moved simultaneously in a radially inwardly direction, the stroke is decreased until it is zero when elements 128 have been brought into operative contact with flat annular surface portions 122.

The formation of cams 112–112' and 124–124' with their cam surfaces may be obtained with a stamping operation known as coining. The peaks 114 on the pair of opposing cams 112 and 124 are all located in a common "peak" plane with flat annular surface portions 122 with this peak plane extending generally transverse from axis 20. The valleys 118 of cams 112–124, however, are located on a conical surface of a large solid angle cone which is intersected by the peak plane. Note that cam surfaces 114'–126' so shaped that their valleys 118 lie in a common"valley" plane generally transverse to axis 20 and that the peaks 116 of cam surfaces 114'–126' are located on a conical surface of large subtended angle equal to that for the conical surfaces on which valleys 118 of cams 112 and 124 are located. Further note that peaks 116 on cam 112' are parallel with valleys 118 on cam 112 and that peaks 116 on cam 124' are parallel with valleys 118 on cam 124.

Radial position control of antifriction elements 128 may be accomplished even while elements 128 are rotating at high operating speeds. This radial control is obtained with pairs of antifriction element retainers (See also FIG. 9) 130–132 and 130'–132' and stroke control mechanism 106. The retainers are each provided with slots such as 134 and 136 each of which captures an antifriction element 128. Each antifriction element 128 is retained by a pair of overlapping slots 134–136 respectively formed in retainers 130–132.

Radial positioning of elements 128 is obtained as follows. Assume that retainer 132 is angularly located with respect to retainer 130 so that its slot 134 is at the indicated position B in FIG. 9. As a result the overlap of slots 134 and 136 is so radially located that antifriction element 128 is at radially inward position C. When retainer 132 is now rotated counterclockwise with respect to retainer 130 slot 134 is rotated to the position at B' and the slot overlap now occurs at a radially outward position to correspondingly move antifriction element 128 to position C'.

In like manner the other antifriction elements 128 are radially positioned so that rotation of retainer 130–132 relative to each other simultaneously radially moves all antifriction elements 128 between opposing cam surfaces and thus controls the magnitude of the stroke of the piston.

As illustrated in FIG. 9 the radial Peripheries of retainers 130–132 are provided with gear teeth so they may be driven at one half of the rotational speed of driving structure 102. Stroke control mechanism 106 provides both the desired rotational drive to retainers 130–132 as well as the angular displacement therebetween for stroke control.

Stroke control mechanism 106 essentially is like the speed control mechanism disclosed in a copending patent application of mine entitled MECHANICALLY INTERLOCKED INFINITELY VARIABLE TRANSMISSION, Ser. No. 813,019 and filed on Apr. 3, 1969. The copending application describes the speed control mechanism and how it obtains a selected angular displacement between a pair of rotating members.

Returning to FIG. 1, a gear 150 is shown in meshing relationship with a gear 152 on rotating piston driving structure 102. Gears 154–154' mesh with the peripheral gears on retainers 130–130' respectively and gears 156–156' mesh with the peripheral gears on retainers **132–132'** respectively. Gears **150, 152** and those engaging the retainers are so selected that the retainers are rotated at one half the speed of rotating piston driving structure **102**.

Gears **154–154'** are coaxially mounted with a cylindrical control member **158**. Control member **158** is affixed to a shaft **160** mounted for rotation about an axis **162** to housing **15** with bearings **164**. Gears **154–154'** are shown connected to axial ends of control member **158**.

Gears **156** are coaxially affixed to axial ends of a cylindrical sleeve **166** which in turn is coaxially mounted with control member **158**. Sleeve **166** may be angularly displaced with respect to control member **158** so that gears **156–156'** may be also angularly displaced relative to gears **154–154'**. Gear **150** is affixed to sleeve **166** to cause a rotation thereof in response to rotation of structure **102**.

As depicted in FIG. 7 control member **158** is provided with a helical groove **168** which commences at an axial end to permit assembly of stroke control **106**. Specially adapted antifriction elements **170** are provided to ride in helical groove **168**.

These antifriction elements **170** are each formed of an upper cylindrical rolling element **172**, a central cylindrical section **174** and a lower conically shaped roller element **176** sized to roll on a side of groove **168**. The central cylindrical sections **174** of elements **170** fit within axially extending slots **178** of sleeve **166**. FIG. **8** illustrates a slot **178** in sleeve **166**.

The cylindrical rolling elements **172** engage radially inwardly facing grooves **180** formed in a control ring **182** and located in planes generally transverse to axis **162**. Hence, when sleeve **166** is rotating at high speed it also rotates elements **170** by means of the sleeve engagement with central cylindrical segments **174**. Furthermore, rotation of elements **170** also causes a rotation of control member **158** whose helical groove **168** is engaged by conical elements **176**. Control ring **182** does not rotate about axis **162**, but is mounted for axial movement by means of slide rods such as **184** and a control screw **186** supported by housing **15** by a bearing **188**. A retainer ring **189** is employed to retain antifriction elements **170** in desired operative locations.

Control screw **186** is threaded for a major portion thereof and its thread meshes with a threaded bore **188** in control ring **182**. The control screw **186** and slide rods **184** are supported by a stationary ring **190** with the control screw being rotationally mounted therewith and extending therethrough for attachment to a miter gear arrangement **192**. A control shaft **194** extends from miter gear arrangement **192** to provide a stroke selection described as follows in conjunction with FIGS. **1, 10, 11** and **12**.

As shown in FIG. **10** the antifriction elements **128** are at a radially outward location and are illustrated when in contact with valleys **118** on cams **112** and **124**. In FIG. **11** these rollers are shown in contact with peaks **116** on cams **112** and **124**.

With antifriction elements **128** shown in FIGS. **10** and **11** in the maximum stroke position, assume that the relative angular displacement of retainers **130–132** is such that location of antifriction elements **128** requires control ring **182** to be axially positioned as shown in FIG. **1**.

A stroke reduction may be obtained by rotating control shaft **194** in a direction that will cause rotation of screw **186** and axial movement of control ring **182** towards the right side in the drawing of FIG. **1**. Axial motion of control ring **182** forces antifriction elements **170** to move in like direction and drives conical element portions **176** against the side of groove **168**. This latter action of element portions **176** forces control member **158** to rotate out of the way by an amount sufficient to accommodate axial motion of ring **182**. Rotation of control member **158** in turn produces a rotational displacement of retainers **130–130'** with respect to retainers **132–132'** so that antifriction elements **128** are moved radially inwardly towards the zero stroke cam surface regions **122** as depicted in FIG. **12**.

The continuous stroke variation obtainable with the drive of FIG. **6** is depicted graphically with curve **200** in FIG. **12**.

Having thus described a reciprocating pressurizing apparatus and a method and apparatus for continuously varying the stroke thereof its numerous advantages may be appreciated. Enormous pressures may be generated with rotational input drive and with high mechanical advantages. A maximum torque available from a prime mover may be fully utilized for all flow conditions of a pressurizing device of this invention.

What is claimed is:

1. In a powerful fluid pressurizing device of the positive displacement reciprocating type the improvement comprising

a housing having a cylinder for pressurizing fluid and a piston member operatively located for reciprocation in the cylinder for pressurizing fluid therein,

a pair of generally parallel mounted cams having like cam surfaces facing each other, antifriction elements located between the cam surfaces and continuously in operative contact therewith, one of said cams being effectively mounted to the piston to cause an advance thereof in the cylinder and the other of said cams being mounted for rotation about an axis relative to said one cam to roll said antifriction elements over said cam surfaces and axially advance said one cam in a first direction, said cam surfaces being provided with undulations having peaks and valleys determinative of said piston advance and sized to advance said piston with great mechanical advantage,

a second pair of generally parallel mounted cams having like cam surfaces as said first cam surfaces, said second cam surfaces being spaced from one another, antifriction elements mounted between said second cam surfaces, one of said second cams being effectively mounted to the piston with an orientation placing said one second cam surface operatively opposite to that of the first cam surface, with the other second cam being mounted for rotation with the other first cam, with the antifriction elements between the second pair of cam surfaces being selectively angularly located to effect a driven return of said piston in a direction opposite to the first direction upon completion of a piston advance by the antifriction elements located between the first pair of cam surfaces.

2. In a powerful fluid pressurizing device of the positive displacement reciprocating type driven by a rotating input member the improvement comprising

reciprocating means for pressurizing fluid, said reciprocating means including first and second reciprocating cam surfaces operatively oriented to respectively produce opposite driven motions of the reciprocating means along a reciprocating axis for a bidirectional drive,

rotating means driven by the input member for producing first and second rotating cam surfaces respectively spaced from and generally parallel to the reciprocating cam surfaces to form oppositely acting first and second pairs of cam surfaces with said reciprocating cam surfaces, antifriction elements operatively located between the facing rotating and reciprocating first and second pairs of cam surfaces to effect powerful reciprocation of said reciprocating means, each of said cam surfaces being provided with undulations formed of symmetrical peaks and valleys having angular spacings and axial amplitudes selected to generate a high mechanical advantage bidirectional reciprocating drive of said reciprocating means upon rotation of said first and second rotating cam surfaces.

3. The power fluid pressurizing device as claimed in claim **2**, wherein said reciprocating means further includes

a housing having a pair of chambers for pressurizing fluid and effectively aligned along the reciprocating axis, and a pair of pistons individually operatively located in said chambers to compress fluid therein, said pistons being rigidly connected to one another to move in unision with one another, each of said pistons being provided with one of said reciprocating cam surfaces oriented to advance the connected pistons in opposite directions.

4. The powerful fluid pressurizing device as claimed in claim 3 wherein the chambers are located at spaced opposite locations along the reciprocating axis, a reciprocating shaft coaxially aligned with the reciprocating axis to rigidly interconnect said pistons, a pair of annular cams each coaxially mounted with the reciprocating axis to a piston and each cam having a reciprocating cam surface facing the other cam surface with a predetermined axial spacing,

a rotating disc mounted for rotation about the reciprocating shaft and located between the reciprocating cam surfaces, said rotating disc being axially locked to the housing to transmit axially directed thrusts during operation of the pressurizing device, said rotating disc being provided on axial sides with annular cams each mounted coaxially with the axis of rotation of the disc and each cam having a rotating cam surface axially facing a reciprocating cam surface across a spacing selected to receive the antifriction elements, the antifriction elements being angularly located to enable said pistons to reciprocatingly move in unision.

5. A compact powerful positive displacement pump providing substantially continuous fluid flow from a source of fluid into a common discharge conduit and being effectively actuated by a continuously rotating driving member comprising

a housing having first and second spaced fluid compression chambers each provided with inlet ports coupled to the source of fluid and outlet ports coupled to the common discharge conduit,

first and second pistons reciprocatingly mounted to the housing and respectively located in the first and second chambers to compress fluid therein, said pistons being rigidly connected to one another with preselected operative relationship wherein fluid compression by one chamber is accompanied with fluid intake in the other chamber,

first and second reciprocating cams respectively connected to the first and second pistons to move therewith, said first and second reciprocating cams being operatively mounted to cause opposite driven movements of the connected first and second pistons, each of said cams being provided with annular cam surfaces oriented generally transversely to the reciprocating directions of the pistons, each of said cam surfaces being provided about an axis with annularly varying symmetrical undulations alternately composed of a peak and valley,

first and second rotating cams spaced rotatably mounted to the housing and respectively disposed opposite the first and second reciprocating cams, said rotating cams each having a cam surface facing the reciprocating cam surfaces and being provided with undulations composed of smoothly interconnected peaks and valleys similar to those on the reciprocating cam surfaces,

antifriction elements located between facing cam surfaces and,

means for interconnecting the driving member to the rotating cams to drive said cams into rotation and cause said antifriction elements to roll over said cam surfaces and provide a bidirectional drive of the pistons.

6. An apparatus for varying the stroke of a reciprocating member comprising

generally parallel mounted cams having cam surfaces facing one another, said cam surfaces being provided with like undulations having peaks and valleys varying in amplitude in a selected direction, antifriction elements located between the cam surfaces and continuously in operative contact therewith, one of said cams being effectively mounted to the member to cause an advance thereof and the other cam being mounted for rotation relative to said one cam to roll said antifriction elements over said cam surfaces and undulations,

means for controlling the position of the antifriction elements along said selected direction over the cam surfaces to correspondingly vary the magnitude of the stroke of the reciprocating member, and

means for returning said reciprocating member to complete reciprocating motion thereof.

7. An apparatus for varying the stroke of a reciprocating member comprising

first and second pairs of opposing spaced cam surfaces, one cam surface in each pair being coupled to the member to control reciprocation thereof along an axis, the other cam surface in each pair being mounted for rotation relative to said one cam surfaces, said cam surfaces being provided with at least three undulations in the form of circumferentially distributed peaks and valleys with the peak to valley amplitude being varied along a radial direction, antifriction elements mounted between the pairs of opposing cam surfaces to roll thereover upon rotation of said other cam surfaces and

means for controlling the radial position of said antifriction elements along said radial direction to vary the stroke of said reciprocating member.

8. An apparatus for pressurizing fluid and varying the fluid flow therefrom comprising a housing having a piston therein for pressurizing fluid with reciprocating motion along an axis,

first and second pairs of opposing spaced cams having facing cam surfaces provided with undulations in the form of peaks and valleys with the peak to valley amplitude varying along a selected direction from a minimum stroke determining amplitude level to a maximum stroke determining level, one cam in each pair being operatively coupled to the piston and oriented to cause reciprocating motion thereof and the other cams in each pair being mounted for rotation relative to the reciprocating cams, first means located between the pairs of cams to couple said reciprocating and rotating cams to one another,

means for controllably varying the effective location of the first means between the minimum stroke determining amplitude level and the maximum stroke amplitude determining level of the cam surfaces to control the magnitude of the stroke of the reciprocating piston.

9. The fluid pressurizing apparatus as claimed in claim 8 wherein the rotating cams are mounted for rotation about an axis coaxial with the reciprocating axis, ans wherein the cam surfaces peak to valley amplitudes vary in magnitude in a generally radial direction.

10. The fluid pressurizing apparatus as claimed in claim 9 wherein the means between cam surfaces comprises antifriction elements angularly distributed as a function of sets of peaks and valleys.

11. The fluid pressurizing apparatus as claimed in claim 10 wherein the stroke varying means includes

first and second pairs of antifriction element retainers, a pair of retainers being located between each pair of facing cam surfaces with the retainers being mounted for rotation about the axis, each pair of retainers being provided with slots oriented to overlap at a selected radial location dependent upon the angular displacement between retainers in each pair, with an antifriction element captured by both overlapping slots at their overlap,

means for driving said retainers in rotation at one half the speed of the rotating cams, and

means for angularly displacing retainers relative to each other in each pair to correspondingly vary the radial position of the antifriction elements by the radial shift of the slot overlap of adjacent retainers.

12. The fluid pressurizing apparatus as claimed in claim 11 wherein the means for driving said retainers into rotation and said retainer angular displacing means include

a control member rotatably mounted to the housing about an axis and provided with a helical groove coaxial with said latter axis, said control member being coupled to one retainer in each pair to cause rotation thereof,

a control sleeve coaxially mounted with the control member and coupled to the other retainer of each retainer pair to cause rotation thereof and means coupling the helical groove of the control member with the control sleeve to obtain like rotation therebetween, means for axially moving said coupling means along the axis of rotation of the control member to cause an angular displacement between the control sleeve and the control member representative of the desired angular displacement between retainers in each pair.

13. The fluid pressurizing apparatus as claimed in claim 12 wherein the axial moving means includes a control ring coaxially mounted with the control member and control sleeve and operatively coupled by the coupling means, said control ring being movable along the axis of rotation of the control member for axial shifting of the coupling means by an amount representative of the desired angular displacement between retainers in each pair.

14. A set of cams for use in reciprocating pressurizing device comprising first and second pairs of annular shaped cams having like cam surfaces each of which is provided with at least three undulations formed of peaks and valleys distributed annularly about an axis with the height between adjacent peaks and valleys being varied in a predetermined manner along a radial direction, the peaks of cams in a first pair each lying in a common peak plane, and with the valleys of the cams in the second pair each lying in a common plane which is parallel with the common peak plane upon a coaxial mounting of the cams, and with the valleys of said cams in the first pair and the peaks of cams in the second pair each located on inclined surfaces which are respectively parallel with one another upon a coaxial mounting of the cams.

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