United States Patent [19]

Flynn et al.

[11] **3,816,032**

[45] **June 11, 1974**

[54] **METERING PUMP**

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[73] Assignee: **Hills McCanna Company,** Carpentersville, Ill.

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[21] Appl. No.: **288,532**

[52] **U.S. Cl.** **417/388,** 417/389, 417/478

[51] **Int. Cl.** **F04b 35/02**

[58] **Field of Search** 417/383, 384, 385, 386, 417/387, 388, 389, 478

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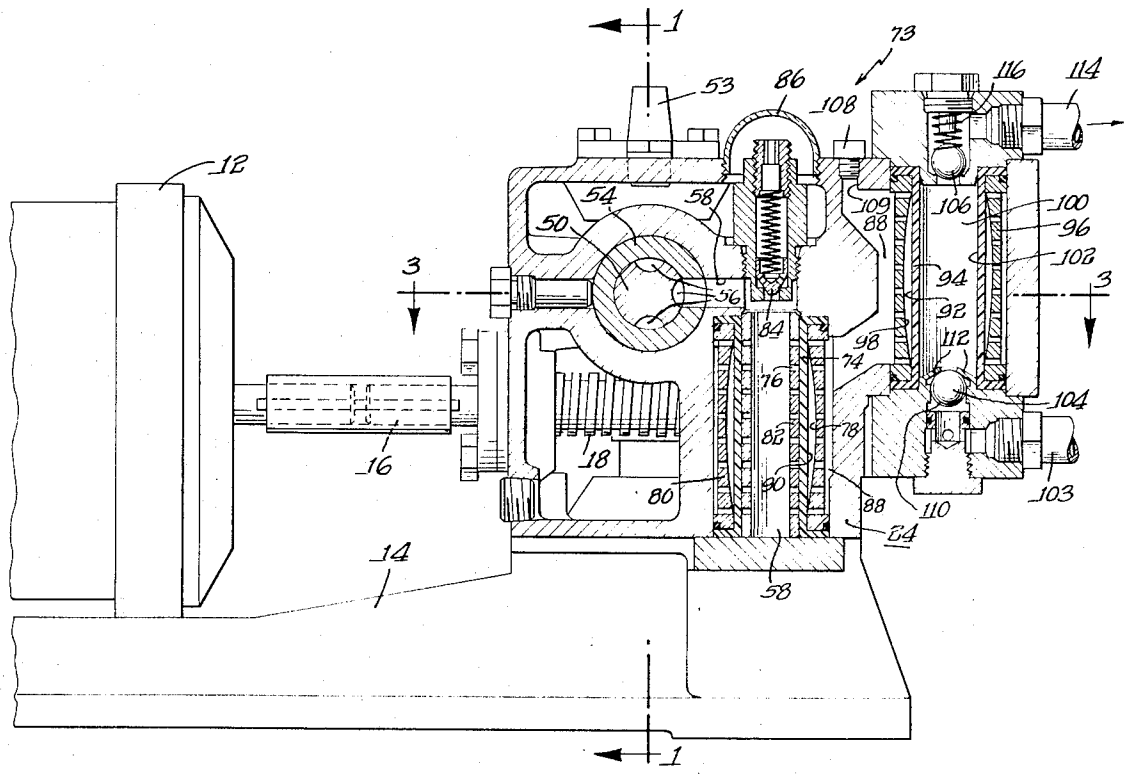
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Primary Examiner—Carlton R. Croyle
Assistant Examiner—Richard Sher
Attorney, Agent, or Firm—Olson, Trexler, Wolters, Bushnell & Fosse

[57] **ABSTRACT**

A pump has a reservoir containing a supply of driving fluid. Mounted within the reservoir is a drive mechanism for reciprocating a piston within a cylinder. The drive mechanism has an eccentrically mounted member which is slidably surrounded by a sliding block, the latter being enclosed by a cross-head which is connected to the piston and confined so as to reciprocate within the pump. Thus, roary motion is converted to oscillating motion in the sliding block, and the oscillating motion is then converted to reciprocating motion in the cross-head, the reciprocating motion then being transmitted to the piston. Two elastomeric tubes have their ends mounted within the pump so as to permit distortion (expansion or compression) of the central portions of the tubes. In the preferred embodiment of the invention, a displacement chamber is defined between the piston and the inner surface of the first tube, and an intermediate chamber is defined between the outer surfaces of the tubes, the intermediate chamber containing an intermediate driving fluid. As the piston reciprocates, driving fluid alternately passes into and out of the displacement chamber to cause the first tube to alternate between a first configuration, and a second configuration, which in turn causes the second tube to do likewise via the intermediate driving fluid. As a result of the alternating configurations assumed by the second tube, process fluid is pumped through the second tube. Other modifications are also disclosed.

11 Claims, 15 Drawing Figures



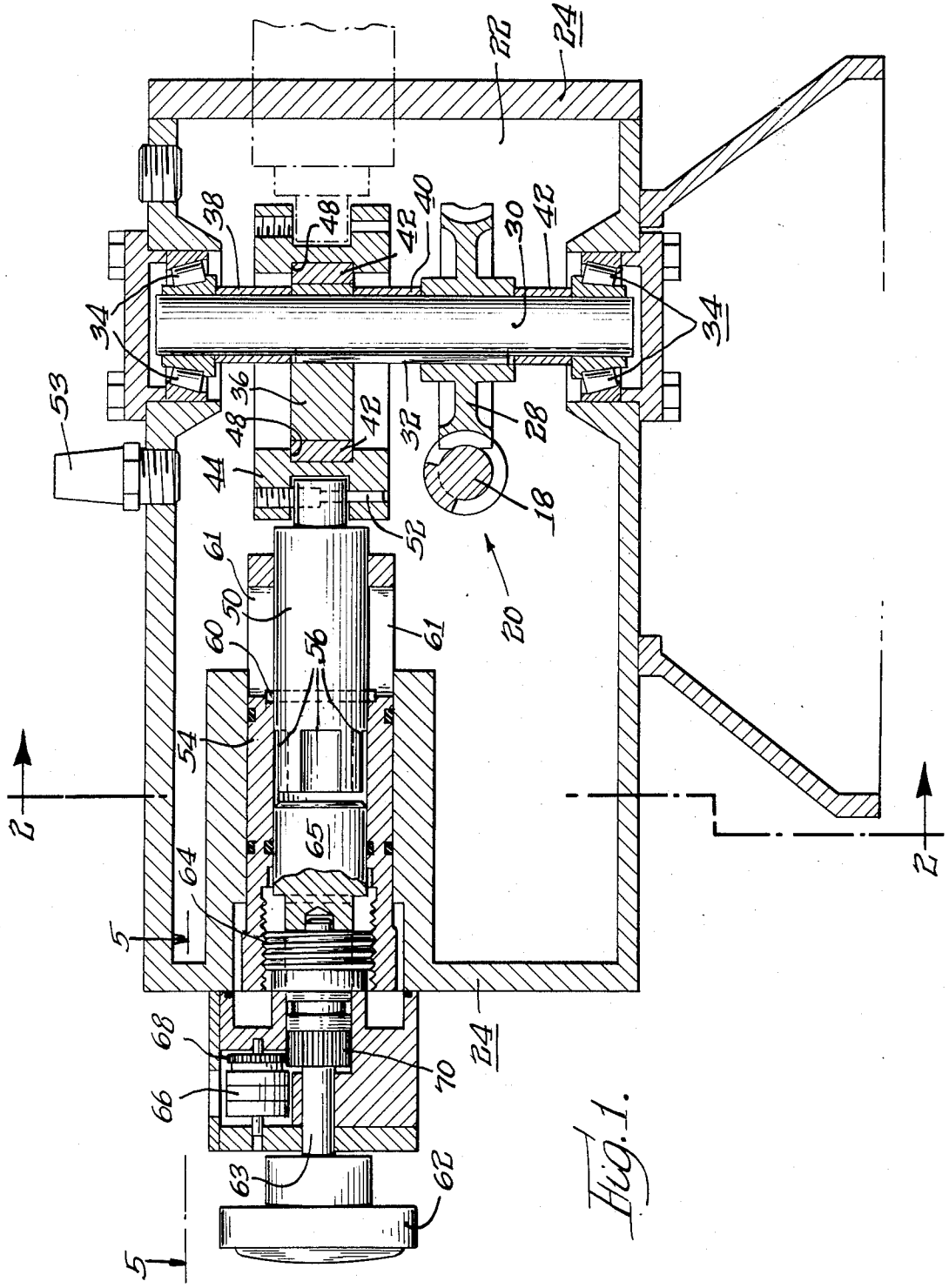

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Fig. 1.

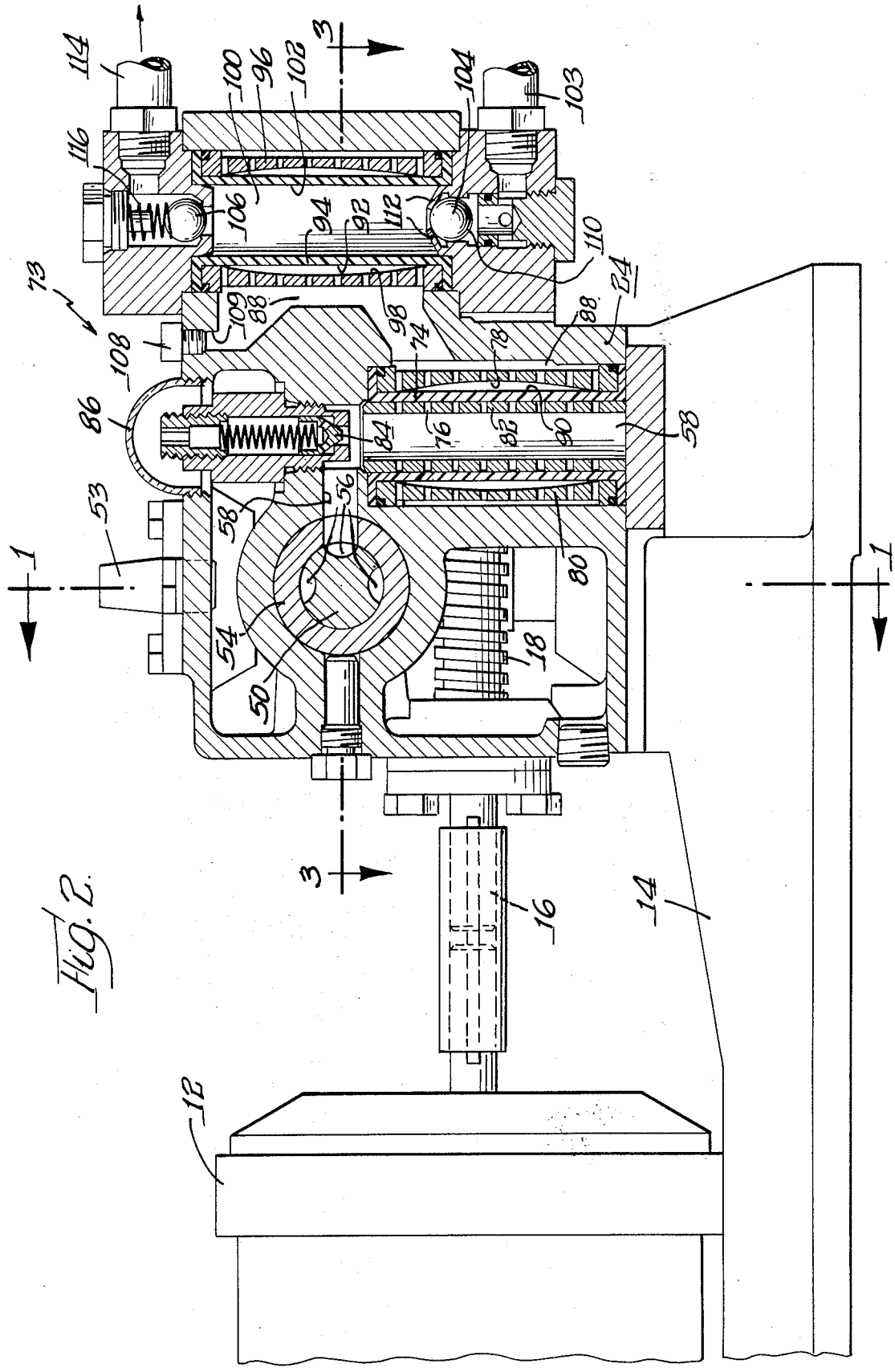

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Fig. 2.

Fig. 3.

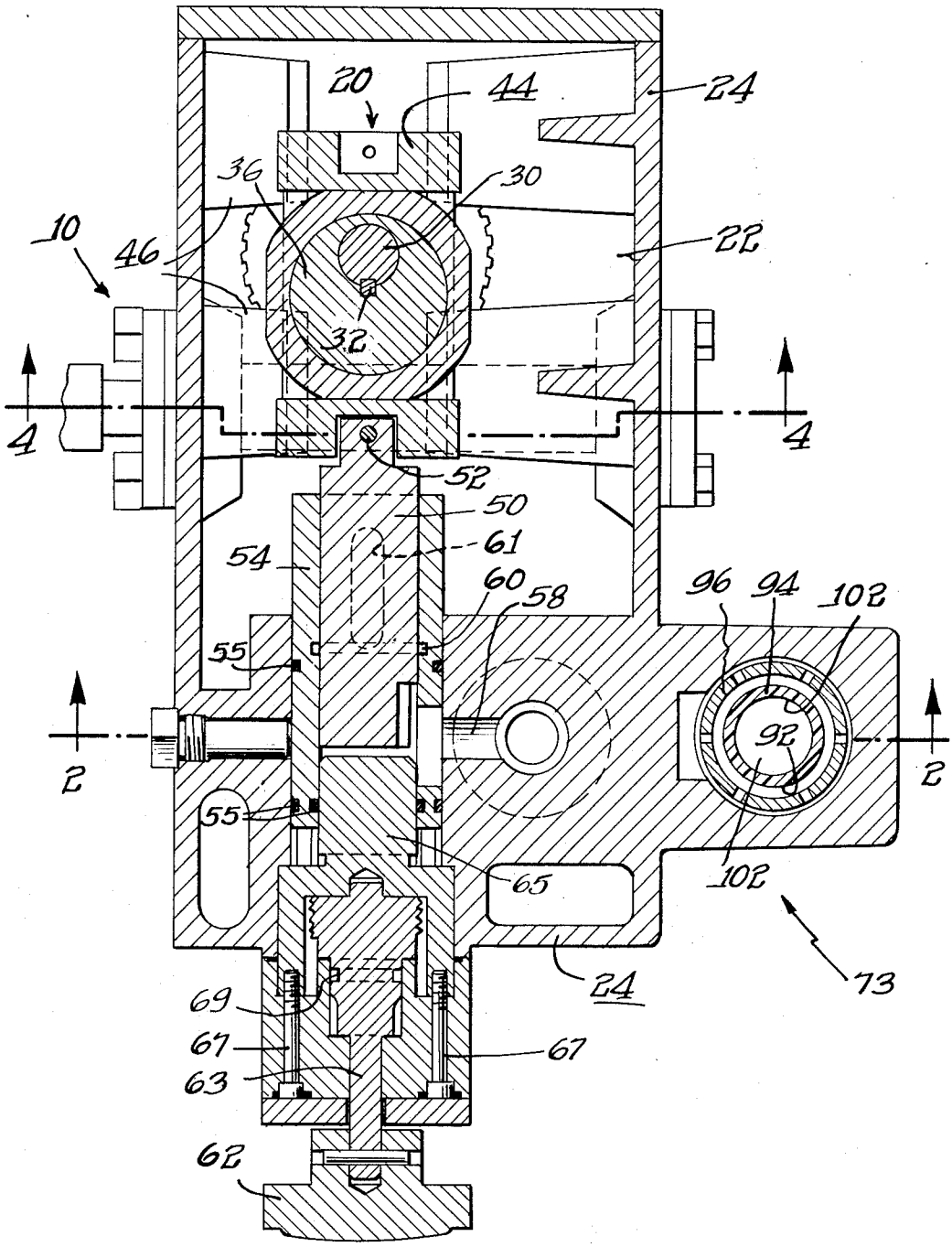

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Fig. 4.

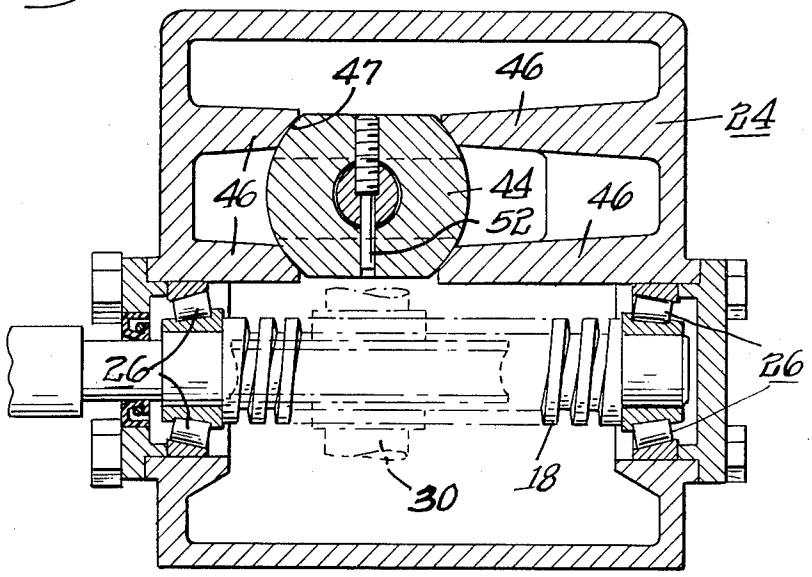


Fig. 5.

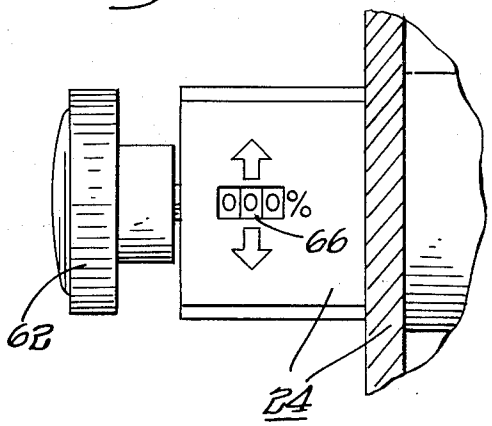


Fig. 6.

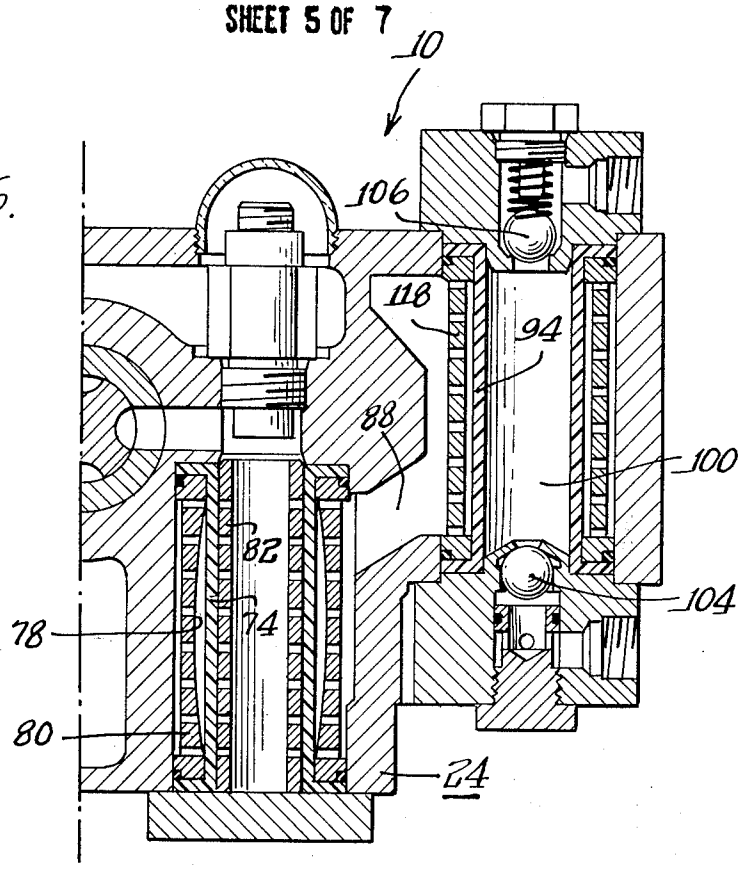

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Fig. 7.

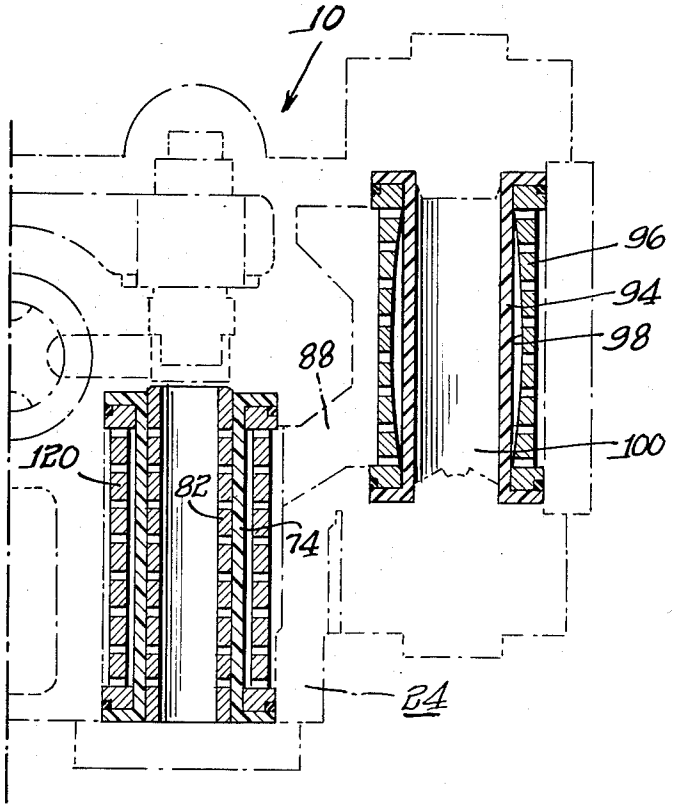

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Fig. 8.

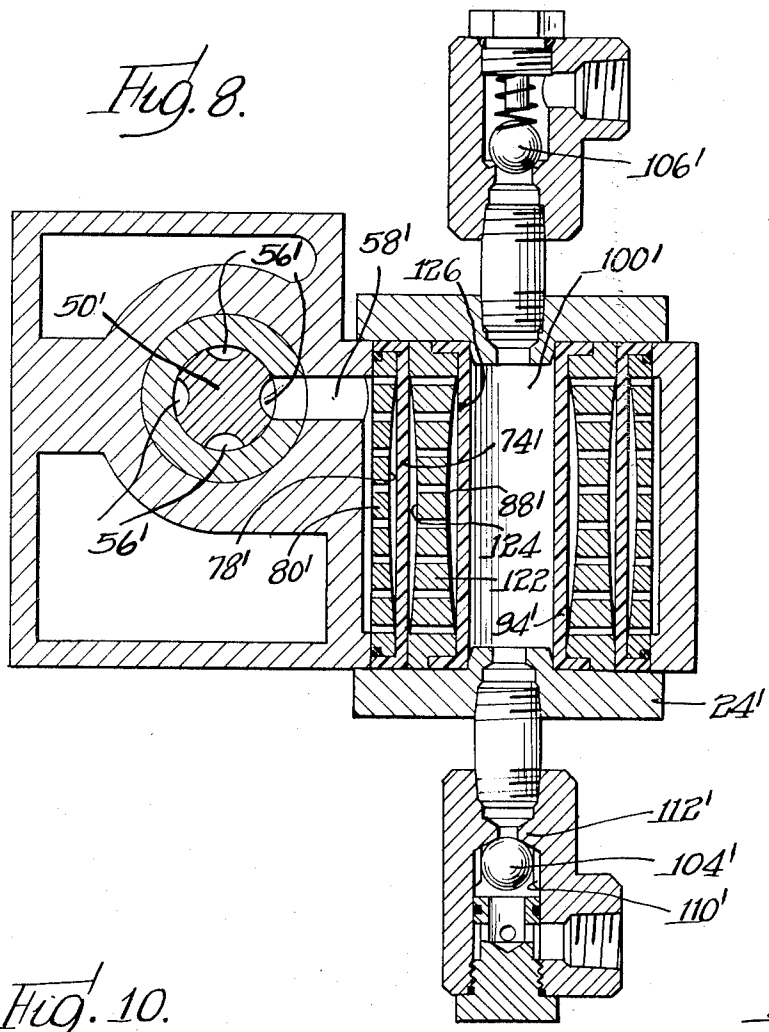


Fig. 10.

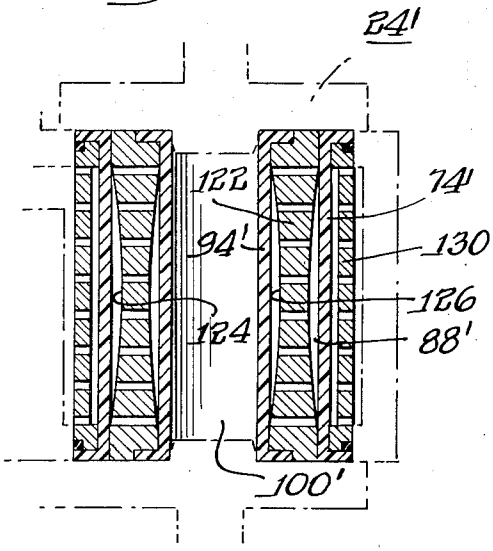


Fig. 9.

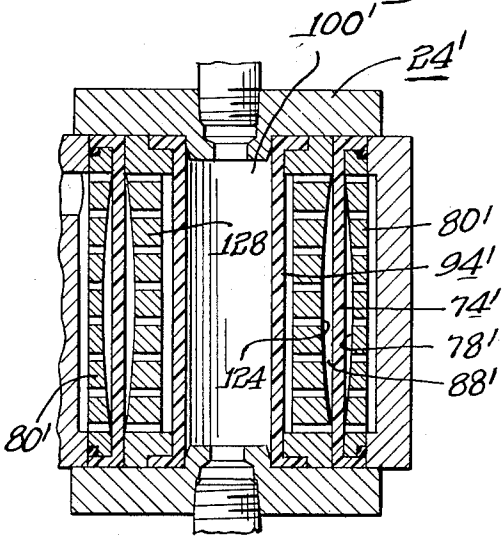


Fig. 11.

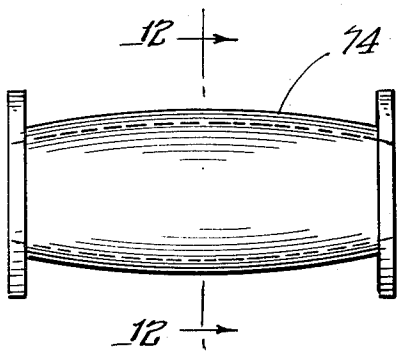


Fig. 12.

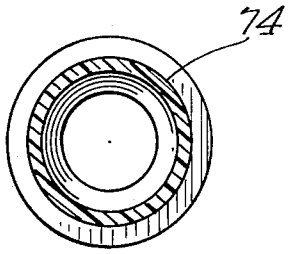


Fig. 13.

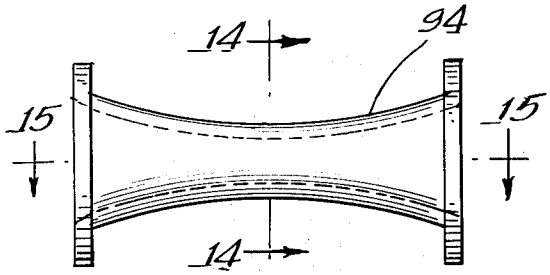
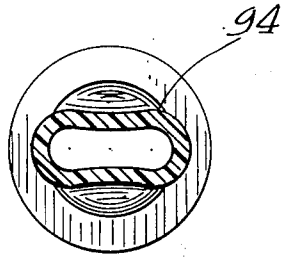


Fig. 14.



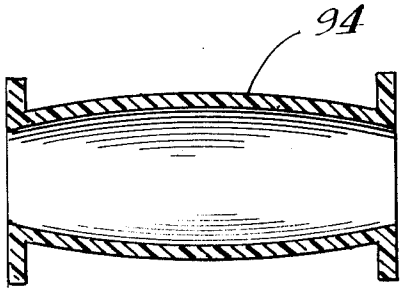


Fig. 15.

METERING PUMP

BACKGROUND OF THE INVENTION

Various types of metering pumps are presently in use. Some utilize a single diaphragm in combination with a piston and cylinder, while others utilize a double diaphragm. In both types, a displacement and a pumping chamber is partially defined by a diaphragm within the pump. In the double diagraphm type, however, an intermediate chamber is also defined between the process diaphragm and the hydraulic diaphragm, the intermediate chamber giving additional protection in the event of diaphragm failure. The intermediate chamber contains a liquid which is compatible with the process fluid being pumped. Thus, the process fluid is not contaminated in the event of diaphragm failure. All diaphragm type pumps, however, have the same disadvantage. Because the process fluids being pumped are often corrosive, the diaphragms per se are often made of noble metals for their corrosion resistance. The diaphragm heads on these pumps require a pressure vessel flange which is exposed to the process fluid. Consequently, this part of the pump housing must be manufactured from exotic metals which are also resistant to corrosion. As a result, the diaphragm heads are large, heavy, and expensive.

In some instances, the diaphragm head may be manufactured from plastics which are resistant to corrosion rather than exotic metal. In contrast to the metal diaphragm heads, however, plastic diaphragm heads are limited to low pressure applications.

Other metering pumps utilize a single elastomeric tube to partially define a pumping chamber and a displacement chamber, and ultimately effect the pumping of process fluid through the pumping chamber. In these pumps, if the tube ruptures during the operation of the pump, the displacement chamber will be damaged if the process fluid being pumped is corrosive. In any event, the process fluid will be contaminated by oil.

Another type of pump utilizes a single elastomeric tube in combination with a diaphragm, there being an intermediate chamber between the diaphragm and the outer surface of the tube. Thus, by using a fluid in the intermediate chamber which is compatible with the process fluid being pumped, contamination of the process fluid can be prevented in the event of tube failure. With this construction and that described in the previous paragraph, however, there is nothing to restrain or limit the movement of the tube beyond its elastic limit in the event there is an abnormal occurrence. Thus, in either construction, the tube is still likely to rupture given an abnormal occurrence. Furthermore, this type of pump construction is subject to the disadvantages inherent with the use of diaphragms.

Another problem encountered with present metering pumps relates to the drive mechanism for converting the rotary motion of an electric motor to reciprocating motion. In many pumps, as it became necessary to handle higher horsepower transmissions, more complex drive mechanisms were developed. The complex drive mechanisms developed are consequently more likely to malfunction; also, larger pump housings are required to accommodate the more complex drive mechanisms.

Examples of pumps which incorporate flexible diaphragms and tubes are shown in U.S. Pat. Nos. 1,282,145; 2,345,693; 2,812,716; 3,250,226; 3,318,251; 3,489,096; 3,527,550; and 3,551,076.

SUMMARY OF THE INVENTION

The present invention relates to an improved metering pump and pumping head, The pump includes an improved drive mechanism for reciprocating a piston within a cylinder. The pumping head employs, two elastomeric tubes, a hydraulic tube and a process tube, each have their ends secured within the pump housing so as to permit distortion of the central portion of each tube by a driving fluid. In the preferred embodiment of the invention, a displacement chamber is defined between the piston and the inner surface of the hydraulic tube for confining a volume of driving fluid during the reciprocation of the piston. During the reciprocation of the piston, passageways between the piston and cylinder periodically communicate with a reservoir containing a supply of the driving fluid to allow driving fluid to enter and exit the displacement chamber.

During the discharge stroke of the piston, driving fluid confined within the displacement chamber is displaced and causes the hydraulic tube to expand. As the hydraulic tube expands, it causes displacement of an intermediate driving fluid which is confined within an intermediate chamber, the latter being partially defined by the outer surface of each of the tubes. Consequently, displacement of the intermediate fluid compresses the process tube, causing process fluid contained within a pumping chamber within the tube to be discharged around an outlet valve.

During the suction stroke of the piston, each tube returns to its original position, causing process fluid to enter the pumping chamber around an inlet valve. On the next discharge stroke of the piston, the process will be repeated, and the process fluid which has entered the pumping chamber, will be discharged around the outlet valve once again.

The cylinder includes an adjustably mounted sleeve, the position of which can be changed with respect to the piston so as to vary the degree to which the passageways between the piston and the sleeve communicate with the reservoir during the reciprocation of the piston.

The drive mechanism for reciprocating the piston includes a first member having a circular cross-section which is eccentrically mounted so as to rotate about an axis normal to the cross-section. The cross-section of the first member is surrounded by a sliding block, the block in turn being enclosed by a crosshead which is confined so as to reciprocate within the pump housing. As the first member rotates, it causes the block to oscillate, which in turn causes the crosshead to reciprocate within the pump housing. The crosshead is connected to the piston, thus causing the latter to reciprocate.

Perforated sleeves define limit surfaces which surround both tubes, and in addition a perforated sleeve is mounted within the hydraulic tube to stop the tube in its original position during the suction stroke of the piston.

A second embodiment of the pumping head of the present invention is disclosed in which the elastomeric tubes are concentrically arranged; perforated sleeves define limit surfaces on each side of the hydraulic tube, and a limit surface surrounds the process tube. Various modifications are disclosed for both embodiments which include different sleeve arrangements.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of the preferred embodiment of the invention taken through line 1—1 of FIG. 2, and showing the piston and cylinder arrangement, and the drive mechanism for reciprocating the piston.

FIG. 2 is a sectional view taken through line 2—2 of FIG. 1, and showing the pumping head and the two elastomeric tubes in combination with various perforated sleeves which form limit surfaces for the tubes.

FIG. 3 is a sectional view taken through line 3—3 of FIG. 2, and showing another view of the piston and cylinder arrangement, and the drive mechanism for reciprocating the piston.

FIG. 4 is a sectional view taken through line 4—4 of FIG. 3 and showing how the crosshead of the drive mechanism is confined within the pump.

FIG. 5 is a view taken through line 5—5 of FIG. 1, and showing the counter readout.

FIG. 6 is a partial sectional view through a modified form of the pumping head of the present invention, similar to the right hand portion of FIG. 2, and specifically illustrating the two elastomeric tubes in a different combination with perforated sleeves.

FIG. 7 is a partial sectional view of still another embodiment of pumping head, similar to FIG. 6.

FIG. 8 is a partial sectional view of a further embodiment of the pumping head of the present invention in which the two elastomeric tubes are concentrically arranged in combination with various perforated sleeves which form limit surfaces for the tubes.

FIG. 9 is a partial sectional view of the embodiment shown in FIG. 8, but showing the two elastomeric tubes in a different combination with perforated sleeves.

FIG. 10 is a partial sectional view of the embodiment shown in FIG. 8, but showing the two elastomeric tubes in a different combination with perforated sleeves.

FIG. 11 shows a side view of an expanded elastomeric tube.

FIG. 12 is a sectional view taken through line 12—12 of FIG. 11.

FIG. 13 is a side view of a compressed elastomeric tube.

FIG. 14 is a sectional view taken through line 14—14 of FIG. 13.

FIG. 15 is a sectional view taken through line 14—15 of FIG. 13.

DETAILED DESCRIPTION OF THE INVENTION

Referring to FIGS. 1 through 4, it can be seen that the metering pump 10 is driven by a conventional electric motor 12, the latter being mounted to the pump frame 14. The motor 12 is coupled by suitable means 16 to worm 18, the latter being an element of the overall drive mechanism 20 which is mounted within the reservoir 22 defined within the pump housing of 24. The drive mechanism 20 is utilized to convert the rotary motion of the motor 12 into reciprocating motion at the proper pumping speed. As can be seen, the worm 18 is mounted in the housing 24 on suitable tapered roller bearings 26.

As worm 18 rotates, it drives a worm gear 28 with which it meshes. Because the latter is mounted on shaft 30 by means of a key 32, the shaft is caused to rotate on tapered roller bearings 34. As the shaft 30 rotates, it drives a rotary member 36 which is eccentrically mounted on said shaft 30 by means of the key 32. Both the gear 28 and member 36 are locked in predetermined positions along the axis of shaft 30 by means of sleeve sections 38, 40, and 42, section 40 having a slot formed therein to permit the key 32 to extend therethrough.

Surrounding member 36 in rotatable relationship thereto, is sliding block 42, and enclosing the latter is a crosshead 44. Crosshead 44 is supported within housing 24 by bearing support means 46, that define a bore 47, the bore confining the movement of the crosshead to a reciprocating movement. Because member 36 is eccentrically mounted on shaft 30, and block 42 is confined by its engagement in slots 48 formed in the crosshead 44, the movement of the block 42 surrounding member 36 will be an oscillating movement. This oscillating movement of block 42 is converted into a reciprocating movement by the crosshead 44, the latter being confined to move within bore 47 as described above.

In order to transmit the reciprocating motion of crosshead 44 to piston 50, the latter is connected to the crosshead by a connecting pin 52, the pin being accessible from the exterior of pump 10 when the vent cap 53 is removed. The cap 53 is located in line with the path of the pin 52, and when it is desired to disconnect the crosshead 44 from the piston 50, the cap is removed and the coupling 16 is rotated until the pin is accessible by means of a suitable wrench inserted through the opening into which the cap was screwed.

As illustrated in phantom, in the right hand portion of FIG. 1, the crosshead 44 may be employed to dirve a pair of pistons, and correspondingly, two separate pumping heads.

The piston 50 reciprocates within a cylinder defined by housing 24, which cylinder includes a sleeve 54 which surrounds the piston in sliding relationship thereto. It is noted that piston 50 has a plurality of ports or passageways 56 in the form of end milled undercuts on the perimeter of the piston. During reciprocation of piston 50, passageways 56 periodically communicate with reservoir 22, resulting in driving fluid passing into and out of the displacement chamber 58 during alternating strokes of piston 50, the reservoir 22 containing a supply of driving fluid. To vary the degree to which the displacement chamber 58 is placed in communication with reservoir 22 during the reciprocation of piston 50, sleeve 54 is adjustably mounted so that it can be moved with respect to the piston in a direction parallel to the direction or reciprocation thereof. It is noted that the sleeve 54 contains a circumferential groove 60 which communicates with openings 61 formed on opposite sides of the sleeve.

To adjust the position of sleeve 54 with respect to piston 50, a handwheel 62 has a shaft 63 which is threadably connected to sleeve 54 as indicated at 64, FIG. 1, the shaft being rotably confined within pump housing 24. Rotating handwheel 62 consequently causes sleeve 54 to move toward or away from crosshead 44, depending upon the direction of rotation. A digital counter 66 (commercially available fro Veeder Root Corporation of Hartford, Conn.) has a gear 68 which meshes with gear 70 on the handwheel shaft 63, the counter reading from 0 to 100 percent of pump capacity, a universal readout that does not require conversion to other units. Thus, the longitudinal position of sleeve 54 with respect to piston 50 determines the effective hydraulic

displacement of the pump **10.** Shaft **63** rotates within cylinder end closure **65,** the latter being secured in place by bolts **67.** An O-ring seal **69** is mounted between the shaft **63** and pump housing **24.**

When sleeve **54** is in the position which is farthest from crosshead **44,** there is zero effective hydraulic displacement. In this position, the circumferential groove **60** communicates with passageways **56** throughout the reciprocation of piston **50,** thus driving fluid is alternately withdrawn from and forced into the sump or reservoir **22** without driving force being applied to the driving fluid in the displacement chamber **58.** As the sleeve **54** is moved closer to crosshead **44,** passageways **56** will eventually override circumferential groove **60** during the discharge stroke of the piston **50** (leftward as viewed in FIG. **1**), resulting in an effective hydraulic displacement for the remainder of the discharge stroke, and the application of driving force to the driving fluid in chamber **58.** The maximum hydraulic displacement is achieved when sleeve **54** is in the position which is closest to crosshead **44.** Even in this latter position, however, there is still a very slight communication between passageways **56** and circumferential groove **60** to assure expulsion of air from displacement chamber **58;** there is about a 0.125 inch overlap between the passageways **56** and groove **60.** The degree to which passageways **56** communicate with reservoir **22** is proportional to the position of sleeve **54** within the pump housing **24.** The generous porting provided by passageways **56** reduces the pressure required to expel oil from the displacement chamber during reciprocation, consequently permitting higher pumping speeds since the fluid flow velocities are easily kept within allowable limits.

It is noted that sleeve **54** is hydraulically balanced and requires little more force for adjustment than is necessary to overcome the frictional resistance of the O-ring seals **55** located on the inner and outer diameters of the sleeve. Each end of the sleeve **54** is subjected to atmospheric pressure which thus permits the sleeve to be easily adjusted regardless of whether the piston **50** is on a suction or discharge stroke.

Associated with the piston **50,** and the above discussed drive mechanism, is a pumping head, designated generally **73.** The pumping head **73,** in the illustrated embodiment, is formed as an integral unit of the entire pump assembly, sections thereof serving to define a portion of the displacement chamber **58.** Generally, as will be detailed more fully hereinafter, the pumping head **73** includes an elastomeric hydraulic tube **74,** a chamber **88** for intermediate fluid, and a flexible process tube **94** providing a pumping chamber **100.** It is to be understood that the pumping head **73** may be formed as a separable unit, with means being provided for operable connection of the displacement chamber of the pumping head, with that of a displacement piston and drive arrangement. Further, while the tubes **74** and **94,** as illustrated, are elastomeric, they may be constructed of any suitable flexible material.

The displacement chamber **58** which confines a volume of driving fluid during the discharge stroke of piston **50,** is partially defined by an elastomeric hudraulic tube **74,** in particular by the inner surface **76** of the tube. During the discharge stroke of piston **50,** a volume of driving fluid confined in chamber **58** between the piston **50** and tube **74** will be displaced, thus expanding the tube. The tube **74** will expand until it is ultimately restrained by an annular limit surface **78** which is defined by a perforated sleeve **80** which surrounds the tube. The contoured limit surface **78** is designed so as to support the expanded tube **74** within its elastic limit. As used herein, "annular limit surface" means that surface which is defined by a perforated sleeve, and which allows an elastomeric tube to expand or contract freely, but which will keep the tube from being distorted beyond its elastic limit. In other words, the elastomeric tube will not be forced to follow a shape other than the one which is freely formed as it is distorted by the driving fluid. The term "annular limit surface" also applies to one defined by a perforated sleeve which stops the tube in its nondistorted or original position.

The latent energy imparted to tube **74** during its expansion during the discharge stroke of piston **50,** is given up on the return or suction stroke of the piston. As the tube **74** elastically returns to predistorted position, it maintains a pressure in the displacement chamber **58** until passageways **56** once again communicate with the reservoir **22** which is connected to the atmosphere. The pressure maintained in the displacement chamber **58** keeps the driving fluid above its vapor pressure, and also keeps dissolved gases in solution that would otherwise break out of solution and subtract from the volumetric intake, and subsequently the volumetric efficiency of the pump.

A sleeve **82** is also mounted or arranged within the tube **74,** this sleeve defining an annular limit surface which stops the tube in its original or nondistorted position, and which prevents the tube from moving beyond its elastic limit. By mounting sleeve **82** within the hydraulic tube **74,** the pump **10** can continue to operate even if the process tube **94,** to be discussed in detail hereinafter, ruptures during operation of the pump. If sleeve **82** were not provided, the hydraulic tube **74** might be totally collapsed on the suction stroke of the piston **50** to such an extent that damage and eventual rupture might occur. Sleeve **82** also serves to assist in keeping the ends of tube **74** secured in place within the pump.

As driving fluid is displaced within displacement chamber **58** during the discharge stroke of piston **50,** the pressure of the driving fluid will rise. A hydraulic relief valve **84** is provided which protects the pump **10** from over-pressure conditions, possibly caused by an accidental blockage of the pump discharge, etc. Relief valve **84** will permit the driving fluid to be passed to the reservoir **22** in the event a predetermined pressure is reached during the discharge stroke of piston **50.** A clear plastic window or bubble **86** is provided above the relief valve **84** in order to permit visual inspection of the valve operation to determine when driving fluid is being passed to the reservoir.

As tube **74** is expanded on the discharge stroke of piston **50,** it causes displacement of an intermediate driving fluid which is contained within an intermediate chamber **88,** this latter chamber being partially defined by the outer surface **90** of tube **74,** and the outer surface **92** of an elastomeric process tube **94.** The displacement of the intermediate fluid compresses process tube **94,** causing a discharge displacement from the pump **10.** Surrounding tube **94,** is a perforated sleeve **96** which defines an annular limit surface **98.** Annular limit surface **98** also serves to maintain tube **94** within its elastic limit. As with limit surface **78,** limit surface

98 is coincident with the contoured shape which is freely assumed by the distorted tube 94 as a result of the pressure of the driving fluid and while the tube is within its elastic limit. Therefore, tube 94 cannot be forced to assume a shape other than one freely formed as it is compressed by the intermediate driving fluid. This results in an elastic distortion without the stress concentrations caused by stretching the tubes irregularly beyond its normal freely distorted shape.

It is noted that each of the tubes 74 and 94 have their ends secured within the pump 10 so as to permit distortion of the central portion of each tube. Each of the tubes alternates between a first configuration and a second configuration during the reciprocation of piston 50. The driving fluid confined within displacement chamber 58 causes the tube 74 to assume its alternate configurations, while the intermediate driving fluid within intermediate chamber 88 causes tube 94 to assume its alternating configurations during the operation of the pump. The alternating configurations assumed by tube 94 results in a process fluid being pumped through the pumping chamber 100, the latter being partially defined by the inner surface 102 of the tubes 94, and an inlet valve 104, and an outlet valve 106.

Tube 94, when operating with a suction lift, compresses elastically on the discharge stroke of piston 50, and cannot be compressed beyond the volume displacement extension limit of the hydraulic tube 74. Therefore, tube 94 cannot be over compressed to an extent such that the walls of the tube come into contact with each other.

Referring to FIGS. 11 through 15, the shapes assumed by tubes 74 and 94 during distortion (expansion or contraction) are shown. FIGS. 11 and 12 show the configuration assumed by a tube when it is expanded. FIGS. 13 through 15 show the configuration assumed by a tube when it is contracted. It is noted that the perimeter of an expanded tube is greater than when the tube is in its nondistorted position, while the perimeter of a compressed tube is the same or substantially the same as when the tube is in its nondistorted position. Thus, more energy is required to expand the tube 74 than is required to compress the tube 94. Both tube 74 and tube 94 aid in the suction lift capability of pump 10. The energy imparted to tube 74 to expand the same, and the energy imparted to tube 94 to compress the same, during the discharge stroke of piston 50, is given up on the return stroke of the piston. The additive aid given to the suction lift capability of the pump 10 is substantially greater for tube 74, however, than for tube 94, because as stated above, more energy is required to expand tube 74 than is required to compress tube 94.

The ability of the combined hydraulic tube 74 and process tube 94 to aid in the suction lift capability of the pump 10 is dependent upon the physical characteristics of the flexible material from which each tube is manufactured. Because tube 74 is the most highly stressed and consequently the greatest aid when pumping with a suction lift, its physical characteristics are most important. Fortunately, tube 94 is isolated from the process fluid being pumped through pumping chamber 100, and consequently does not come into contact with the process fluid.

In a suction lift application, the elastomeric material from which the tube 74 is constructed, is selected for its high modulus of elasticity. Therefore, as previously explained on the suction stroke of piston 50, hydraulic tube 74 maintains a hydraulic pressure in the displacement chamber 58 until the tube 74 returns to its predisplaced position. The suction lift capability of pump 10, however, is not dependent solely upon the elastic characteristics of the tubes 74 and 94. The suction lift capability of the pump is also aided by the suction stroke of the piston 50 which creates a pressure which is less than atmospheric in the displacement chamber 58, thus aiding in the total suction lift potential of the pump 10.

Because process tube 94 is almost always coming into contact with process fluids that are corrosive or abrasive, the selection of the elastomeric material for the process tube is based on its resistance to these fluids. The best materials for corrosion resistance do not necessarily have the physical properties required for the hydraulic tube 74. A pump utilizing the present double tube arrangement, has an excellent suction lift capability which is aided by the characteristics of the hydraulic tube 74; in addition such a pump is resistant to abrasive or corrosive process fluid due to the characteristics of the process tube 94.

It is noted that all of the perforated sleeves disclosed in the application, have openings or holes which are symmetrically located around the sleeve, and which openings are large enough to allow the driving fluid to distort the elastomeric tubes as desired, but small enough to prevent damage to the tube due to elastic deflection of the tubes into the openings.

As a result of the alternating configurations assumed by tube 94 during the reciprocation of piston 50, process fluid enters the pumping chamber 100 via inlet line 103 and inlet valve 104, and exits the chamber via outlet valve 106. As stated above, the alternating configurations assumed by tube 94 are effected by the displacement of the intermediate driving fluid which is contained within the intermediate chamber 88. By removing plug 108, the intermediate fluid is introduced through the opening 109 into the latter chamber prior to start-up of the pump.

Inlet valve 104 is guided by four lands 110, and is stopped in an upper position by a plurality of ball stops 112, the lands and stops being sized so as to provide an adequate flow passage around the inlet valve in this upper position. When valve 104 is in this upper position, the piston 50 is on its suction stroke; during the suction stroke the energy imparted to the tube during its compression is given up, thus causing a decrease of pressure in pumping chamber 100 which is satisfied by process fluid entering the latter chamber via line 103.

As piston 50 begins its return or discharge stroke, tube 94 is compressed, thus causing the process fluid to be discharged around the outlet valve 106 and through the outlet line 114. It is noted that process fluid enters the latter line above the outlet valve 106 to assure that the flow around the outlet valve is evenly distributed so that the latter bears evenly on discharge valve spring 116. Discharge spring 116 requires a minimum discharge pressure to lift outlet valve 106 off its seat which is equal to the minimum pressure required to displace driving fluid out of the displacement chamber 58 via passageways 56 during the discharge stroke of piston 50, and back to the reservoir 22.

FIGS. 6 and 7 show an alternate embodiment of the pumping head of FIGS. 1 through 5, but in combination with different sleeve arrangements. In FIG. 6, the hydraulic tube 74 is combined with the same sleeve arrangement as shown in FIGS. 1 through 5. The process tube 94, however, is now surrounded by a perforated sleeve 118 which does not employ an annular limit surface as defined above. Sleeve 118 serves to assist in securing the ends of the process tube 94 in place within the pump.

In FIG. 7, the process tube is combined with the same sleeve arrangement as shown in FIGS. 1 through 5, but the hydraulic tube 74 is combined with a different sleeve arrangement. Perforated sleeve 82 is still mounted within tube 74, but surrounding the tube is a perforated sleeve 120, which similar to sleeve 118 in FIG. 6 does not employ a contoured annular limit surface as defined herein; sleeve 120 also serves to assist in securing the ends of tubes 74 in place within the pump.

FIGS. 8, 9, and 10 show a second embodiment of the invention which differs from the embodiment shown in FIGS. 1 through 7 in that the two elastomeric tubes are concentrically arranged, i.e., the hydraulic tube 74' surrounds the process tube 94'. FIGS. 8, 9 and 10 differ with respect to each other only as to the sleeve arrangements. In each of the latter figures, it can be seen that driving fluid is now confined between the piston 50' and the outer surface of tube 74', rather than between the piston and the inner surface of the tube. Consequently, on the discharge stroke of piston 50', tube 74' will now be compressed rather than expanded as in the embodiment shown in FIGS. 1 through 7.

Referring to FIG. 8, it can be seen that during the discharge stroke of piston 50', driving fluid confined within the displacement chamber 58' will be displaced so as to compress the hydraulic tube 74' to the configuration shown in FIGS. 13 through 15. To control expansion and compression of tube 74', perforated sleeves 80' and 122 are provided, these sleeves defining contoured annular limit surfaces 78' and 124, respectively, which serve to control expansion and contraction of tube 74', thereby preventing damage to said tube and, to some extent, regulating or defining the limits of the pumping action.

The intermediate driving fluid contained within intermediate chamber 88' is displaced during the alternate compression and expansion of tube 74', thus producing corresponding contraction and expansion of the process tube 94'. Tube 94' cannot be over expanded because of the annular limit surface 126 which is contoured to define or limit maximum expansion thereof. As tube 94' is compressed, process fluid is displaced from pumping chamber 100' around outlet valve 106'. On the suction stroke of piston 50', tubes 74' and 94' are expanded, consequently causing process fluid to enter the pumping chamber 100' around inlet valve 104'. When the latter valve is in the position shown in FIG. 8, it is guided by four lands 110' and stopped by ball stops 112', these lands and stops being sized so as to provide adequate flow passages around the valve.

In FIG. 9, the arangement is the same as shown in FIG. 8, except that the perforated sleeve 128 does not define a contoured annular limit surface surrounding the tube 94'.

In FIG. 10, the arrangement is the same as shown in FIG. 8, except that the perforated sleeve 130 does not define a contoured annular limit surface surrounding the tube 74'.

It is noted that in the embodiment shown in FIGS. 8, 9 and 10, the hydraulic tube 74' does not aid substantially in the suction lift capability of the pump, because the tube is now compressed rather than expanded as in the embodiments shown in FIGS. 1 through 7. Consequently, because much less energy is required to compress the tube than expand it, much less energy is also released by the tube during the suction stroke of piston 50'.

We claim:

1. An improved pump which comprises: a housing defining a cylinder therein; a piston slidably mounted within the cylinder; drive means for effecting relative reciprocation of said piston in said clyinder; a pumping head in operative communication with said cylinder, said pumping head comprising: a housing; a first flexible tube having an inner surface and an outer surface, the ends of said first tube being mounted within the housing so as to permit distortion of the central portion of the first tube; internal limit sleeve means disposed within said first flexible tube, and external sleeve means disposed about the exterior of said first tube; means for confining a volume of driving fluid between the piston and the inner surface of the first tube during the reciprocation of said piston to cause said first tube to alternate between a first configuration and a second, expanded configuration; a flexible second tube having an inner surface and an outer surface, the ends of the second tube being mounted within the housing so as to permit distortion of the central portion thereof; an external limit sleeve disposed about the periphery of said second flexible tube; means including the exterior surface of the first tube, the inner wall surfaces of the housing and the outer surface of the second tube defining an intermediate chamber, said intermediate chamber being adapted to contain a given volume of intermediate fluid, the second tube being adapted to alternate between the first configuration and a second configuration during expansion and contraction of said first flexible tube; means including valve means and the internal surface of said second tube, defining a pumping chamber through which a process fluid is pumped as a result of the alternating configuration assumed by said second tube; said first tube being constructed of an elastomeric material having a high modulous of elasticity, said external limit sleeve about the exterior of said first tube confining the expansion thereof on the piston displacement stroke, such that said expansion does not exceed the elastic limits of said first tube, said first tube thus absorbing and storing latent energy during expansion, and giving up said energy upon contraction during the suction stroke of the piston to maintain the driving fluid at a pressure level sufficient to prevent any dissolved gases in said fluid from coming out of solution.

2. A pump as defined in claim 1, wherein said housing includes a relatively movable sleeve which defines said clyinder, said sleeve having port means formed therein and communicating with a reservoir for operating fluid, means mounting said sleeve for axial movement relative to said piston and said housing, such that the point in the piston stroke wherein said piston blocks said port means and exerts compressive, displacement force on the operating fluid may be varied, thereby varying the displacement of said pump.

3. A pump as defined in claim 2 wherein said mounting means comprises a threaded coaxial extension on said sleeve, a threaded member mounted coaxially of said sleeve and in threaded driving engagement with said threaded extension operating means extending exteriorly of said housing for effecting rotation of said threaded member, and means journalling said threaded member to said housing in a fixed, axial position, whereby rotation thereof will produce axial movement of said sleeve relative to said piston.

4. An improved pump according to claim 1, which includes means defining a reservoir within the housing for containing a supply of the driving fluid, and means for periodically placing the inner surface of the first tube in communication with the supply of driving fluid during the reciprocation of the piston, and wherein the drive means is disposed within the reservoir.

5. An improved pump according to claim 1, comprises which a relief valve for passing a portion of the volume of driving fluid to a reservoir if a predetermined pressure is reached during the reciprocation of the piston, and wherein the housing further includes means permitting visual observation of the driving fluid passing to the reservoir.

6. An improved pump according to claim 1, wherein the drive means comprises:

a. a first member having a circular cross-section, the first member being eccentrically mounted within the housing to rotate about an axis of rotation which is normal to the cross-section;

b. means at least partially surrounding the cross-section in sliding relationship to the first member; and

c. means for causing the surrounding means to reciprocate within the housing during the rotation of the first member, and means connecting the piston to the surrounding means so as to cause the piston to reciprocate during the rotation of the first member.

7. An improved pump according to claim 6, wherein the surrounding means comprises:

a. a second member surrounding the cross-section in sliding relationship to the first member; and

b. a crosshead, the crosshead being connected to the piston and having means for holding the second member in a position along the axis of rotation so as to be driven by the first member.

8. An improved pump according to claim 7, wherein the crosshead also includes means for operatively connecting a second piston thereto, and the housing includes a cover removably mounted adjacent the latter means, the construction and arrangement being such that when the cover is removed, the second piston can be connected to the crosshead.

9. An improved pump according to claim 1, wherein the drive means includes means for connecting a second piston thereto, and the housing includes a cover removably mounted adjacent the latter means, the construction and arrangement being such that when the cover is removed, the second piston can be connected to the drive means.

10. An improved pump according to claim 1, wherein the housing further defines a reservoir for containing a supply of the driving fluid, and the drive means further comprises:

a. a shaft mounted within the housing for rotation about the axis of rotation, the first member being connected to the shaft so as to rotate therewith;

b. a worm gear connected to the shaft so as to rotate therewith; and

c. a worm disposed in driving relationship to the worm gear, the drive means being at least partially disposed within the reservoir. 9

11. An improved pump according to claim 10, wherein the housing includes a removably mounted cap which is disposed in line with the path of the crosshead to provide for the insertion of a tool to disconnect the piston from the crosshead.

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