

June 6, 1967

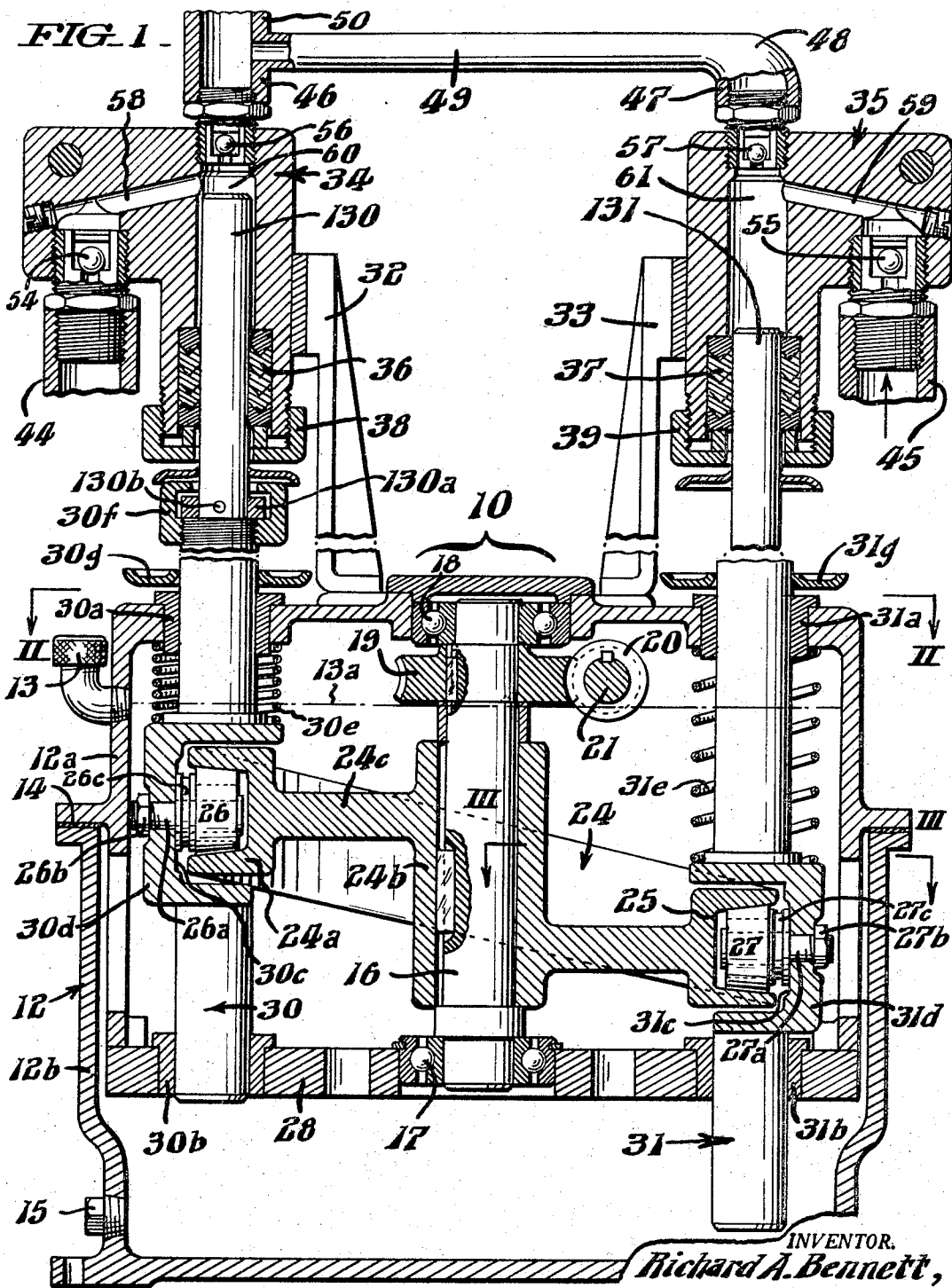
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3,323,461

METERING PUMP

Filed Jan. 21, 1965

6 Sheets-Sheet 1



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6 Sheets-Sheet 1

FIG. 2

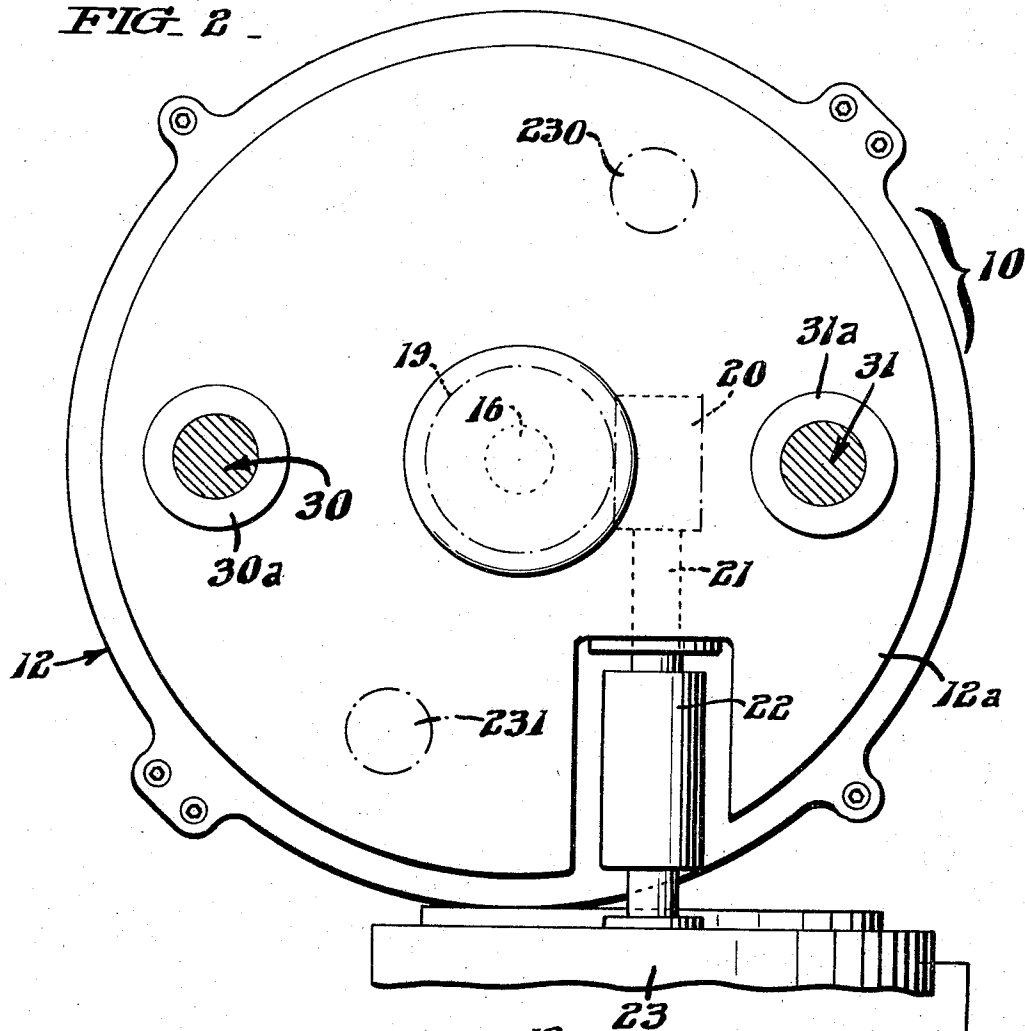
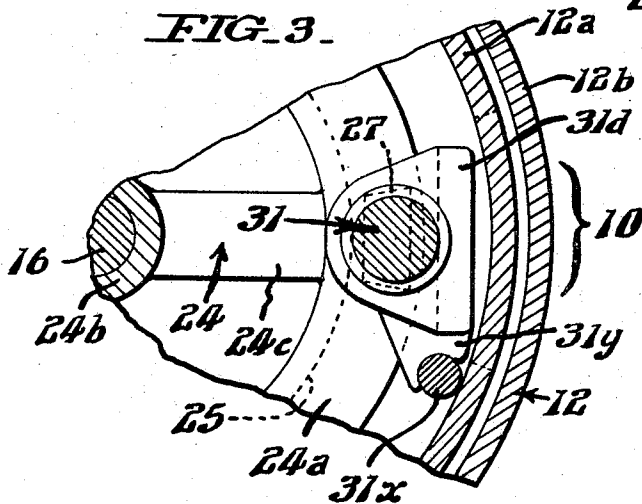


FIG. 3



29 SPEED CONTROL

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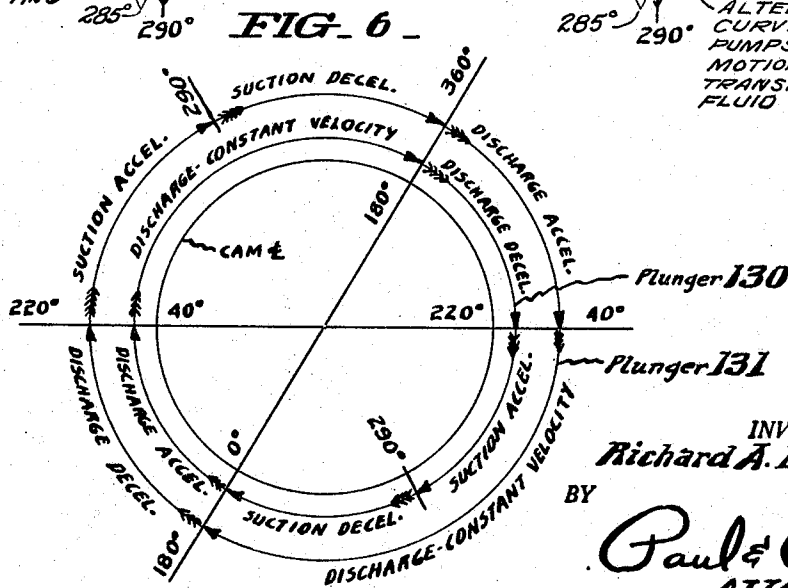
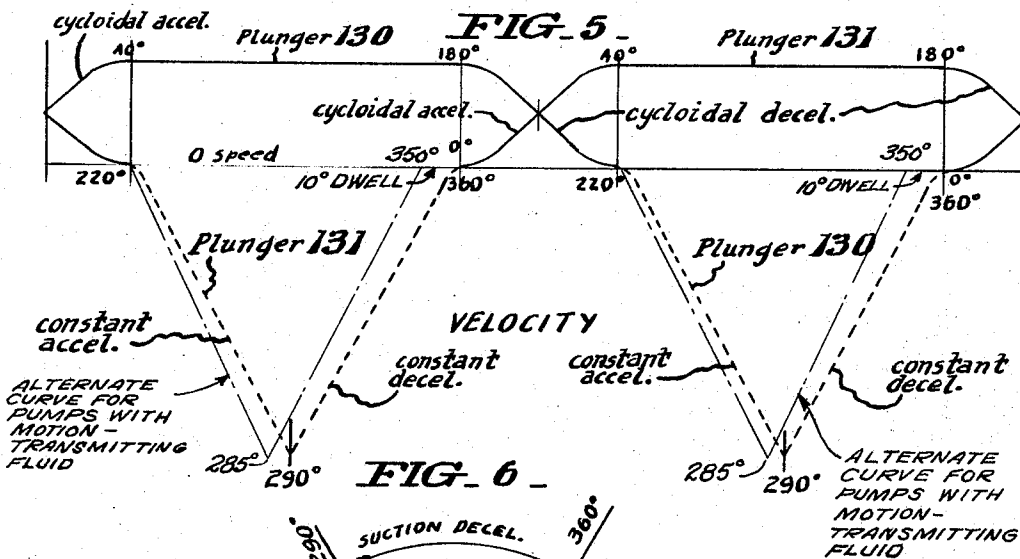
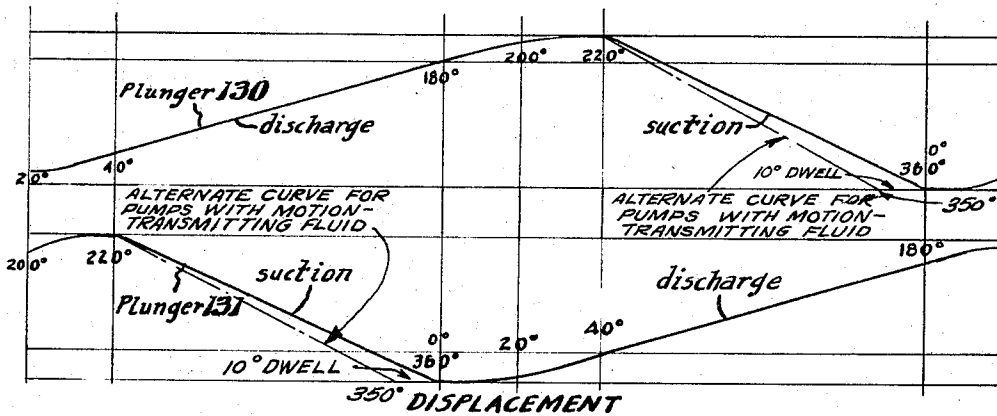
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6 Sheets-Sheet 3

FIG. 4 -



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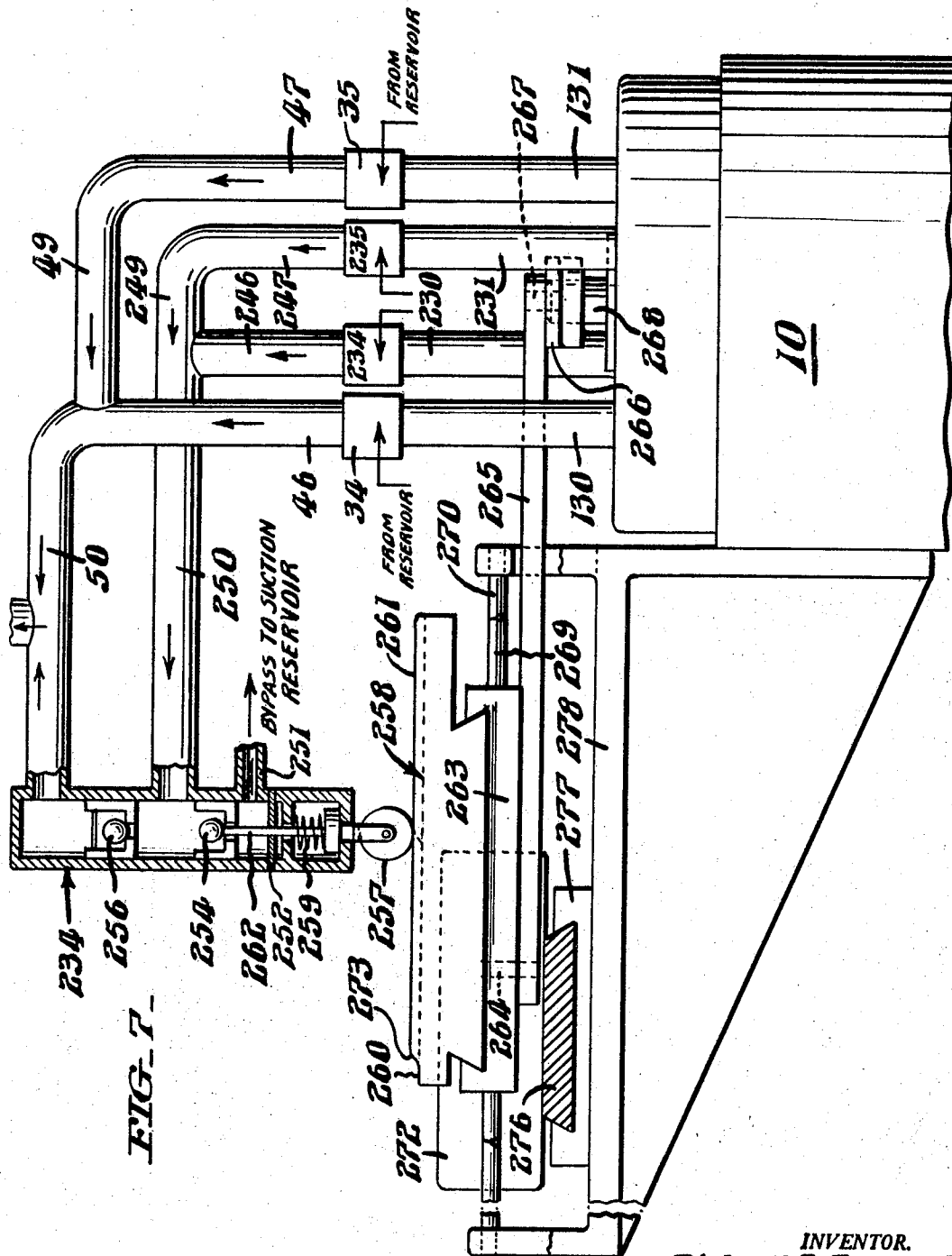
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6 Sheets-Sheet 4



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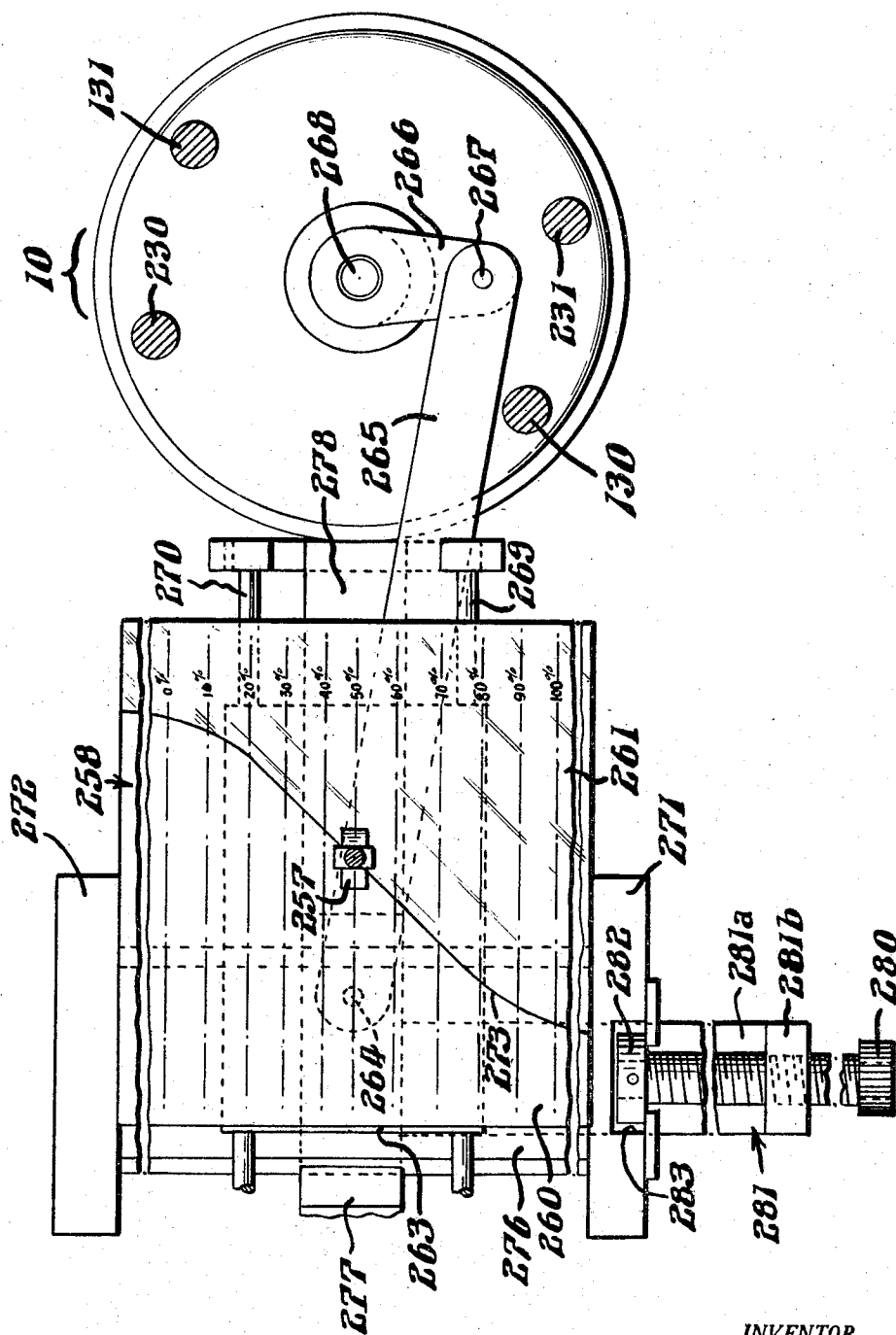
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METERING PUMP

Filed Jan. 21, 1965

6 Sheets-Sheet 5

FIG. 8 -



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METERING PUMP

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15 Claims. (Cl. 103—11)

This application is a continuation-in-part of an earlier-filed pending application, Ser. No. 404,259, filed Oct. 16, 1964, entitled, "Metering Pump," now abandoned.

This invention relates to an improvement in metering pumps.

In its most simple form, the metering pump provided by the present invention delivers a steady continuous unfluctuating flow of liquid which is derived from the discharge of a pair of vertically-disposed plungers whose crossheads are driven by a single cam. The design of the cam is such that the horsepower requirement of the pump drive motor is much smaller than (approximately 41% of) the peak motor horsepower required to drive a conventional prior-art slider-crank pump delivering the same capacity and against the same discharge pressure. This low horsepower requirement enables the use of a relatively inexpensive SCR (silicon controlled rectifier) controller in combination with a small DC shunt wound motor to give variable speed control of the motor and thus variable volume control of the fluid pump. The drive motor may, for example, be a 1/8 horsepower DC shunt wound motor controlled by a Minarik Electric Company type SH-54 motor speed control with silicon controlled rectifier circuit. This controller contains a torque "feedback" feature which, coupled with the relatively constant horsepower requirement of the constant discharge velocity pump of the present invention, results in excellent motor speed regulation and accurate fluid discharge. The Minarik speed control employs a dial control which linearly varies the motor speed over a 20 to 1 range. This equipment also includes a dynamic brake feature which allows the motor to be left in the brake position for any length of time without damage to the control or to the motor. Thus, fluid may be delivered on a "one-shot" basis and then the pump may remain at rest until it receives a signal to repeat.

In accordance with another aspect of the present invention, a bypass form of control means is provided for varying the volume output of the above pump in a controlled manner without changing the speed of the drive motor. Thus, the pump may be driven by a constant speed motor even though controlled variations in output are desired.

Prior art pumps adapted to give a variable controlled output flow ordinarily employ lost-motion devices and techniques. For example, in a typical prior art pump of this type, an adjustable cam stops the plunger for an adjustable period of time on the return or suction stroke. This type of pump is subject to at least one important disadvantage. The liquid tends to follow the head of the plunger as it returns on the suction stroke, and, when the plunger is brought to an abrupt and complete stop, by the cam just referred to above, the liquid is reflected and rebounds in the discharge direction. In this way, unwanted pressure is applied in the pump discharge direction, and, in a low pressure system, this unwanted back pressure will be sufficient to force liquid in the output direction. This unwanted action is called "overpumping." The flow control means provided by the present invention for varying the flow does not introduce "overpumping"; and freedom from "overpumping" is one of the advantages of the new pump and flow control means therefor, since "overpumping" obviously adversely affects metering accuracy.

2

In another form, the pump includes two (or more) pairs of vertically-disposed plungers, together with control means for combining the discharge of the two pairs of plungers to provide a flow which is steady, continuous and unfluctuating except when the flow is being purposely changed. In the latter form, the pump may be also driven by a constant speed motor, and the volume of flow from the pump may be varied by control means which control automatically the mixing of the outputs of the two pairs of plungers. It will also be seen that the pump of the present invention in the form just referred to having two pairs of vertically-disposed plungers will deliver two continuous streams, thus assuring perfect formulating or mixing. (The slider-crank pump of the prior art does not have this feature since it does not deliver two continuous streams. To the contrary, in the slider-crank pump, the flow from each plunger is harmonic, not constant velocity.)

In yet another form, the pump is adapted for the pumping of corrosive, or radioactive, or like materials without the possibility of leakage to the atmosphere. In this form, a diaphragm (preferably a tubular diaphragm) contains the corrosive liquid. The diaphragm is compressed by motion-transmitting liquid during the discharge stroke and is expanded during the suction stroke preferably by the assistance of leaf springs. The replenishing valve in the motion-transmitting liquid system is not required to act as a relief valve (as is ordinarily the case) since the pump is on constant stroke and the leaf springs keep the chamber at equilibrium.

An important object of the present invention is to provide a novel metering pump adapted to give a steady, continuous, unfluctuating flow which, however, may be controllably varied without changing the speed of the drive motor and while the pump is running.

Another object is to provide a metering pump which is controllably varied by an inexpensive silicon controlled rectifier controller and a DC shunt wound motor.

Another object is to provide a metering pump adapted to discharge two (or more) continuous streams generated by a single cam, which arrangement results in an inexpensive formulating (mixing) device.

Another object is to provide a metering pump which requires little or no compensation or correction.

Another object is to provide a metering pump adapted to discharge two continuous streams and flow control means therefor whereby the second stream may be metered independently of the first.

Another object is to provide a flow adjustment for a constant speed motor driven pump.

Another object is to provide a pump adapted for the pumping of corrosive, or radioactive, or like, materials without the possibility of leakage to the atmosphere.

These and other objects and advantages of the present invention will be clear from the following description of one or more preferred embodiments selected for illustration in the drawing in which:

FIG. 1 is an elevational sectional view of a pump in accordance with the present invention;

FIG. 2 is a top plan view of the pump of FIG. 1 looking down along the line II—II of FIG. 1;

FIG. 3 is a fragmentary view in section looking down along the line III—III of FIG. 1;

FIG. 4 is a graphical representation illustrating the preferred displacement relationships of the plungers;

FIG. 5 is a graphical representation illustrating the preferred velocity relationships of the pump plungers;

FIG. 6 is a different type of diagram also illustrating the preferred velocity relationships of the pump plunger during the cycle;

FIG. 7 is a side elevational view illustrating one preferred form of flow control means for varying the discharge from the pump;

FIG. 8 is a top plan view of the flow control means of FIG. 7;

FIG. 9 is an elevational sectional view of a pump basically similar to that of FIG. 1 but modified at the liquid end for handling corrosive, or radioactive, or like, materials;

FIG. 10 is a view of the arm which actuates the replenishing valve as seen looking down along the lines X—X of FIG. 9;

FIG. 11 is a view in section of the tubular diaphragm in extended condition, as seen looking down along the lines XI—XI of FIG. 9;

FIG. 12 is a view similar to FIG. 11 but illustrating the tubular diaphragm in compressed condition; and

FIG. 13 illustrates a modified form of diaphragm.

Referring now to FIGS. 1 and 2, the reference numeral 10 identifies comprehensively a pump which, except for its protruding plungers, valve bodies, and related parts, is contained within and supported by a cylindrical housing 12 having a vertical axis. Housing 12 is shown to be comprised of two sections, an upper section 12a, and a lower section 12b which supports the upper section. The housing 12 is filled with oil to the level of fill plug 13, the oil level being indicated by the dot-and-dash line 13a. An annular sealing gasket 14 is provided to seal the junction of the upper and lower sections 12a, 12b of the housing. A drain plug 15 is provided in lower section 12b.

Mounted in housing 12, on the vertical center axis thereof, is a cam shaft 16 supported for rotation by a lower cam shaft bearing 17 and by an upper cam shaft bearing 18. The lower cam shaft bearing 17 is supported in a bearing plate 28 bolted or otherwise secured to the upper section 12a at the bottom thereof.

Keyed or otherwise fixed, to the cam shaft 16 is a worm gear 19 driven by a worm 20 keyed to a drive shaft 21 (FIG. 2) which is coupled through a suitable coupling 22 to the drive motor 23. A speed control 29 (FIG. 2) may be connected to the motor 23. Drive motor 23 may preferably be a constant speed motor, for example, a 1/8 horsepower shunt-wound motor. Speed control 29 may, for example, be a type SH-54 manufactured by the Minarik Electric Company, previously referred to herein.

Also keyed to cam shaft 16 is a barrel cam 24 having a single peripheral cam track or groove 25. While barrel cam 24 may be a solid cylindrical barrel (having the single contoured cam track or groove 25 in its peripheral surface) cam 24 is preferably in the form of an annular rim 24a connected to the hub 24b as by radial members or spokes 24c, thus forming the spider-like structure illustrated in FIG. 1.

Supported within the cam track or groove 25, at opposed locations, are cam followers 26 and 27 supported for rotation on cam-follower studs 26a and 27a, respectively. These studs are secured, as by lock nuts 26b and 27b, to the inner wall of recesses 30c and 31c provided in crossheads 30 and 31, respectively. Thrust bearings 26c and 27c are interposed between the followers 26, 27, and the inner walls of the recesses 30c, and 31c.

A pair of crossheads 30 and 31 are vertically disposed in housing 12 at opposed locations, equally distant radially from the vertically disposed cam shaft 16. The crossheads 30 and 31 are mounted for reciprocating movement in the axial direction in crosshead bushings 30a, 30b, 31a and 31b.

The cam-receiving recesses 30c and 31c of crossheads 30 and 31 may preferably be formed in enlarged portions 30d and 31d of the crossheads. These enlarged portions 30d and 31d are preferably formed integral with the cylindrical portions of the crossheads. In practice, the crossheads may be formed out of rectangular stock by turning down, as on a lathe, the cylindrical portions above

and below the rectangular portions 30d, 31d containing the cam-receiving recesses 30c, 31c.

As seen more clearly in FIG. 3, crosshead 31 may be provided with a yoke 31y for engaging a crosshead guide pin 31x for preventing rotation of the crosshead on its own axis. A similar arrangement may be employed for the other crosshead 30.

Surrounding the cylindrical portions of the crossheads 30 and 31 above the cam-receiving portions 30d, 31d, are compression springs 30e and 31e. The function of springs 30e and 31e is to reduce the peak horsepower requirement (which occurs when one plunger is on constant velocity discharge and the other plunger is at maximum suction velocity, i.e., at 290° in FIG. 6). The bearing plate 28 which supports the lower crosshead bushings 30b and 31b is located sufficiently above the floor of housing 12 to allow the crosshead to take, on the suction stroke, the fully returned position shown in FIG. 1 for crosshead 31.

The crossheads 30 and 31 project upwardly and outwardly from the housing 12 through the upper crosshead bushings 30a and 31a, and are threaded at their upper ends for receiving the crosshead nuts 30f and 31f (the latter is not shown in FIG. 1 due to the break in the drawing). Just below the crosshead nuts, catch-alls 30g and 31g are provided.

Supported above the upper housing 12a, as by liquid-end support members 32 and 33, are valve bodies 34 and 35, each having a main bore 60 and 61 in coaxial alignment with the crossheads 30, 31, for receiving the plungers 130 and 131. Plungers 130 and 131 are provided at their lower ends with collars 130a and 131a (the latter not shown) pinned to the lower ends of the plungers by pins 130b and 131b (the latter not being shown). The collars 130a and 131a are held captive within the crosshead nuts 30f and 31f, with some play allowed. In this manner, the plungers 130 and 131 are secured in a floating manner to the crossheads 30 and 31, which makes for low transverse loading on the crosshead bushings and minimizes packing wear. It is also to be noted that the weight of the plungers in the vertical pump of the present invention is on the crossheads, and not on the packing as in a horizontal pump.

Suitable stuffing box seals 36 and 37, preferably having V-type packing, are provided and are compressed and supported by gland nuts 38 and 39.

In operation, the cam shaft 16 is driven by the motor 23 through the coupling 22, drive shaft 21, worm 20 and worm gear 19. As previously indicated, motor 23 may preferably be a constant speed motor. Rotation of the cam shaft 16 drives the barrel cam 24 rotationally and the contour of its cam groove 25 causes the same followers 26 and 27 to be reciprocated up and down, thereby reciprocating the cross heads 30 and 31 and the plungers 130 and 131. The phase relations between the reciprocal motions of plungers 130 and 131 will be described in more detail below in connection with a discussion of FIGS. 4-6. Generally speaking, plunger 130 is driven upwardly (on the discharge stroke) while plunger 131 is being pulled downwardly (on the suction stroke) but, as will be discussed below, both plungers move simultaneously upward for a portion of each cycle.

Referring now to FIG. 4, the plunger 131 completes its down or suction stroke at the time marked 360° (or 0°) on the displacement curve of plunger 131. The cam groove 25 is preferably so contoured that this instant is forty degrees before the other plunger 130 completes its upward or discharge stroke. Similarly, plunger 130 completes its return or suction stroke at 360° (or 0°) on its displacement curve which is forty degrees before plunger 131 completes its upward or discharge stroke. During each of these two periods (of forty degrees each in the present example) both plungers are moving upwardly on the discharge stroke.

5

FIG. 5 depicts graphically the relative velocities at which the plungers 130 and 131 are moving at different times during the cycle. As shown in FIG. 5, for the first forty degrees, (between 0° (or 360°) and 40° on its curve) plunger 130 moves upwardly on the discharge stroke while accelerating cycloidally. Then, for a total of one hundred and forty degrees (between 40° and 180° on its curve) plunger 130 moves upwardly at constant velocity. Then, for forty degrees (between 180° and 220°) plunger 130 slows down, moving upwardly on the discharge while decelerating cycloidally. Then for the next seventy degrees (220° to 290°), plunger 130 moves downwardly on the suction stroke at constant acceleration attaining maximum velocity at 290°. Finally, to complete the cycle, plunger 130 moves downwardly for the final seventy degrees at constant deceleration, reaching zero speed at the bottom of the suction stroke (360° or 0° on its curve in FIG. 5).

The other plunger 131 moves at velocities substantially identical to those of plunger 130, but out of phase by one hundred eighty degrees, as seen in FIGS. 4 and 5. The velocity relationships are also depicted graphically in FIG. 6.

The velocities at which the plungers 130 and 131 move during the various parts of the cycle are, of course, controlled by the speed of the drive motor 23 and the contour of the cam groove 25. Since a single barrel-cam groove 25 controls both plungers, the displacement and velocity relationships between the two plungers, once established, remain invariable. There is no need to make compensating adjustments as conditions change due to cam wear or other causes, since cam wear affects both plungers alike. This is an important advantage of the single cam pump of the present invention as compared with pumps driven by two cams.

Returning now to FIG. 1, the liquid to be pumped is supplied from a reservoir (not shown) to the input ports of the valve bodies 34 and 35 by way of conduits or pipes 44 and 45, and is discharged from the output ports of the valve bodies by way of discharge pipes 46 and 47. The conduits 46 and 47 are then manifolded or merged into a common discharge or output pipe 50. In FIG. 1, pipe 37 is shown joined to pipe 46 by elbow 48 and cross pipe 49.

Valve bodies 34 and 35 are provided, at their input and output ports, with ball check valves. In FIG. 1, valve body 34 is provided with ball check valve 54 at its input port and with ball check valve 56 at its output port. Similarly, valve body 35 is provided with ball check valves 55 and 57 at its input and output ports, respectively. Cross duct 58 in the valve body 34 connects the input check valve 54 to the bore 60 of plunger 130 and also to the output check valve 56. Similarly, cross duct 59 in valve body 35 connects the input check 55 to the bore 61 of plunger 131 and also to the output check valve 57.

In operation, when, for example, plunger 131 is moving downwardly on the suction stroke, the ball of output check valve 57 is forced down on its seat and the ball of input check valve 55 is pulled above its seat by the suction forces present; and liquid from the reservoir is pulled through pipe 45 and through the open valve 55 into the cross duct 59 and bore 61 of plunger 131.

In accordance with a preferred form of the present invention, the cam groove 25 of cam 24 is so contoured that the velocity of plunger 131 is decelerating at a constant rate during the lower half of the suction stroke and slows down to a sufficiently low velocity just before it reaches the end of the down stroke to reduce the suction forces sufficiently to allow the ball of input check valve 55 to seat and close the valve, thereby cutting off the flow of liquid into cross duct 59 and bore 61.

On the up or discharge stroke of plunger 131, the ball of output check valve 57 is forced off its seat, the valve 57 opens, and liquid is forced through valve 57 and into

6

the cross duct 49, which connects the discharge side of valve body 35 with the discharge side of valve body 34.

The action of the other plunger 130 is out of phase with that of plunger 131 by 180° but is otherwise similar to that just described.

The cam groove 25 of single cam 24 is also so designed that the up or discharge stroke of each plunger occupies 220 degrees and the down or suction stroke only 140 degrees. Thus, for two period of forty degrees each, both plungers are pumping on the up or discharge stroke. During these two forty-degree periods, the velocity of the one plunger which is approaching the top of its stroke is decelerating cycloidally whereas the other plunger is just starting up and is accelerating cycloidally, all as depicted graphically in FIGS. 4, 5 and 6. As the midpoint of the forty-degree overlap, just referred to, both plungers are moving at the same velocity, the velocity of each being one-half the velocity at which the plungers move during the constant velocity portions of the up or discharge stroke. As a result, the discharge or flow of liquid pumped through the common output duct 50 is at a steady and constant rate at all times in the cycle.

Reference is now made to FIG. 2 which is a top plan view looking down along the line II—II of FIG. 1. Crossheads 30 and 31 (which drive plungers 130 and 131) are shown occupying opposed radial positions. The vertical metering pump of the present invention is not limited, however, to two crossheads and two plungers. The pump may, if desired, be equipped with one or more additional pairs of crossheads and plungers, all driven from the same barrel cam 24 and from the same cam groove 25. The location of one such additional pair of crossheads and plungers is indicated in FIG. 2 by the dotted-line representations indicating an additional pair of plungers 230, 231. The crossheads of this additional pair likewise occupy radially opposed positions.

The discharge or output of each plunger of a pair is combined with the discharge of the other plunger of the pair to provide a steady constant flow of liquid. The liquid being pumped by one pair of plungers may be the same type as, or different from, the liquid being pumped by another pair of plungers of the same pump. In either case, i.e., whether the same or different liquids are being pumped by two pairs of plungers of the same pump, the discharge of one pair of plungers may be manifolded or combined with that of another pair of plungers of the same pump to provide a combined steady flow of the same liquid or of a mixture of different liquids.

According to another aspect of the present invention, novel flow control means are provided for controlling the discharge of a single-pair-of-plunger pump driven by a constant speed motor, or for controlling the combining of the discharge of two pairs of plungers of the same pump. By the use of such flow control means, it is possible to adjust the discharge of the pump while the pump is running, as well as while the pump is stopped. It is also possible to vary the mix of different liquids while the pump is running at constant speed. Apparatus for accomplishing such results is illustrated in FIGS. 7 and 8 of the drawing.

Referring now to FIGS. 7 and 8, FIG. 7 is an elevational view of the pump 10 of FIGS. 1 and 2 equipped with flow control mechanism for controlling the total discharge from two pairs of plungers. One pair of plungers may, for example, be plungers 130 and 131 and the other pair of plungers may be plungers 230 and 231 (FIGS. 1 and 2). The valve bodies 34, 35 and the discharge or output conduits 46, 47, 49 and 50 of plungers 130 and 131 are merely illustrated diagrammatically in FIGS. 7 and 8 since these conduits have already been described in more detail in connection with FIG. 1.

The corresponding valve bodies 234, 235 and the corresponding discharge conduits 246, 247, 249 and 250 of the second pair of plungers 230 and 231 are, in FIG. 7, merely shown diagrammatically also. Their structure

and arrangement may be similar to that of the valve bodies 34, 35 and discharge conduits 46, 47, 49 and 50 of the first pair of plungers, already described in detail in connection with FIG. 1.

The purpose and function of the flow control mechanism shown in FIGS. 7 and 8 is to control how much of the flow in discharge conduit 250 from one pair of plungers is added to that discharging through the conduit 50 from the other pair of plungers. The control mechanism includes a valve body 234 having therein two ball check valves 254 and 256.

When pump 10 is running, there is normally an upward pressure against the ball of check valve 256, and an equal pressure downwardly against the ball of check valve 254. Accordingly check valve 254 is normally closed and check valve 256 is normally open. Under these conditions, all of the liquid being discharged through conduit 250 will be combined and mixed with that discharging through conduit 50, as is indicated by the flow arrows in FIG. 7.

If, however, the ball of the bypass check valve 254 be raised off its seat, as by rod 262, the upward pressure against the ball of check valve 256 will be relieved and this check valve will be forced closed by the discharge pressure from conduit 50. All of the liquid discharging through conduit 250 will now pass through the open check valve 254 into the bypass conduit 251 which is connected to the suction reservoir of the pump. Below bypass conduit 251 is a diaphragm seal 252 flexibly connected to the valve rod 262 and sealing the bypass system from the atmosphere. Whether or not the ball of the bypass check valve 254 is raised off its seat is controlled by cam plate 258 and cam follower 257, as will now be described.

Cam follower 257 is kept in pressure contact with the horizontally disposed cam plate 258 by spring 259. Cam plate 258 is a rectangular plate (FIG. 8) with its surface portion divided into two generally triangular halves, one of which 261 is elevated above the other half portion 260 of the plate. The two half portions 260, 261 are joined by an inclined ramp or camming surface 273 whose angle of inclination determines the rate at which the ball of check valve 254 is raised from, and lowered to, its seat. As seen in the top plan view of FIG. 8, the camming surface 273 between the low and elevated portions 260, 261, follows a harmonic curve which is the complement of the slider-crank drive. This enables linear adjustment of the flow control, as will be described.

As indicated previously, when cam follower 257 rests on the lower portion 260 of cam plate 258, the ball of check valve 254 is on its seat and check valve 254 is closed. When, however, the cam follower 257 rises up the camming ramp 273 to the elevated portion 261 of cam plate 258, the rod 262 is moved upwardly against the action of compression spring 259 and the ball of the check valve 254 is lifted off its seat, thereby opening the bypass check valve 254. The check valve 254 then remains open until the cam follower 257 moves down the camming ramp 273 to the lower level portion 260 of the cam plate.

Cam plate 258 is mounted for rapid reciprocating movement in the horizontal plane of the plate, and for lateral adjustment in the transverse horizontal direction. In the structure shown in FIGS. 7 and 8, the cam plate 258 is dovetailed in a crosshead 263 which is supported and guided for back and forth motion on a pair of guide rods 269 and 270 supported by the frame 278. The crosshead 263 is driven by a connecting rod 265 to which it is secured by wrist pin 264. Thus, the cam plate 258 is reciprocated by connecting rod 265 and a crank arm 266 to which it is connected by crank pin 267. Crank arm 266 is keyed or otherwise fixed to a stub shaft 268 which is connected to and forms an upward extension of the cam shaft 16 of pump 10. Thus, when the pump 10 is driven, the extension 268 of cam shaft 16 drives

the crank arm 266 rotationally at the same speed as the pump, and cam plate 258 is reciprocated in timed relation with the crossheads and plungers of the pump.

When cam plate 258 occupies the transverse or lateral position illustrated in FIG. 8, in which cam follower 257 is centered laterally on the plate, as the cam plate is moved back and forth by the connecting rod 265, the cam follower 257 is moved relatively back and forth along the line indicated 50% in FIG. 8 since the follower 257 will be in engagement with the lower-level portion 260 of the cam plate 258 for one-half of the cycle and will be in engagement with the elevated portion 261 of the cam plate during the other one-half portion of the cycle. Thus, bypass check valve 254 will be closed for one-half of the cycle and will be open during the other one-half of the cycle.

When bypass check valve 254 is closed, check valve 256 is open and the entire output of discharge conduit 250 flows through check valve 256, and may be manifolded with the output of the other discharge conduit 50. When the ball check valve 254 is lifted from its seat by rod 262, and the check valve 254 is open, the other check valve 256 closes, and the entire discharge through conduit 250 now flows through the bypass check valve 254.

The time interval during which the bypass check valve 254 is open during the cycle may be varied, and is controlled by the transverse or lateral position of the cam plate 258. In FIG. 8, if the transverse position of cam plate 258 be adjusted in one direction, the time interval during which the bypass check valve 254 would be open would be increased, and if adjusted in the other direction, the time interval would be decreased. For example, if cam plate 258 be so adjusted transversely that the cam follower 257 moves back and forth along the line marked 100% in FIG. 8 (the lowermost line as viewed in FIG. 8), cam follower 257 would be on the elevated portion 261 of the cam plate during the entire cycle, and thus the bypass check valve 254 would be open the entire cycle.

If, on the other hand, the cam plate 258 be so adjusted transversely that as it is moved back and forth, the cam follower 257 moves along the line indicated as 0% in FIG. 8 (the topmost line as viewed in FIG. 8) then cam follower 257 would be on the lower level portion 260 of the cam plate 258 during the entire cycle and the bypass check valve 254 would be closed during the entire cycle. Or, if cam plate 258 be placed in such transverse position that cam follower 257 moves back and forth along the line indicated as 30% in FIG. 8, then bypass check valve 254 would be open for 30% of the cycle and closed for 70% of the cycle.

Any suitable means may be provided for adjusting linearly the position of cam plate 258 in the transverse direction. In FIGS. 7 and 8, the dovetailed cam plate 258 is movable transversely relative to the reciprocating crosshead 263 by means of side blocks 271 and 272 which abut against the sides of the cam plate 258. The side blocks 271 and 272 are mounted on a crosshead 276 which is dovetailed in a support member 277 below the cam plate 258. The support member 277 is supported by the frame 278. The side blocks 271, 272 are movable in the lateral direction, thereby to move the cam plate 258 laterally, by means of an adjusting screw 280 (FIG. 8). Screw 280 is shown to be supported at one side of the mechanism by an L-bracket 281 fixed to frame 278. Bracket 281 has a horizontal base portion 281a and a vertical portion 281b. The vertical portion 281b has a threaded hole for receiving the adjusting screw. A collar 282, pinned to screw 280 at its inward end, is held captive in a recess 283 in side block 271. Thus, screw 280 pushes or pulls the side blocks 271, 272, according to the direction in which the adjusting screw 280 is turned. The threads of adjusting screw 280 preferably have a constant pitch so that linear adjustments may be made to the pump output. For example, if screw 280 has ten threads per inch, and if the total transverse adjusting

travel of cam plate 258 is two inches, each single turn of the screw will move the cam plate one-twentieth of two inches or one-tenth of an inch. Assume that at the start of the lateral adjustment, the cam plate 258 is at the extreme near position in which the cam follower 257 runs back and forth along the line marked 100% in FIG. 8 (the full bypass or bleed-off condition). Then, if the adjusting screw be turned say six revolutions, the cam plate 258 will be moved six-tenths of an inch, or three-tenths of the full transverse travel of the cam plate which will cause the cam follower to travel back and forth along the line marked 70% in FIG. 8. When the cam follower 257 travels along the 70% line, the bypass valve 254 is open for 70% of the cycle and closed for 30%. The adjustment is linear because the camming surface 273 follows a harmonic path which is the complement of the harmonic stroke motion of the crank-driven connecting rod.

It will be seen from the foregoing description that the present invention provides a single cam vertical pump capable of operation as a constant speed pump requiring little or no compensation or adjustment due to wear; that the pump is adapted to have combined with it a pump-actuated flow-control means adjustable to vary the flow volume of the discharge; that such adjustment may be made while the pump is running as well as while stopped; that the pump may be constructed and operated to deliver two or more continuous streams which may be merged into a common discharge stream; and that the pump-actuated flow control means may be used to control the flow of one of said two continuous streams, thereby to vary in a controlled manner the total discharge from the pump operating as a constant speed device.

The vertically disposed metering pump described and claimed herein, a pump in which the crossheads and plungers move vertically, has several advantages over the conventional metering pump in which the crossheads and plungers move laterally. In the vertical pump, the oil level is below the point at which the crossheads emerge from the housing and as a consequence no oil seals are required at the point where the crossheads emerge. Secondly, packing wear is reduced since the vertical plungers are self-centering or self-aligning and the weight of the plunger is on the crosshead, not on the packing as it is in the case of a horizontal pump. Third, with the plungers operating vertically, there is no air entrapment problem in the plunger chamber since air does not go down. If air is pulled in on the suction stroke, it is pushed out on the discharge stroke; it is not trapped. Trapped air is compressible and is very undesirable in metering applications. In all of these respects, the vertical metering pump of the present invention is superior to the conventional horizontal pump.

Also, in the pump of the present invention, the employment of a cam to reciprocate the plungers is advantageous. By properly designing the cam track contour to be compatible with the proper functioning of both the suction and discharge ball check valves, a pump is provided which delivers very accurately metered flow. By properly designing the cam at the start of the suction stroke (constant acceleration of the plunger) the optimum NPSH (not positive suction head) requirements can be obtained.

The bypass type of flow control means described in the present application allows the vertical pump of the present invention to be driven as a constant speed device, with the pump-actuated cam plate controlling the net flow from the pump. This structural arrangement has several advantages over conventional lost-motion devices. First, there is no over-pumping such as is caused in a lost-motion arrangement when the plunger suddenly stops on the return stroke. Secondly, there is no shock; the new pump-actuated cam plate, used with a constant speed motor driven pump, provides variable net flow without shock. And good seating action of the bypass ball check valve is ob-

tained by properly designing the camming slope of the cam plate (where the depressed surface joins the elevated portion).

The bypass flow control feature may be used with a constant speed pump having a single pair of crossheads and plungers to vary the output; or it may be used with a constant speed pump having two (or more) pairs of crossheads and plungers to vary the output of one or both pairs of plungers, to control the net flow through the manifolded output duct.

The pump of the present invention may also be readily modified for the pumping of corrosive, radioactive, and like materials without the possibility of leakage to the atmosphere. Such a pump is illustrated in FIGS. 9-13.

In FIG. 9, instead of the liquid ends shown in FIG. 1, valve bodies 334 and 335 are employed. Each valve body may be contoured inwardly, as at 334a and 335a, in order to reduce the size of the chambers 360, 361, containing the motion-transmitting liquid 370, 371.

Each valve body 334, 335, is provided with an assembly comprising, in the case of valve body 334, suction ball check valve 354, discharge ball check valve 356, and the tubular diaphragm 364 later described in further detail. Valve body 335 is similarly equipped with ball check valves 355 and 357, and with a tubular diaphragm 365.

The assembly described above may be assembled outside of the valve body, and then dropped into the motion-transmitting fluid chamber. Flanges 376, 377 are then connected at the top.

In FIG. 9, the cam shaft 16 is provided with an extension 316 having at its upper end a cam arm 317 which when driven rotationally, as indicated by the arrow in FIG. 10, cams open the ball check replenishing valve 358 for a portion (10°) of the cycle.

In FIG. 9, the plunger 130 driven by the crosshead 30 is shown at the end of its suction stroke, while the other plunger 131, driven by the crosshead 31, is near the forward end of its discharge stroke.

When the pump is intended for use in pumping corrosive, radioactive and like materials, as now being described, cam 24 of FIG. 1 is replaced with a cam (not shown) to provide a dwell period of 10° at the end of the suction stroke of each piston, thereby shortening the durations of the suction periods from 140° to 130°. Stated another way, the cam is so contoured that the suction stroke of each piston terminates at 350° (instead of at 360° as illustrated in FIG. 4) and a dwell period extends from 350° to 360°. During this 10° dwell period, the cam arm 317 depresses the spring-loaded ball check valve 358, thereby opening chamber 360 to the reservoir 362 from which the motion-transmitting fluid 370 is obtained. In this manner, reservoir 362 replenishes, in chamber 360, any motion-transmitting liquid which has leaked past the spring-loaded packing 336. In addition, it is pointed out that any air which may have entered the chamber 360 is dispelled past the valve 358 by the tendency of the air to rise. This is an important feature of the vertical type of pump. Air is not trapped.

In FIG. 9, each of the valve bodies 334, 335 is provided with a tubular diaphragm 364 and 365, respectively, made from Viton or rubber-backed Teflon or other satisfactory material. In FIG. 9, tubular diaphragm 364 is shown in its extended condition, which is the condition it acquires during the suction stroke, while the other tubular diaphragm 365 is shown in its compressed condition, which is the condition it acquires during the discharge stroke.

The corrosive, or radioactive, or like fluid 382 is delivered into the interior of the tubular diaphragms 364, 365, through conduits 344 and 345, respectively. Ball check valves 354, 355, are open during the suction stroke and closed during the discharge strokes of the respective pistons. During the suction strokes, the tubular diaphragms fill to capacity. Any addition void in the motion-transmitting-fluid chambers 360, 361, which may occur

during the suction strokes is made up from the fluid from reservoir 362, 363.

FIGURES 11 and 12 show in cross section the tubular diaphragms 364, and 365, respectively, the diaphragm 364 being shown in fully extended condition, and the diaphragm 365 in collapsed condition. Each of the tubular diaphragms is provided with three bent pins, identified collectively by reference numeral 366 for diaphragm 364, and by the reference numeral 367 for diaphragm 365. These bent pins are disposed 120° apart; their function is to radially position the diaphragm and to restrict its position in collapsed condition.

To assist the tubular diaphragms 364, 365, to expand during the suction stroke, each diaphragm is provided with three cantilevered flat springs, identified collectively as 380 for diaphragm 364, and as 381 for diaphragm 365. These flat springs, which in a typical case may be about ¼" wide, 1½" long and 0.020" thick, are disposed at 120° spacings outside the tubular diaphragms and are trapped by resilient bands, 384 for diaphragm 364, and 385 for diaphragm 365. At the end of the suction stroke, when the diaphragm is fully extended, the force exerted by the springs is almost negligible, but at the end of the discharge stroke, when the tubular diaphragm is in compressed condition, each spring exerts a radial outward force of about one pound. While the force exerted at the end of the suction stroke is ordinarily very small, as indicated above, the leaf springs do function to keep the chamber 360, 361 at equilibrium and to force any excess motion-transmitting fluid (resulting, for example, from an increase in temperature) from chambers 360 or 361 into the reservoirs 362 or 363 at the end of the suction stroke.

It is to be noted that the difference in volume of the corrosive fluid in the tubular diaphragms at the end of the suction stroke, as compared with the volume at the end of the discharge stroke, is equal to the volume displaced by the plungers 130 and 131, and that this is constant with the type of pump described and claimed in the present application. It will be understood that the volume displaced by the plunger is constant since the stroke is not changed to vary the discharge rate. The discharge rate is varied by varying the speed of the pump, not by varying the stroke.

The pump of the present invention, as modified for pumping corrosive, radioactive, and like materials, as compared with prior art pumps of which I am aware, has the following advantages:

(1) The replenishing valves 358, 359, do not have to act as relief valves since the pump is on constant stroke and since the leaf springs 380, 381, keep the chambers 360, 361, at equilibrium.

(2) The Viton or rubber-backed Teflon tubular diaphragms 364, 365, contain the corrosive fluid and prevent it from contacting all components of the pump except the ball check valves 354, 355, 356 and 357. Thus, only the ball check cartridges need be made of corrosion-resisting materials.

(3) Replenishing of the motion-transmitting fluid occurs during the dwell period provided for in the cam which drives the crossheads. Replenishing does not occur during either the suction or discharge periods.

(4) The packings 336, 337 may be spring-loaded, as by spring 368, 369; no adjustment is needed since the plungers are vertically disposed and the springs cannot trap air.

While the tubular form of diaphragm 364, 365, illustrated in FIGS. 9, 11, 12, and described above is preferred (for pumping corrosive, radioactive and like materials) a flat circular diaphragm structure, such as is illustrated in FIG. 13, may also be employed. In FIG. 13, the corrosive fluid enters, through the ball check valve 354, into a passageway 390 formed by the walls 392 of the valve body and on one side by the diaphragm 394 made from Viton or rubber-backed Teflon or other satisfactory materials. A tension spring 395 assists the dia-

phragm 394 to return from the compressed condition into which it is placed during the discharge stroke.

While the preferred embodiments of this invention have been described in some detail, it will be obvious to one skilled in the art that various modifications may be made without departing from the invention as hereinafter claimed.

Having described by invention, I claim:

1. A liquid metering pump comprising: a housing; a cam shaft vertically disposed and mounted for rotation in said housing; a single cylindrical cam having a single annular peripheral cam track, said cam being fixed to said cam shaft for rotation therewith; first and second vertically-disposed crossheads in opposed radial positions relative to said cam shaft; a plunger secured to the upper end of each crosshead; a valve body for each plunger, said valve body having therein at least one ball check valve in which the ball moves vertically to seat; a cam follower secured to each crosshead and adapted to follow said cam track, said cam track having such contour that when said cam shaft is driven rotationally, said cam followers, crossheads, and plungers are driven up and down in a reciprocating motion; and drive means for driving said cam shaft, said cam track having such contour that during a portion of the cycle both plungers are moving upwardly on the discharge stroke, with one of said plungers nearing the end of its discharge stroke and decelerating cycloidally, and the other of said plungers just starting its discharge stroke and accelerating cycloidally.

2. A liquid metering pump comprising: a housing; a cam shaft vertically disposed and mounted for rotation in said housing; a single cylindrical cam having a single annular peripheral cam track, said cam being fixed to said cam shaft for rotation therewith; first and second vertically-disposed crossheads in opposed radial positions relative to said cam shaft; a plunger secured to the upper end of each crosshead; a valve body for each plunger; drive means for driving said cam shaft; a cam follower secured to each crosshead and adapted to follow said cam track, said cam track having such contour that when said cam shaft is driven rotationally, said cam followers, crossheads, and plungers are driven up and down in a reciprocating motion, said cam track contour being such that during a portion of the cycle both plungers are moving upwardly on the discharge stroke, with one of said plungers nearing the end of the discharge stroke and decelerating cycloidally, and the other of said plungers just starting its discharge stroke and accelerating cycloidally; a discharge conduit; flow control means actuated by said pump for varying the flow through said discharge conduit, said flow control means including a bypass conduit, bypass valve means connecting said discharge and bypass conduits, and cam means connected to and actuated by said pump for controlling the condition of said bypass means according to the angular position of said cam shaft, said flow control cam means also including a cam plate having an elevated portion, a depressed portion, and a camming surface therebetween, a cam follower, means connecting said cam follower to said bypass valve means, means maintaining said cam follower in engagement with said cam plate, and means, including a crank and connecting rod, connecting said pump cam shaft to said cam plate for reciprocating said cam plate in timed relation to the rotation of said pump cam shaft.

3. A metering pump as claimed in claim 2 further characterized in that the camming surface of said cam plate follows a harmonic path which is the complement of the harmonic motion of said crank-driven cam plate, and in that adjusting means are provided for adjusting linearly the position of said cam plate in the direction transverse to that of said reciprocation for adjusting linearly the flow through said discharge conduit.

4. A metering pump as claimed in claim 3 characterized in that said adjusting means includes means for mounting said cam plate for adjusting movement in said

transverse direction, and further characterized in that said adjusting means includes a linearly threaded adjusting screw adapted for manual operation.

5. A liquid metering pump comprising: a housing; a cam shaft vertically disposed and mounted for rotation in said housing; a single cylindrical cam having a single annular peripheral cam track, said cam being fixed to said cam shaft for rotation therewith; at least two pairs of vertically-disposed crossheads, the crossheads of each pair occupying opposed radial positions relative to said cam shaft; a plunger secured to the upper end of each crosshead; a valve body for each plunger; drive means for driving said cam shaft; a cam follower secured to each crosshead and adapted to follow said cam track, said cam track having such contours that when said cam shaft is driven rotationally, the cam followers, crossheads and plungers are driven up and down in a reciprocating motion; said cam track contour being such that during a portion of the cycle both plungers of each pair are moving upwardly on the discharge stroke with one of said plungers of the pair nearing the end of the discharge stroke and decelerating cycloidally while the other plunger of the pair is just starting the discharge stroke and is accelerating cycloidally; a separate discharge conduit for each pair of plungers; means combining the separate discharge conduits into a common discharge conduit; and flow control means actuated by said pump for varying the flow from one of said separate discharge conduits into said common discharge conduit, said flow control means including a bypass conduit, bypass valve means for connecting said bypass conduit to said one discharge conduit, and cam means connected to and actuated by said pump for controlling the condition of said bypass valve means according to the angular position of said pump cam shaft.

6. In a metering pump; a cam shaft vertically disposed and mounted for rotation; a single cylindrical cam fixed to said cam shaft and having a single peripheral cam track most of which is outside a plane normal to said cam shaft; at least one pair of vertically-disposed crossheads equally distant radially from said cam shaft; a cam follower secured to each crosshead; means for driving said cam shaft and cam rotationally to drive said cam followers and crossheads vertically reciprocally; a plunger connected to each crosshead; a liquid-end for each plunger, said liquid-end including a chamber for motion-transmitting fluid, each chamber being adapted to be entered by one of said plungers to actuate the motion-transmitting fluid therein, each chamber having a valved flexible-walled passageway therethrough for passing corrosive and like liquids requiring isolation from metering pump, the flexible wall of said passageway being adapted to be moved inward and outward by said motion-transmission fluid in response to reciprocation of the plunger, thereby to pump said isolation liquid; a reservoir of motion-transmission fluid; a pressure-insensitive replenishing valve providing communication between said reservoir and said chamber;

and means connected to said cam shaft for opening said replenishing valve during a portion of each rotation of said cam shaft.

7. In a metering pump as claimed in claim 6 further characterized in that said cam track has such contour that each plunger has a dwell period at the end of the suction stroke.

8. In a metering pump as claimed in claim 7 further characterized in that said dwell period coincides with the period during which said replenishing valve is opened.

9. In a metering pump as claimed in claim 6 characterized in that said flexible-wall passageway is formed by a tubular diaphragm.

10. In a metering pump as claimed in claim 9 further characterized in that cantilever leaf springs are connected to said diaphragm for assisting said tubular diaphragm to move outwardly during the suction stroke of the plunger.

11. In a metering pump as claimed in claim 10 further characterized in that the tubular diaphragm is provided with an interior frame for limiting the collapsed condition of said diaphragm during the discharge stroke of the plunger.

12. In a metering pump as claimed in claim 10 further characterized in that said tubular diaphragm takes the form of two frustrums of cone at the end of the suction stroke with the large diameter at the center and the small equal diameters at each end.

13. In a metering pump as claimed in claim 11 further characterized in that said interior frame comprises three bent pins at 120° spacing.

14. In a metering pump as claimed in claim 11 further characterized in that said tubular diaphragm takes the form of two frustrums of cone at the end of the suction stroke with the large diameter at the center and the small equal diameters at each end.

15. In a metering pump as claimed in claim 13 further characterized in that said tubular diaphragm takes the form of two frustrums of cone at the end of the suction stroke with the large diameter at the center and the small equal diameters at each end.

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