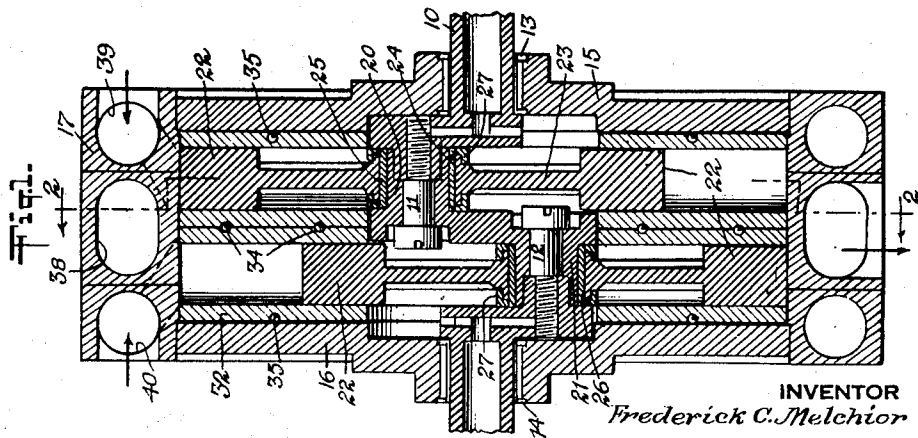
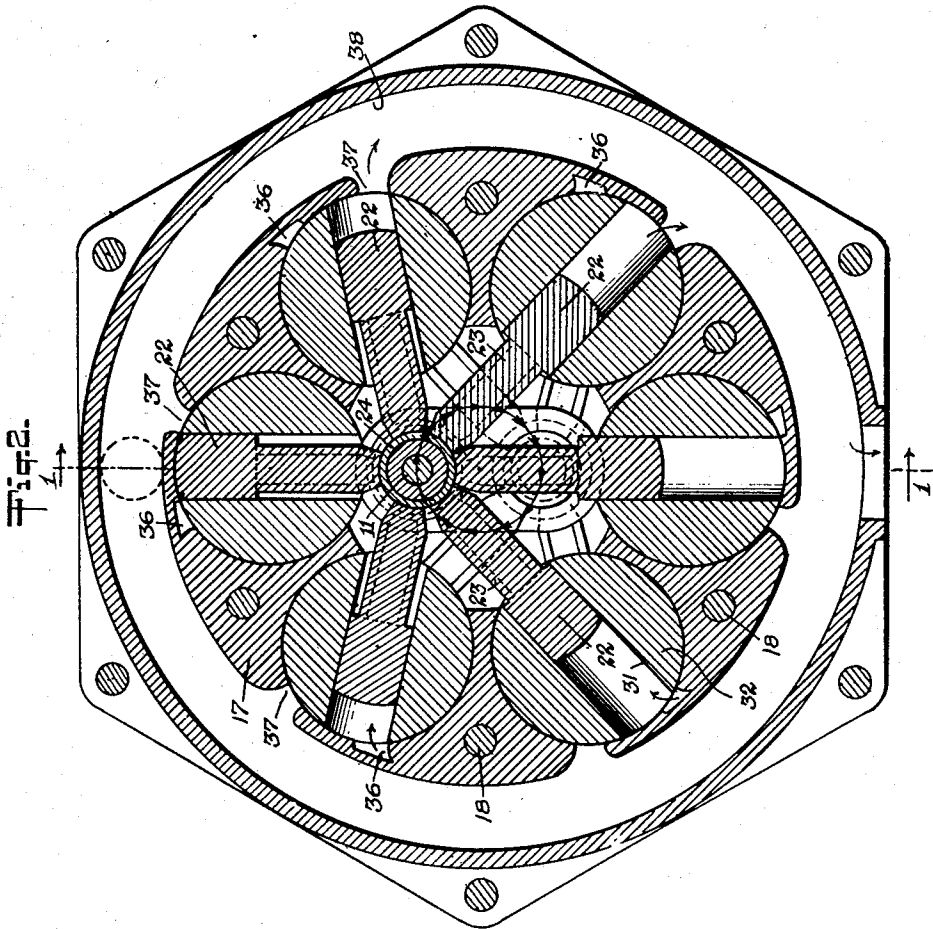


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VALVELESS RECIPROCATING PUMP

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VALVELESS RECIPROCATING PUMP

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1 Claim. (Cl. 103—159)

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This invention relates to pumps of the reciprocating type, and more specifically to the valveless variety in which the clearance or head space may be eliminated when desirable for a particular application.

One object of the invention is to render possible a combination of high capacity and high pressure in a unit of extraordinary compactness.

Another object is to obtain optimum volumetric efficiency, through elimination of clearance and valve losses common in other designs.

A further object of the invention is to bring about relatively slow piston speeds for improved flow conditions and overall efficiency, as well as reliability and prolonged life.

A still further object is to provide for a perfectly smooth flow of the fluid to be pumped, and thus to eliminate appreciable pulsation and pressure fluctuation.

It is also an object of this invention to provide a basic design of unusual versatility, thus making available a pump of superior performance and compactness for a great variety of applications.

Generally, the performance of any device or machine may be said to be the resultant of compromise between various component factors, mostly of a conflicting nature. Thus, in a pump, performance is expressed in terms of delivery capacity and pressure, said factors being substantially functions of size, weight and efficiency, and while applied practice to date exhibits a great diversity of design principles, these are all subject to such compromise. For example, a centrifugal pump or compressor may have a relatively large delivery capacity for its size and weight, but the ultimate pressures run low as compared with those possible in a reciprocating piston pump where the delivery capacity is correspondingly modest. In cases where a combination of high pressure and great volume is desired, it will be necessary to resort to multi-stage units of considerable bulk and weight.

It is not suggested or implied herewith that the subject invention is immune to the laws and limitations of such compromise as referred to above; however, as will be seen from the ensuing description, the particular features of its preferred structure—which are believed to be novel—will be instrumental in combining more favorably the various conflicting factors such as delivery capacity, pressure, compactness, light weight and efficiency, and will, therefore, also accomplish the foregoing desiderata in a superior manner.

The exact nature of the invention, together with its several objects, may best be understood from

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the accompanying drawings embodying the various details of apparatus for carrying the invention into effect, as described in the following specification.

Referring now to said drawings:

Fig. 1 is a longitudinal section in the vertical plane through the crankshaft axis of a pump designed in accordance with the invention; and

Fig. 2 is a cross-section of said pump, normal to the axis of the crankshaft and offset on the dot and dash lines 2—2 in Fig. 1.

As is readily seen, the design chosen to illustrate my invention is of the radial, multi-cylinder, double row type, showing in some respects similarities to radial aircraft engine practice. However, it should be clearly understood that the number of cylinders and the manner in which they are interconnected and jointly operated will in no way affect the scope and usefulness of my invention which may apply to various accepted arrangements such as in-line or opposed, single or multi-stage, according to space and performance requirements. The illustration as here presented is, therefore, only to be construed as an example of what may well be a preferred execution for optimum compactness of installation on a common shaft with a suitable prime mover, and it must in no case be interpreted as constituting any limitation relative to scope and usefulness of the invention as it is hereinafter set forth.

Referring now to Fig. 1, there is shown a two-throw crankshaft 10 of accepted design, comprising three members rigidly held together by means of bolts 11 and 12. This crankshaft is mounted in bearings 13 and 14 lodged in end plates 15 and 16 which are affixed to the frame casting 17 by means of radially spaced bolts 18. On each of the crankpins 20 and 21 is a plurality of radially extending plungers or pistons 22. These pistons are integral with what might be termed their respective piston rods 23 which are provided with flanged feet 24 fitted to the crank-bearings 25 and 26 and held in place on said bearings by means of ring shaped bushings 27.

Reference to Fig. 2 will disclose that the frame casting 17 is provided with circular recesses in which are mounted cylindrical discs 32. These discs are free to rotate in said recesses but axial movement thereof is prevented. Actually there are two discs in each aperture as will be seen in Fig. 1. These discs are provided on their facing surfaces with registering annular grooves adapted to accommodate ball bearings 34. On their opposite or outside surfaces these discs are provided with a center hole or socket adapted to carry ball bear-

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ings 35. The ball bearings 34 hold each pair of discs 32 in perfect reciprocal alignment. The single balls 35 which are centered in the outer faces of the discs are left to find their own locations on the hardened inner surfaces of the end plates 15 and 16, said balls protruding sufficiently for the discs 32 to clear the end plates, at the same time obtaining against said end plates a slight pressure or touch fit for perfect and rigid axial alignment.

Each of the discs is provided with a diametrically bored and centered hole 31 which is adapted to accommodate one of the pistons 22. As the rigid acting rods of the pistons follow the motion of the crankpins, a reciprocating movement of the pistons will take place. In addition to this reciprocating movement, a pivotal or rocking movement will take place by reason of the fact that uniform lateral movement of the pistons is prevented by the discs 32 in which they are mounted.

It is evident from the foregoing that as the crankshaft is put into motion by conventional means, a reciprocating and rocking movement of the pistons will take place, and at the same time the discs which provide bearings for the pistons will engage in angular movement, first in one direction and then in the opposite direction. As will hereinafter become clear, it is this reciprocating movement of the pistons that provides the pumping action of the pump, and it is this angular movement of the discs that opens and closes the inlet and outlet passages of the pump.

Considering now Fig. 2, it will be seen that adjacent the outer part of the periphery of each of the rocking discs 32 are two passages or ports 36 and 37, respectively, formed in the frame casting 17. Port 36 is the inlet port into the disc hole or piston chamber 31 and port 37 is the outlet from said chamber. It will be understood that as the discs engage in their respective angular movements, as above described, these ports will alternately be opened and closed. For example, at the very top of Fig. 2 both ports 36 and 37 are closed, that is there is no communication between either of said ports and the piston chamber 31. The discs immediately below and to the right of the topmost disc in Fig. 2, however, blocks communication between inlet port 36 and the piston chamber, but outlet port 37 is open to said chamber. On the other hand the disc immediately below and to the left of the topmost disc in Fig. 2 obstructs communication between outlet port 37 and the piston chamber, but passage between inlet port 36 and said chamber is provided. It is clearly evident, therefore, that the angular or rocking motion of the discs 32 controls the opening and closing of ports 36 and 37.

Reference to Fig. 2 will show that ports 37 communicate with an annular passage or channel 38. Reference to Fig. 1 will show that this channel occupies a central position relative to the two banks of discs therein shown. In other words this one channel affords communication with ports 37 of both rows of discs. The ports 36, however, of one of the rows of discs 32 communicate with annular passage or channel 39, and the corresponding ports of the second bank of discs communicate with a similar channel 40. Since channels 39 and 40 each serve a single bank of discs and channel 38 serves both banks, the cross-sectional area of the latter channel is approximately equal to the cross-sectional areas of channels 39 and 40 combined.

The manner of operation may best be under-

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stood from a study of Fig. 2. While it is axiomatic that rotation of the crankshaft 10 will cause the centers of crankpins 20 and 21 to follow the periphery of a circle, as indicated by arrowheads, it is also clear that the plunger feet 24 gliding on bearings 25 and 28 and being snugly held to same by ring bushings 27, will describe an identical motion. Inasmuch as the connecting rods 23 are rigid integral parts of the plungers 22, it follows that any peripheral motion, such as hereinafter referred to, will cause the rocking discs 32 to turn pivotally about their center axis which is parallel to that of the crankshaft. For clarity of orientation in reference to the illustration, it may be well to define "horizontal" as any direction or line parallel to the axis of the crankshaft, and "vertical" as any direction or line normal thereto, all in the plane of the drawings. As motion of the crankpin 20 commences along the aforesaid periphery, we find the uppermost plunger 22 in its vertical top position with its rocking disc 32 closing off the ports 36 and 37. At the outset, the motion is entirely in the direction of the horizontal tangent, there being no downward component of motion and therefore no downward motion of the plunger; however, upon appreciable progress—with the disc 32 turning in the opposite direction—port 36 is partly uncovered at the same time as downward motion sets in, causing the plunger to follow. With further progress it is evident that the downward component increases, as does the downward velocity of the plunger and the exposure of the port 36, while the horizontal component decreases until it reaches zero at a crank position approaching 90 degrees where, inevitably, the downward component (plunger speed) is at its maximum, as is also the opening of the port 36. Beyond the 90 degree position a reversal takes place in that the downward component decreases, with corresponding decrease in plunger speed, at the same time as horizontal motion is reversed, with the result that the rocking disc 32 also reverses its motion and begins to close up the port 36. At the 180 degree or 6 o'clock position the downward component has reached zero with the plunger in the vertical bottom position; the horizontal component of motion has attained its maximum, and both ports 36 and 37 are closed. Beyond this point there is a reversal of the vertical component, causing an upward motion of the plunger, and also a decrease in horizontal motion as the disc 32 moves over and exposes the port 37. At the 270 degree position the horizontal component again becomes zero, the opening of the port 37 at its maximum as is the vertical component and the upward velocity of the plunger. Again there is a reversal of the horizontal motion, a decrease in vertical velocity, and the cycle is completed with the plunger in the vertical top position, ports closed.

From the foregoing it is clear that there is inherently, at all times, a direct proportion between plunger speeds and port openings. This makes for consistently uniform flow conditions throughout the operating range, which in turn promotes improved volumetric efficiency. Likewise, other specific features of the subject design and operating principle are now apparent, and we may therefore review the various objects of the invention for the purpose of determining the degree and manner in which they have been fulfilled.

Considering first the question of capacity versus pressure we may note that, say, a bore of

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1¼" and a stroke of 2¼" would result, in a total net displacement for the 12 cylinders of 33 cu. in. Thus, such a pump with a diameter of about 16" would deliver approximately 13 gallons per minute per 100 R. P. M. As the rocking discs follow the crank motion, eliminating side thrust, the forces are always normal, and so there is no practical limit to the pressure for which the pump may be designed.

Volumetric efficiency depends on several factors, some of which have already been referred to above, and it has been clearly illustrated how the rocking discs take the place of conventional valves and thus eliminate failures and leakage common in other designs. Furthermore, for use as a compressor, the subject pump may be designed with convex plunger heads, shaped and polished to maintain in their vertical top or zero positions the same tolerances as the discs 32 in frame casting 17. This will completely eliminate all clearance or head space which is otherwise detrimental to volumetric efficiency because of re-expansion of trapped gas.

Permissible piston speeds may vary considerably according to the nature of the application and the fluid to be pumped; however, regardless of such variation, it is obvious that for a given delivery capacity piston speeds are lower where a greater number of cylinders are employed. The subject invention is particularly advantageously suited for multi-cylinder design, and so we find that any given dimensions and R. P. M. will render piston speeds substantially below those which would prevail in conventional types for the same delivery capacities. Conversely, for the same piston speeds the subject design will offer very much higher delivery capacity in a smaller and lighter unit.

The smooth flow characteristics are established by the well known fact that the 120 degree cylinder or crank diagram shows at all times perfect distribution of free forces and components for various crank positions. With double the number of cylinders (60° diagram) in each row, it is quite certain that the subject design will have a perfectly smooth delivery flow without any pulsation whatever.

The versatility of the invention as embodied in a basic design, such as here illustrated, is quite evident. As already pointed out, adaptation for use as a compressor is easily and simply accomplished by means of convex plunger heads, whereas for use as a liquid pump, as here shown, the plunger heads are dished in to provide for adequate head space for smooth and orderly flow of the liquid into and out of the cylinder by way of the ports 36 and 37. As here shown, the passage 38 serves the discharge while 39 and 40 supply the intake; however, these roles may readily be reversed by merely reversing the direction of rotation, when the nature of a particular application so requires. Likewise, by interconnecting these passages in a suitable manner, the pump may be

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built up in multiple stages, in series or in parallel, as desired—the various possible applications being too numerous to here describe.

It will be appreciated from the foregoing that the essence of this invention resides in the use of rocking piston cylinders which open and close the inlet and outlet ports. Applicant utilizes rigid piston and piston rod members, said pistons and rods being either integral with each other or rigidly fixed to each other so that they are adapted to move as a single integral unit. The rocking motion of the piston cylinders is caused by the engagement of these rigid piston and piston rod assemblies with a crank. The longitudinal axis of the piston cylinders and of the piston and piston rod assemblies is at all times perpendicular to the axis of the crankpin. The axis on which the piston cylinders rock or pivot, intersects the longitudinal axis of the cylinder at right angles, and it is at all times parallel to the axis of the crankpin and crankshaft. It is accordingly possible to provide an in-line pump or compressor incorporating the principles of the present invention, although only a radial design is shown in the drawing.

I claim:

A device of the character described comprising a frame, disc shaped members rotatably mounted in said frame having an angular movement about their longitudinal axis, said discs having a diametrically extended hole formed therein and being provided about their periphery with ball bearings partially inbedded therein, inlet and outlet ports formed in said frames adjacent each of said discs and located to alternately communicate with the hole in said disc when the disc is pivoted back and forth upon its said axis, a crank shaft in said frame, a rigid piston and piston rod assembly connected to said crank shaft by means of flanges at the ends of said piston rods and ring shaped bushings on the crank shaft engaging said flanges and preventing movement of said piston rod away from said crank shaft, said piston being mounted in said diametrically extending hole, whereby a reciprocating and pivotal movement of the piston and piston rod assembly is caused when the crank is worked and said pivotal movement is transmitted to said disc thereby causing said disc to alternately open and close said port holes.

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