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(54) Single acting piston pump.

(57) The pump (18) has a barrel (36) including a bore (38), an inlet (42) and an outlet (44). Check valves (70,80) control the flow of fluid from the inlet to the outlet during reciprocation of a piston (50) within the bore (38). The piston (50) is driven by a cam (30) having a special profile cam surface with an idle surface (34) of at least 100 degrees. The flow area of the first check valve (70) is greater than that of the second check valve (80).

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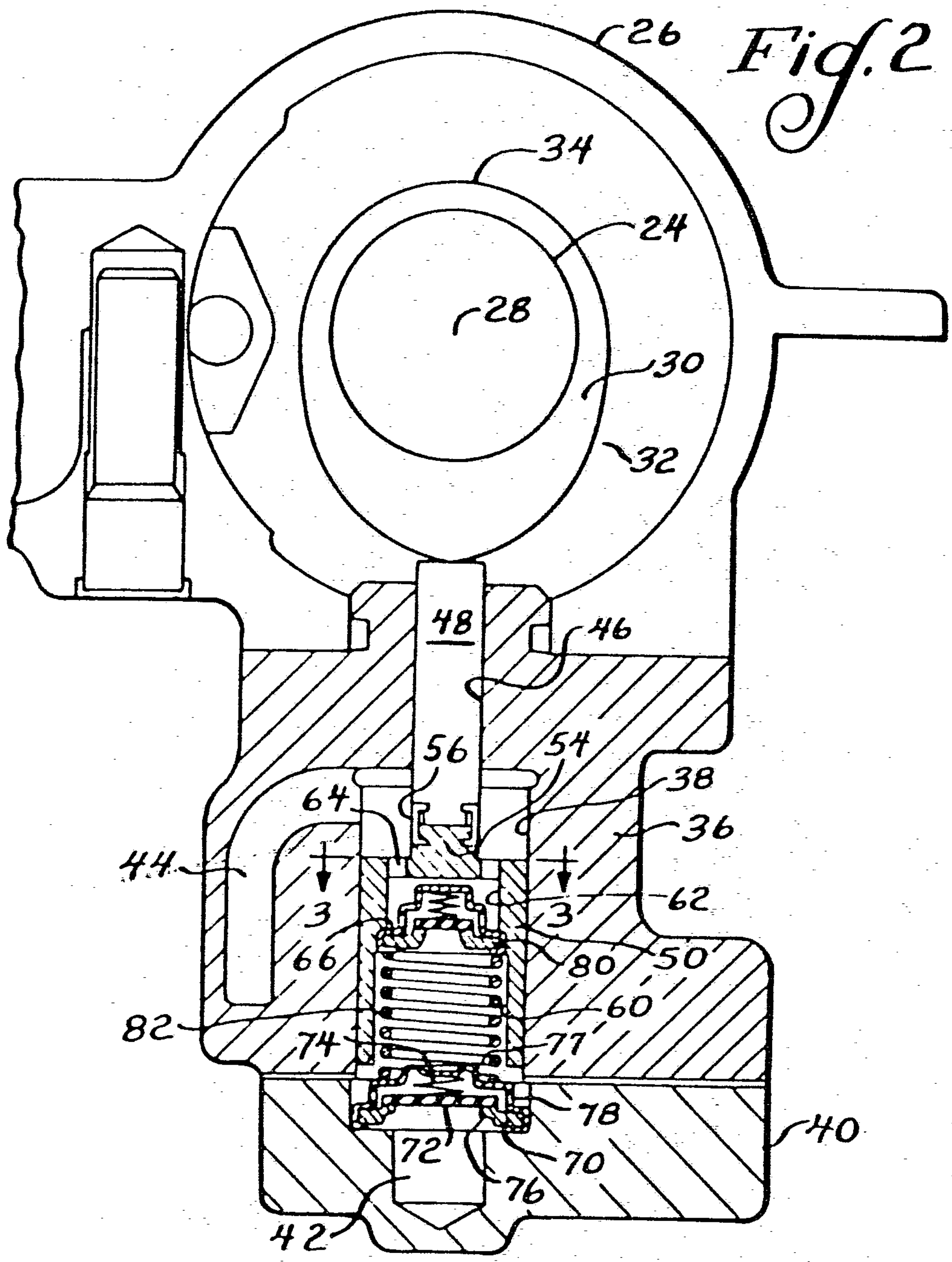

26
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Fig. 2

CATERPILLAR TRACTOR CO. GJE 5180/125

SINGLE ACTING PISTON PUMP

This invention relates to single acting piston pumps which may be used, for example, as transfer pumps in fuel injection systems for internal combustion engines.

Many fuel injected internal combustion engines in use today employ transfer pumps for providing fuel, at relatively low pressures, to a plurality of injection pumps, one for each working chamber of the engine. Frequently, the transfer pumps employed are piston pumps driven by, for example, a cam shaft. In most instances, it would be desirable to utilize a single acting pump by reason of their relative simplicity and low cost. However, the volumetric efficiency of single acting pumps tends to drop off rapidly as engine speed increases due to fuel line inertia losses and/or cavitation losses with the consequence that at higher engine speeds, such pumps may be unable to supply the required fuel volume to the injector pumps to maintain the desired fuel pressure. Where such pumps are capable of delivering the desired fuel quantity, they are therefore necessarily oversized, and more costly, than a pump whose volumetric efficiency is greater. Moreover, at low engine speeds, much of the capacity of such oversized transfer pumps is not utilized.

As a consequence, many such engine systems utilize double acting piston pumps which have a consistently higher volumetric efficiency than conventional single acting piston pumps for substantially all ranges of engine speeds. However, double acting piston pumps are not without their drawbacks. By their very nature, they are more complicated than single acting pumps and therefore more expensive, and because of their greater complexity, they may be more prone to failure, thereby increasing maintenance costs.

According to a first aspect of the invention, a single acting piston pump has a pump barrel having a bore, an inlet on one end and an outlet at the other end; a piston reciprocally received in the bore and defining a flow passage therethrough and a shoulder therewithin; means for reciprocating the piston within the bore and including a rotatable cam having an idle surface of at least 100 degrees; a first check valve in the barrel inlet for allowing liquid flow into the bore and preventing reverse flow; a second check valve connected to the piston within the flow passage for allowing liquid flow from the bore to the outlet and preventing reverse flow; and spring means for urging the second check valve against the shoulder and towards the cam.

According to a second aspect of the invention, a single acting piston pump comprising a pump barrel having a bore, an inlet at one end and an outlet at the other end; has a piston reciprocally received in the bore; means for reciprocating the piston within the bore; a first check valve in the barrel inlet for allowing liquid flow into the bore and preventing reverse flow; a second check valve connected to the piston and arranged to allow fluid flow from the bore to the outlet; the first check valve having a considerably greater flow passage area than the second check valve.

The technical problem solved by the present invention is the low volumetric efficiency of single acting piston pumps, particularly when operated at high speeds. The invention solves the problem by providing a single acting piston pump having a volumetric efficiency of the order of conventional double acting pistcn pumps while yet maintaining the simplicity and low cost associated with single acting piston pumps.

One example of a pump constructed in accordance with the invention will now be described with reference to the accompanying drawings, in which:-

Figure 1 is a schematic diagram of an internal combustion engine and fuel system with which the invention may be employed;

Figure 2 is a sectional view of an embodiment of a single acting piston pump constructed according to the invention; and,

Figure 3 is an enlarged, fragmentary sectional view taken along the line 3-3 in Figure 2.

A single acting piston pump is illustrated in the drawings in the environment of a power plant including an internal combustion engine 10 having four working chambers (See Figure 1). A plurality of fuel injection pumps 12 are provided, one for each working chamber. The pumps 12 may be of the conventional scroll type and form no part of the present invention.

Fuel is contained in a fuel tank 14 having an outlet connected by a conduit 16 to the inlet of a transfer pump 18 construction according to the invention. The outlet of the transfer pump is connected by a conduit 20 to the injection pumps 12 to supply fuel thereto. Excess fuel at the injection pumps 12 may be returned to the tank 14 or the line 16 via a conduit 22 shown in dotted lines.

The engine 10 will typically drive one or more cam shafts or the like for providing power to the injection pumps 12 and the transfer pump 18. Such a cam shaft is shown schematically at 24 in dotted lines in Figure 1.

Turning now to Figure 2, the construction of the transfer pump 18 will be described in greater detail.

The cam shaft 24 is contained within a housing 26 and is rotatable about its longitudinal axis 28. Mounted on the cam shaft 24 for rotation therewith is a cam 30 which has a special profile. The cam 30 has a profile including a pumping surface 32 and a nonpumping idle surface 24. The profile is referably symmetrical on both sides of its apex so as to provide pumping for either direction of cam shaft revolution, but where this is not of concern, symmetry may be dispensed with.

The nonpumping idle surface 34 coincides with the base circle of the cam 30, that is, it is of constant radius about the rotational axis of the cam shaft 24. The surface 34 has as long an angular length as possible, at least 100°, and in the preferred embodiment, 150°. Consequently, the idle surface 34 provides for a relatively long dwell for purposes to be seen.

A pump barrel 36 is secured in any suitable fashion to the cam shaft housing 26 and includes an elongate bore 38 located on a radius extended of the cam shaft 24. The barrel 36 includes a head 40 closing one end of the bore 38 and which includes an inlet passage 42 which is connected to the conduit 16 (Figure 1) from the fuel tank 14.

At the end of the bore 38 opposite from the inlet 42 the barrel 36 includes an outlet 44 which is connected via the conduit 20 to the injection pumps 12.

A relatively small diameter bore 46 extends from the outlet end 44 of the bore 38 to the end of the barrel 36 to emerge within the cam shaft housing 26. A cylindrical cam follower 48 is received within the bore 46 and is of such a length so as to extend into the bore 38.

A piston 50 is received within the bore 38 for reciprocation therein and, at one end, includes a stub shaft 54 in abutment with the end of the cam follower 48 remote from the cam 30. A circular clip 56 having the configuration illustrated is received in peripheral grooves in the stub shaft 54 and the cam follower 48 to unite the two. Extending longitudinally through the piston 50 is a liquid passageway defined by a first, large bore 60 and a second, slightly smaller bore 62 in axial alignment with each other. As seen in Figs. 2 and 3, a series of small bores 64 extend from the bore 62 about the stub shaft 54 and cam follower 58 to open to the outlet end of the bore 38.

The difference in diameter between the bores 60 and 62 defines a small shoulder 66 within the fluid passageway for purposes to be seen.

A first check valve 70 is disposed within the bore 38 at the inlet end thereof and includes a valve disc 72 normally urged by a spring 74 against a valve seat 76. A stamped metal web 77 restrains the spring 74 and the web 77 is provided with a series of openings 78. As can be appreciated, the check valve 70 is operative to allow liquid to flow from the inlet 42 into the bore 36 but precludes reverse flow.

A structurally similar, but smaller check valve 80 is disposed within the liquid passage within the piston 50 in abutment with the shoulder 66. The check valve 80 will not be described in detail in the interest of brevity but it should be noted that the same is operative to allow liquid to flow from the inlet end of the bore 36 to the outlet end 44 and preclude reverse flow, fluid communication being provided by the liquid passage defined by the bores 60,62 and 64 in the piston 50. It should be also noted that the flow area of the check valve 70 is considerably greater than that of the check valve 80 and in a preferred embodiment, will be on the order of at least twice as large.

It should be further noted that the minimum flow area downstream of the valve 80 be equal to or greater than the minimum flow area through the valve 80. Thus the sum of the cross-sectional areas of the bores 64 should equal or exceed the smallest flow area through the valve 80.

A compression spring 82 is located within the bore 60 in the piston 50 and urges the check valve 80 against the shoulder 66 in substantially sealed relation thereto. The spring 82 also biases the check valve 70 toward the inlet 42. Finally, the spring 82 provides a return biasing force to the piston 50 via the check valve 80 to urge the same upwardly within the bore 38 as seen in Fig. 2 so that the cam follower 48 will ride against the cam surface 32 of the cam 30.

Operation is as follows. Because the cam surface 32 is a lobed symmetrical surface, the cam shaft 24 may be rotated in either direction and yet effect operation. In the configuration illustrated in Fig. 2 the cam follower 48 is at its maximum lift position by reason of the angular position of the cam 30. As the cam rotates from the position illustrated in Fig. 2, the spring 82 will urge the piston 50 upwardly within the bore 38. At this time, the check valve 80 will be closed so liquid within the outlet end of the bore 38 will be directed out of the outlet 44 as the piston 50 ascends.

At the same time, the volume within the bore 38 and below the check valve 80 will increase so that liquid may flow into the lower end of the bore 38 through the check valve 70 from the inlet 42.

When the cam has rotated 255° from the position shown, it will begin to drive the follower 48 downwardly to drive the piston downwardly with the bore 38.

At this time the check valve 70 will close and the liquid within the lower end of the bore 38 will cause the check valve 80 to open so that the liquid may flow through the fluid passages 60, 62, 64 within the piston 50 to the outlet end of the bore 38. Such will occur until the cam 30 has returned to the position illustrated in Fig. 2.

The construction maximizes volumetric efficiency as follows. The cam profile described drives the piston 50 to maximum displacement and permits it to return to minimize displacement as rapidly as the diameter of the cam follower 48, cam stress level, and spring force will allow. The rapid fall of the pumping surface 32 generates maximum vacuum in the fill chamber defined by the lower end of the bore 38, and thus in the inlet 42, to draw in fuel for the subsequent stroke. The following period of dwell provided by the surface 34 allows even more time for fluid flow to the bore 38 from the inlet. In other words, pumping occurs during the filling cycle thereby increasing the time available for filling. This in turn results in a decrease in the fuel velocity at both the inlet 42 and the outlet 44. Lower fuel velocity, in turn, minimizes the effect of inertia on the moving column of fuel in the lines due to stopping and starting. At the same time, the long period of time in every pumping cycle during which filling occurs reduces cavitation problems. The minimization of both inertia and cavitation produce a considerable increase in volumetric efficiency. On the average, this increase may be some 60 percent over that obtainable in conventional single acting piston pumps.

Volumetric efficiency is also enhanced by the fact that the flow path through the check valve 70 is considerably greater than that of the check valve 80. The large size of the inlet flow path further enhances filling at low fuel velocities, thereby further reducing inertia losses. The use of a considerably larger valve at the inlet reduces also resistance to the filling operation at the valve thereby further eliminating the deleterious effects of cavitation.

Similarly, the provision of the flow passage through the piston itself tends to minimize the length of the flow path from the inlet to the outlet thereby reducing losses and further provides for a simple construction that can be economically fabricated.

A pump made according to the invention is desirably employed in an engine system such as shown in Fig. 1 but may be utilized in other applications where the simplicity and inexpensive construction of a single acting piston pump is desired and where high volumetric efficiency commensurate with that achieved by double acting piston pumps is required.

CLAIMS.

1. A single acting piston pump comprising a pump barrel (36) having a bore (38), an inlet (42) at one end and an outlet (44) at the other end; a piston (50) reciprocally received in the bore and defining a flow passage (60, 62, 64) therethrough and a shoulder (66) therewithin; means (30, 48) for reciprocating the piston (50) within the bore and including a rotatable cam (30) having an idle surface (34) of at least 100 degrees; a first check valve (70) in the barrel inlet (42) for allowing liquid flow into the bore (38) and preventing reverse flow; a second check valve (80) connected to the piston (50) within the flow passage(60,62,64) for allowing liquid flow from the bore (38) to the outlet (44) and preventing reverse flow; and spring means (82) for urging the second check valve (80) against the shoulder (66) and towards the cam (30).

2. A pump according to claim 1 wherein the first check valve (70) has a flow passage area considerably larger than that of the second check valve (80).

3. A pump according to claim 1 or claim 2, wherein the flow passage (60,62,64) has an area at least equal to that of said second check valve.

4. A pump according to any of claims 1 to 3, wherein the means (30,48) for reciprocating the piston (50) within the bore (38) includes a cam follower (48) centrally secured to the piston (50).

5. A pump according to claim 4, wherein the flow passage (64) directs flow to the outlet (44) around the cam follower.

6. A pump according to any of claims 1 to 5, wherein the second check valve (80) is carried by the piston (50).

7. A single acting piston pump comprising a pump barrel (36) having a bore (38), an inlet (42) at one end and an outlet (44) at the other end; a piston (50) reciprocally received in the bore (38); means (38,40) for reciprocating the piston (50) within the bores (38); a first check valve (70) in the barrel inlet for allowing liquid flow into the bore and preventing reverse flow; a second check valve (80) connected to the piston (50) and arranged to allow fluid flow from the bore (38) to the outlet (44), the first check valve having a considerably greater flow passage area than the second check valve (80).

8. A pump according to claim 7 wherein the reciprocating means (30, 48) includes a rotatable cam shaft (24) and cam (30) mounted on the shaft and having an idle surface (34) and a pumping surface (32), having two sides, the idle surface (24) being of greater angular length than e ither side of the pumping surface (32).

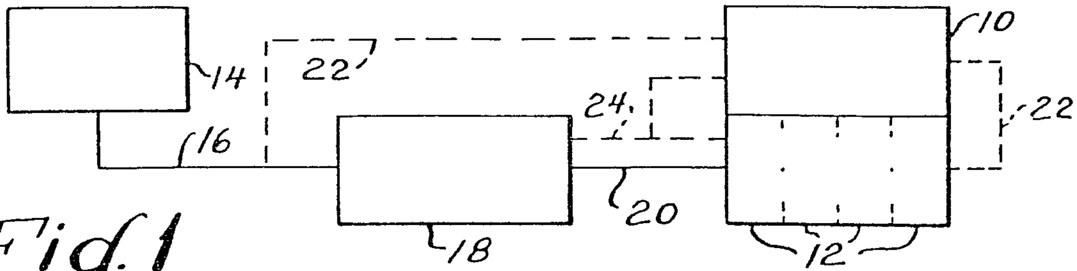
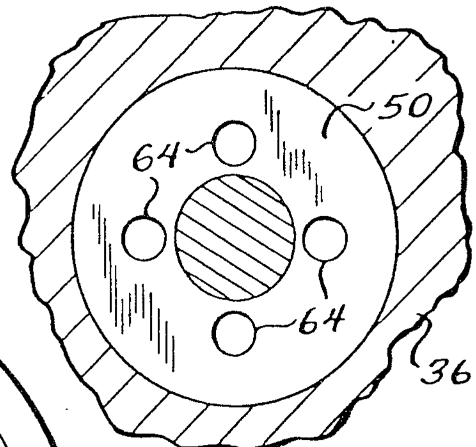


Fig. 1

Fig 3



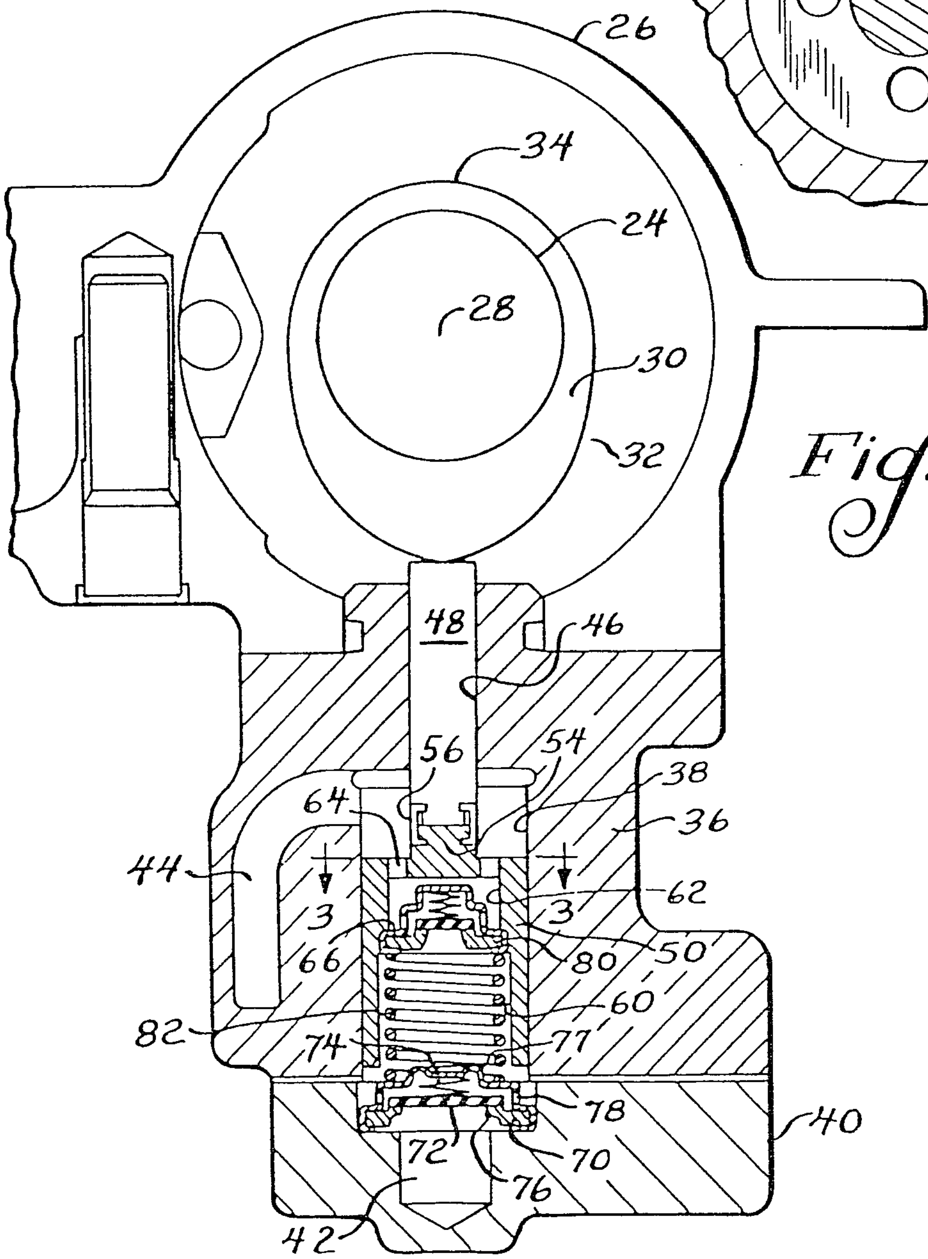


Fig. 2

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EUROPEAN SEARCH REPORT

Application number
EP 80 30 3643

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.3)
	US - A - 3 338 172 (HÄFELE) * Column 2, lines 5-72; column 3, lines 1-2; figure 1 * ---	1,3,4,5,7,8	F 02 M 37/06 F 04 B 21/04 F 04 B 21/02 F 04 B 9/04
	US - A - 3 277 632 (HALBERG) * Column 2, lines 20-50; column 3, lines 2-17; figures 1 and 3* ---	1,2,6,7	
	US - A - 1 783 940 (TRUMBLE) * Page 1, lines 87-98; figures 1 and 2 * ---	1,4,8	TECHNICAL FIELDS SEARCHED (Int. Cl.3)
	US - A - 2 183 510 (THOMAS) * Page 1, left-hand column, lines 28-55; right-hand column, lines 1-21; figure 3 * ---	1,4,5,8	F 02 M F 04 B
	US - A - 1 830 257 (BOHNENBLUST) * Page 1, lines 40-59; figure * ---	1,2,3,6,7	
	GB - A - 1 202 877 (EXPANDITE) * Page 2, lines 59-80; figure 2 * -----	1,2,3,6,7	

CATEGORY OF CITED DOCUMENTS

X: particularly relevant
A: technological background
O: non-written disclosure
P: intermediate document
T: theory or principle underlying the invention
E: conflicting application
D: document cited in the application
L: citation for other reasons

&: member of the same patent family, corresponding document

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
The Hague	22-01-1981	HEINLEIN