

July 11, 1944.

M. CASERTA

2,353,545

FLUID TRANSFER MECHANISM

Filed June 12, 1941

2 Sheets-Sheet 1

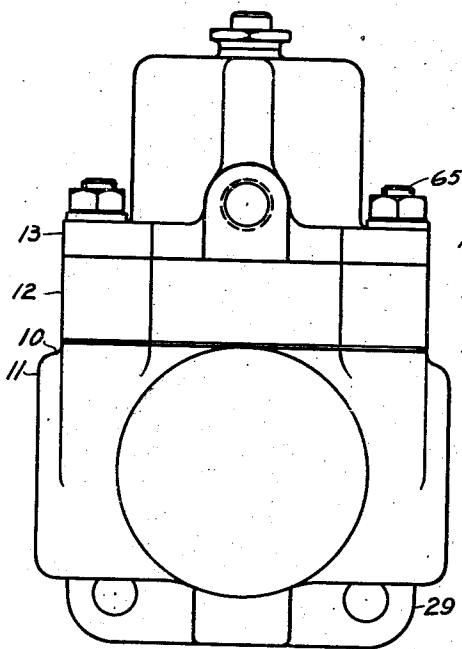


FIG. 1

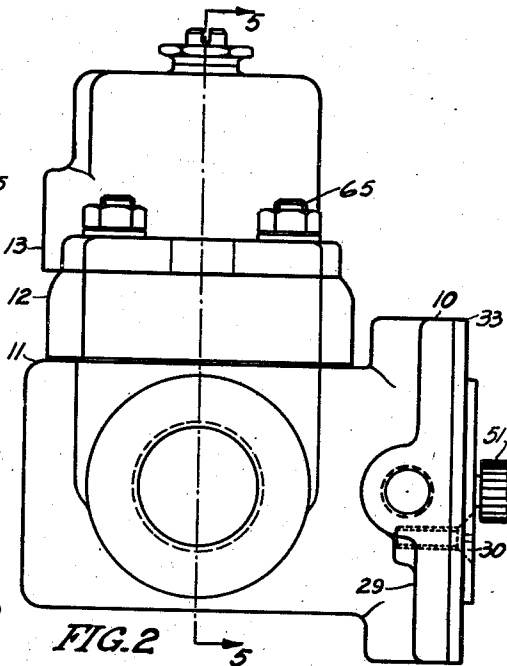


FIG. 2

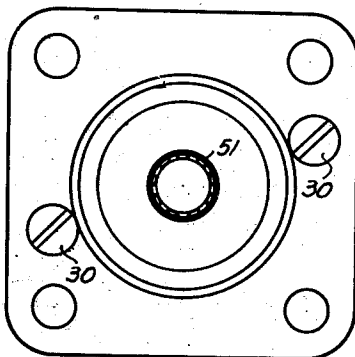


FIG. 4

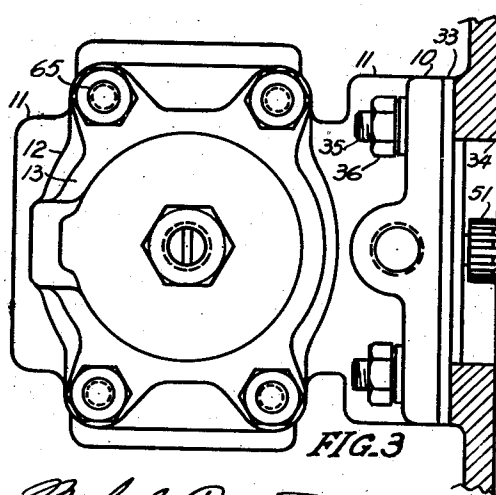


FIG. 3

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2 Sheets-Sheet 2

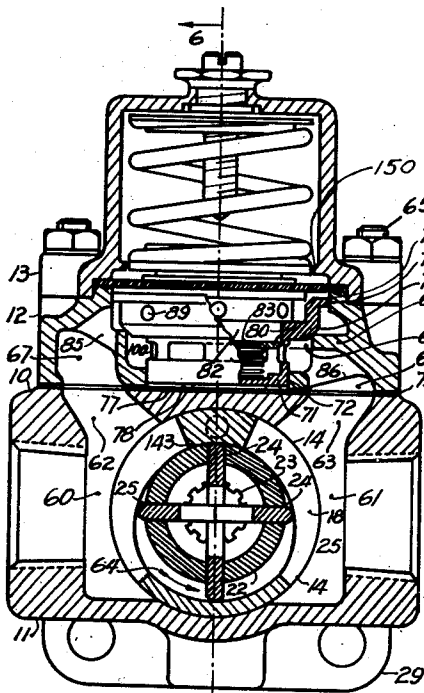


FIG. 5

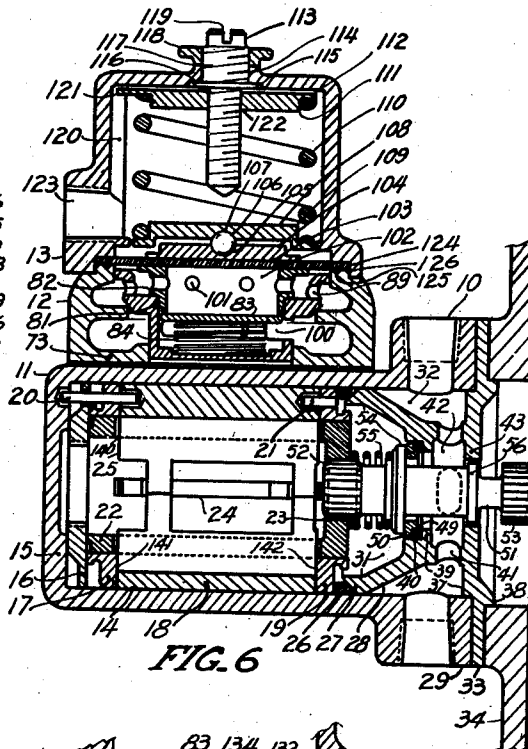


FIG. 6

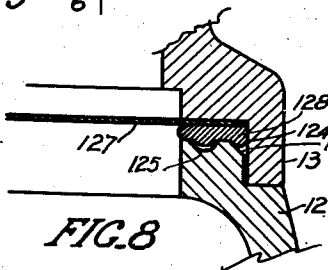


FIG. 8

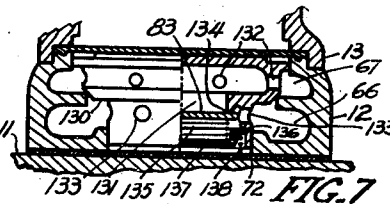


FIG. 7

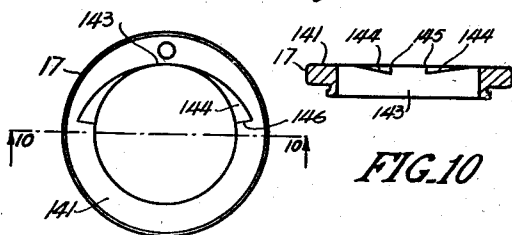


FIG. 9

FIG. 10

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2,353,545

FLUID TRANSFER MECHANISM

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Application June 12, 1941, Serial No. 397,763

17 Claims. (Cl. 277-45)

This invention relates to fluid transfer mechanism and more particularly, although not so restricted, to the type used in aircraft fuel transfer installations.

Usually in such installations a pump, driven for instance by the engine, transfers the gasoline or other fuel from the reservoir to the carburetor or other distributing device; one of the objects of the present invention is to keep a constant fuel pressure at the carburetor even if variations take place in the speed at which the pump is driven, or in the pressure of the fuel at the pump inlet side, or in the atmospheric pressure, or in the fuel flow, or in any other factor tending to affect the fuel pressure.

Another object is to provide means by which fluid may be bypassed from the inlet side to the outlet side with a minimum pressure drop, as is desirable for instance in the event that the pump should be out of order and an emergency pump be used.

A further object is to provide control means which are free in their movements without tendency to stick or bind, so as to respond to the slightest variation in pressure and as a consequence control said pressure within very close limits.

Still a further object of the present invention is to provide a mechanism which is simple, compact, light, easily manufactured and assembled, and generally efficient.

These and other objects will be more apparent from the following description and from the accompanying drawings.

Figures 1, 2, and 3 are an end view, a side elevational view, and a plan view respectively of a fuel pump embodying the present invention, Figure 3 showing also semi-diagrammatically partly in section a fragment of a mounting flange to which said pump may be attached, and some of the mounting bolts.

Figure 4 is an elevational view of the flange end only of said pump.

Figure 5 is a cross sectional view taken substantially on line 5-5 of Figure 2, some of the parts being left not-sectioned for clearness of illustration.

Figure 6 is a similar cross sectional view taken substantially on line 6-6 of Figure 5.

Figure 7 is a semi-diagrammatical fragmentary view showing partly in section and partly in elevation an alternate construction of the valve.

Figure 8 is a semi-diagrammatical fragmentary sectional view of an alternative diaphragm mounting structure.

Figure 9 is an elevational view of a modified form of bearing, and

Figure 10 is a sectional view of said bearing taken on line 10 of Figure 9.

In the figures the fuel pump assembly is indicated by the numeral 10 and includes a pump housing 11, a valve housing 12, and a cover 13.

Pump housing 11 is provided with a bore 14, terminated by an end wall or thrust face 15. In said bore 14 are lodged in line and preferably press fitted a thrust bearing 16, a first journal bearing 17, an eccentric pump chamber or liner 18, and a second journal bearing 19. A pin 20 prevents the rotation of parts 16 to 19 relative to each other and relative to housing 11, while pin 21 prevents the relative rotation of chamber 18 and bearing 19.

A straight cylindrical rotor 22 is supported in bearings 17 and 19, and is provided at one end with a splined hole 23 through which it may be driven, and with four radial slots 24, through which the vanes 25 slide.

A ring or pump gasket 26, preferably made of resilient or deformable material such as for instance synthetic rubber, is pressed against bearing 19 by the bevel end 27 of gland 28 which is assembled to the flange 29 of pump housing 11 by means of screws 30 (Figures 2 and 4). The parts are preferably so proportioned that when the screws 30 are tightened, bearings 16 and 17, liner 18 and bearing 19 are all pressed against the end wall 15 and seal ring 26 is compressed and preferably deformed to produce a fluid tight seal between bore 14 and gland 28, and so prevent the passage of fluid from the pump side 31 of gland 28 to the other or outer side 32 along said bore; excessive squeezing of the pump seal ring 26 is prevented by gland flange 33 which comes in contact with flange 29.

When the whole pump assembly is assembled to the mounting flange 34 by means of the usual bolts 35 and nuts 36, the heads of screws 30 are covered by said flange 34 and the additional powerful action of nuts 36 holds together flanges 33 and 29. It is then clearly apparent that by my structure I avoid the use of large threaded bodies such as used in some structures in which a large gland is threaded into a housing, and not only I avoid the danger of seizure of such large threads particularly if the parts are made of aluminum, but avoid the expensive tools and operations necessary for such threads, and at assembly completely exclude the human element as a factor in the tightness and compression of the pump seal. Furthermore I don't need to make screws 30 very accurate or strong, nor do I need to lock them, as they are relied upon only to hold the parts together during storage and transit, while when the pump is in operation all the parts are securely held by the usually powerful studs 35 and nuts 36. On the other hand to disassemble my pump a screw driver is the only tool needed to loosen the screws 30, after which the gland, shaft, and related parts

slip out of the pump housing and fall apart one from the other.

Gland 28 is provided with a land 37 which holds it in central position relative to bore 14, and with a concentric land 38 which is useful in mounting the pump assembly centrally on for instance the mounting flange 34 usually provided for it on the engine.

In its central portion gland 28 is also provided with an inner flange 39 and recessed seat 40, with four radially spaced openings 41 preferably cored out in casting so as to leave four pillars 42, and with a second inner flange 43. A ring or gasket 49 of soft or resilient material such for instance as synthetic rubber, is lodged in recessed seat 40 and supports centrally and squarely with the bore a seal seat 50 preferably made of wear resistant material such as bronze.

Shaft 51 is provided with two splined ends, one, 52, for driving the rotor 22, and the other, 53, for connection to the pump driving means; the shaft is also provided with a flange 54 which is pressed against seal seat 50 by means of spring 55, and with a second flange 56 substantially in line with flange 43 with which it forms a wall of protection of the inside of the pump from foreign matter. It is clear that the rubber gasket 49 is helpful in taking care of misalignment between shaft and gland or mounting flange, and by its resiliency allows the seal seat 50 to follow flange 54 in its possibly wobbly movement, and so effectively prevents leaks past said seat.

Spring 55 at the other end presses against rotor 22, which in turn rests against thrust bearing 16.

Pump housing 11 is provided with two cavities or openings 60 and 61 symmetrically opposite with reference to the rotor, and with two ports 62 and 63 connected thereto respectively. The pump will operate in either direction but for the purpose of simplicity of description we will assume that it be operating in anticlockwise direction, as shown by arrows 64 in Figure 5; then cavity 60 and port 62 will be on the inlet or low pressure side, and cavity 61 and port 63 will be on the outlet or high pressure side.

Valve housing 12 and valve cover 13 are fastened to pump housing 11 by means of screws 65, which are preferably located in plan view according to a pattern symmetrical with reference to the pump axis, so that the valve housing can be inverted or rotated 180° when the pump is operated in clockwise direction.

Valve housing 12 is provided with an outlet or pressure side opening or chamber 66 which communicates with the outlet port 63 of the pump, and which for simplicity of description will also be called lower chamber; and with an inlet or suction side opening or chamber 67 which communicates with the inlet port 62 and which will also be called upper chamber, with the understanding, however, that the expressions "upper" and "lower" do not in any manner define or limit the relative location of said chambers to each other or to the pump housing.

Said chambers are separated by a partition or wall 68 which is provided with a port or valve seat 69, which may be closed by relief valve 70. The lower part of relief valve 70 terminates in a cylindrical portion 71 which is guided in bore 72 provided in the lower wall 73 of valve housing 12, while the upper part 75 is also cylindrical and is guided in bore 76.

The upper surface 77 of pump housing 11, bore 72, and valve lower wall 71 confine a space

or dash-pot cavity 78. A gasket 79 is preferably provided between pump housing 11 and valve housing 12.

Relief valve 7 itself is divided into two sections by a horizontal wall 80 which provides an opening or seat 81 in which a by-pass valve seat 82 is lodged. A by-pass disc or valve 83 is pressed against said valve seat 82 by spring 84 and completes the horizontal wall of elements 68, 80, 82, and 83 which close communication between upper and lower chambers.

Numerals 85 and 86 indicate ribs cast in valve housing 12 to stiffen the structure and promote smoother flow of the fluid therethrough.

The relief valve 70 is provided with upper lateral ports 89 and lower lateral ports 100, and by-pass valve seat 82 is provided with lateral ports 101 and axial port 102.

A diaphragm 103 is held in fluid tight manner at its outer edge between valve housing 12 and cover 13, and rests in its central portion against by-pass valve seat 82. On diaphragm 103 we find a washer 104 provided with a seat 105 for ball 106, which in turn is held at its upper side in seat 107 provided in spring washer 108. Said washer 108 carries on its locating seat 109 the adjusting spring 110 which at its upper end rests on seat 111 of adjusting washer 112 into which the adjusting screw 113 is threaded.

It will be noted in Figures 5 and 6 that washer 108 is preferably of larger diameter than washer 104, so that if the assembly of washer 108 and washer 104 with the centering ball between them were accidentally displaced laterally, washer 108 would touch the inside of cover 13 but washer 104 would still remain free. The same figures also show a rib 150 with which cover 13 is preferably provided in correspondence with washer 108 to prevent excessive accidental lateral displacement of said washer at assembly or in operation.

Valve cover 13 is formed with a bore 114 and a counterbore or seat 115. Adjusting screw 113 is provided with a flange 116 which rests in counterbore 115, and with an upper threaded stem 117 which fits into bore 114 and may be clamped tight to cover 13 by means of check nut 118, or may be rotated for instance by means of screw driver slot 119.

Valve cover 13 is provided on the inside with a vertical groove 120 in which a finger 121 of washer 112 can slide up and down while at the same time preventing said washer 112 from rotating. When adjusting screw 113 is turned, said finger prevents washer 112 from rotating and the consequent action of threaded hole 122 in the center of washer 112 compels said washer to move up and down and so adjust the spring tension as desired.

An important advantage of this construction is that flange 116 of adjusting screw 113 is pressed against its seat 115 at all times by the tension of spring 110 and therefore is prevented from accidental rotation by the friction and has no tendency to lose its adjustment even if check nut 118 is left loose; furthermore said flange pressed on its seat provides a fluid tight closure, the effectiveness of which is only increased by the application of supercharger pressure above the diaphragm for instance through port 123.

It will be seen that as rotor 22 and vanes 24 rotate in the direction indicated by arrow 64, the fluid is carried from inlet or low pressure port 60 to outlet or high pressure port 61, and if the

free flow thereafter is impeded, pressure is built on the outlet side.

When the pressure attains a predetermined value, relief valve 70 is lifted from its seat 69 against the antagonistic tension of spring 110, and the fluid flows between seat 69 and valve 70 and through port 67 back to the inlet side and to openings 62 and 60.

Should there be any tendency for the fluid pressure to pulsate, and therefore for valve 70 to vibrate or bounce on and off its seat, it will be seen that the lower part 71 of said valve will act as a piston in cylinder 72 and trap fluid in chamber 73, so that a dash-pot action will be obtained and the vibrations of valve 70 will be minimized.

Should it be desirable to pump fluid from port 60 to port 61 by means independent from the pump proper, for instance when the pump is out of order or when the engine is not turning, the fluid from inlet ports 60 and 62 will pass through ports 67, 89, 101, 102, will lower valve disc 83 against the tension of spring 84, and through ports 100, 66, and 63 will reach outlet port 61. It is clear that disc 83 may be made very light and thin, and of light material such as Bakelite; said disc 83 is normally pressed against its seat 82 by the fluid pressure on the outlet side, so that spring 84 may be very light, and the drop of pressure of fluid by-passed through the valve will be very small.

It is a known fact that it is very difficult to produce springs whose ends will stay naturally parallel when free and when under tension; usually, as the spring is compressed its ends tend to twist out of parallel and form an angle, and the direction of the spring action does not remain axial. In other words it may be said that if one end is held stationary and the other one is compressed, said other one will tend to rotate as it moves axially, and the spring will give a torque component besides the expected axial reaction component.

This is an important factor in bringing about stickiness in valves that are spring pressed, as the torque component will twist the valve sideways against its guiding means, and the valve will stick and bind. The present invention provides means whereby any spring tendency to twist spends itself freely and no binding action is transmitted to the valve.

In fact, if the lower face of spring 110 had any tendency to twist or rotate it can freely do so, as it rests on washer 108 which can freely rotate around ball 106, which in turn transmits only an axial component to washer 104 which, through diaphragm 103 and valve 82 presses on valve 70.

It will be noted that the adjusting screw 113 is held or guided on the vertical axis of the pump and valve assembly by bore 114 and counterbore 115, and in turn hold and guides on said axis the adjusting washer 112 which in turn through shoulder 111 guides spring 110; the lower end of spring 110 rests around shoulder 109 and centers washer 108, while ball 106 resting in seats 107 and 105 centers washer 108 and 104, so that washer 104 is finally held on the axis of the valve assembly without having any lateral guides which might introduce friction and binding action. In conclusion, there is nothing in the valve assembly constructed in accordance with the present invention that can produce binding action, and the result, confirmed in practice, is that said valve is extremely free in

its movement and gives very sensitive pressure control.

While the spring and washers are held in line as just described, it is preferable to have washer 108 not much smaller than the inside of cover 13 or otherwise loosely guided in cover 13 to avoid that by error or because of excessive curvature of the axis of the spring said washer 108 at assembly be mounted too far off the axis of the valve, or that it be jarred off said axis by accident during operation. The rib 150, although not essential, facilitates the centering of washer 108.

The structure described will give best results if the spring 110 is of rather large diameter in proportion to its length, so it will rest squarely at both ends and keeps the parts in line at assembly.

The valve seat, the diaphragm, and the related parts are preferably so proportioned that the effective area of the diaphragm equals the effective area of the relief valve, so that variations in the fuel pressure at the inlet side will have no influence on the pressure regulation; furthermore, if pressure is applied to the upper side of the diaphragm for instance by means of the engine supercharger and through port 123, the result will be the same as if said pressure were applied directly on the top of the relief valve, namely the outlet pressure adjustment will be increased in direct proportion to the supercharger pressure applied.

It will be noted that diaphragm 103 is preferably made of resilient and easily compressible material such for instance as synthetic rubber; the shoulder, groove, or free space 124 left between valve housing 12 and cover 13 is preferably made slightly narrower than the diaphragm thickness, so that when the whole assembly is tightened together by means of screws 65 the diaphragm remains clamped all around at its periphery in a fluid tight manner, and fills at least in part circular groove 125 and chamfer space 126 forming undulations or rings which are beneficial in insuring fluid tightness even in the event that the diaphragm material should take a permanent set.

An alternative construction of the diaphragm is shown in Figure 8 and may be preferable when the diaphragm 127 is made of thin and not very compressible material; accordingly, a ring 128 of easily compressible material is inserted in groove 124 with the diaphragm and, being thicker than the groove when free, supplies the necessary pressure to provide a fluid tight joint when all parts are clamped together and bulges somewhat into grooves 125 and chamfer space 126.

In valves provided with diaphragms as known in the art we usually find bolts that go through the diaphragm and washers and nuts to fasten rods or similar parts to the diaphragm and pick up its action; holes in the diaphragm material, and the squeezing action of the bolts and washers are a continuous source of trouble, as the holes are a good start for diaphragm cracks and the nuts or clamping means are often assembled too loose and produce a leak, or too tight and squeeze and break the diaphragm, which again leaks as a result. Other known structures resort to deep-drawn or bellows shaped diaphragms which are expensive to manufacture and unreliable in service. The present invention allows the use of the simplest form of diaphragm, a plain flat disc, without any central holes or tightening means. Of course such a flat dia-

phragm is usually preferable, but other types of diaphragms or bellows or equivalent pressure responsive means known in the art could be substituted without departing from the spirit of the present invention.

Spring 110 presses washer 104 against the diaphragm and the diaphragm against by-pass valve 82 which rests on relief valve 70; whether said relief valve is resting on its seat 69 or is lifted by the outlet pressure, the spring pressure keeps the diaphragm tightly held between washer 104 and valve 82, and said diaphragm transmits its action upwards against spring 110 or downwards against valve 82 and on to valve 70 as the case may be.

In a similar manner the by-pass valve seat 82 is pressed in a fluid-tight manner against relief valve 70 by spring 110 without the use of further fastening means.

The result of this construction is that when the four screws 65 that hold the valve assembly together are released, all the valves and related parts come apart, and then they may be assembled together again with equal ease and without the use of any tools.

It can be noted that fuel pressures used at present in aircraft installations are mostly in the range of 3-20 lbs. per square inch, and as a consequence if the relief valve area is for instance one and a half square inches, spring 110 shall be adjusted to a tension of approximately 4.5 to 30 pounds. On the other hand, by-pass spring 84 is preferably made as light as possible, and since it is assisted by the outlet pressure in keeping valve disc 83 closed, it is usually made just strong enough to hold disc 83 up to valve 82 against gravity and vibrations; the ratio of tension between spring 110 and spring 84 may then be said to be of the order of 400:1 to 3000:1, and the action of spring 84 in tending to lift valve 82 from valve 70 and in counteracting the tension of spring 110 may be disregarded.

In the modified form shown in Figure 7, the relief valve and the by-pass valve seat are all in one piece 130 which still includes a vertical or axial port 131 which communicates with the upper valve housing chamber 67 through a row of radial ports 132 and with the lower chamber 66 through a row of radial ports 133; vertical port 131 is provided with a by-pass valve seat or opening 134 against which the by-pass valve disc 83 is pressed by spring 135, thus closing communication between the upper and lower rows of radial ports.

Below the lower row of ports 133 is another annular seat 136, against which a washer or disc 137 is pressed by spring 138, which at its lower end rests on pump housing 11. As the tension of spring 138 is preferably made greater than the tension of spring 135, disc 137 will remain pressed against its seat 136, and so effectively close the lower end of valve 130, which then will collaborate with valve housing bore 72 in providing a dash-pot action to minimize any tendency of valve 130 to vibrate or bounce up and down.

No claim is made in the present application to any particular type of bore in pump chamber 18, or to any particular type of vanes operable therein, many being known in the art. It is noted however that in all pumps of the solid vane type the vanes 25 must be of length just slightly shorter than the length of chamber 18, so as to avoid loss of fluid from the high pressure to the low pressure side between the vane ends and the faces of the bearings. The rotor slots

24, on the other hand, must be slightly longer than the vanes and the chamber, to allow free movement of the vanes even if there are slight variations in manufacture and assembly. As a

consequence, and particularly with rotors of the preferred straight cylindrical type as shown in the figures, the vanes are allowed a small amount of end play as indicated in exaggerated scale at numeral 140 in Figure 6. While in most angular positions the vanes are located endwise by the faces 141 and 142 of bearings 17 and 18 respectively, in the sealing zone 143 (Figure 5), where the rotor and chamber are practically in contact, said faces disappear and the vanes are free to move endwise to the end of the slots. Then, by continuing to rotate, the vanes pass the sealing zone and begin to emerge from the rotor and hit the inside surface of the bearing, thereby often spoiling said bearing.

To avoid this action I preferably modify my bearings as indicated in Figures 9 and 10, by providing at each side of the sealing zone 143 a ramp or cam surface recess 144 starting from the sealing zone with depth preferably equal to the slot end clearance 140 and zero width, and then gradually decreasing in depth and increasing in width, until it blends with the face 141 of the bearing.

The start 145 of the cam surface 144 is preferably located within the sealing zone, the width of which may vary according to the particular type of bore adopted, and in some cases may be reduced to one generatrix; the end or blending line 146 of the cam surface with the face of the bearing 141 takes place preferably within that part of the bearing that is always open to the inlet or outlet port and in which no actual pumping occurs; the shape of the cam surface as indicated in Figure 9 is preferably made to follow somewhat the shape of the chamber bore and a little wider to allow some clearance at the vane tips.

It will be apparent that as the rotor rotates, each vane upon reaching the sealing zone is free to move endwise and get inside the bearing; as it emerges from the sealing zone and begins to protrude from the rotor it meets the cam surface 144, which gradually brings it back to its central position in the slot. As the cam recess affects only sections of the bearing which are in communication with either the inlet or the outlet port anyhow, no loss of efficiency of the pump takes place, while less accuracy is needed in the manufacture of the rotor and its slots.

What I claim is:

1. In a valve assembly, a valve housing having coaxial cylindrical bores and an intermediate annular partition providing a valve seat, a valve for said seat having guiding portions at all times in contact with said bores and having an axial passage closed at one end, means providing a by-pass valve seat in said passage, a by-pass valve for said seat and resilient means held under compression between said closed end of said passage and said by-pass valve to hold said valve closed.

2. In a valve assembly, a valve housing having axially spaced cylindrical bores of different diameter and an intermediate annular partition of a diameter intermediate those of said bores, said partition providing a valve seat, a hollow valve seated thereon and having cylindrical portions slidable in said bores to serve as guides for said valve, said valve being closed at its smaller end and open at its larger end, a member in-

sertable through said open larger end to provide a by-pass valve seat, a by-pass valve for said seat, and resilient means between said closed end and said by-pass valve urging said by-pass valve into seating engagement, said member, by-pass valve and resilient means being removable through said larger open end of said valve.

3. In a relief valve assembly, a valve housing open at one end, a cover therefor, a diaphragm peripherally clamped between said housing and cover, a hollow relief valve mounted in said housing, a member positioned in said valve for transmitting pressure from said diaphragm to close said valve and also providing a by-pass valve seat, a by-pass valve for seating thereagainst and a spring urging said by-pass valve against said seat, said valve, member, by-pass valve and spring being capable of assembly by simply dropping the same into place through the open end of said housing in the proper order.

4. In a relief valve assembly, a valve housing open at one end, a cover therefor, a diaphragm peripherally clamped between said housing and cover, a hollow relief valve mounted in said housing and having extensions for slidably guiding the same for movement therein, a member freely seated in said valve for transmitting pressure from said diaphragm to close said valve and also providing a by-pass valve seat, a by-pass valve for seating thereagainst and a spring urging said by-pass valve against said seat, said valve, member, by-pass valve and spring being capable of assembly by simply dropping the same into place through the open end of said housing in the proper order.

5. In a relief valve assembly, an open-ended valve housing having axially spaced coaxial bores and an intermediate annular partition providing a valve seat, a hollow internally shouldered valve resting on said seat when closed and having extensions slidably guided in said bores, an annular member positioned in said valve to rest on the internal shoulder thereof and provide a by-pass valve seat, a by-pass valve therefor, a spring urging said by-pass valve against said seat, and a peripherally held diaphragm closing one end of said valve housing and responsive to pressures exerted thereon to transmit forces through said annular member tending to hold said relief valve against its seat.

6. In a relief valve assembly, an open ended valve housing having coaxial cylindrical bores extending from said ends and an intermediate annular portion providing a relief valve seat, a hollow, ported relief valve having a closed end slidably fitting in one of said bores and an open end slidably fitting in the other of said bores a ported annular member interfitted within said relief valve to provide a by-pass valve seat, a by-pass valve therefor, and a spring confined under compression between the closed end of said relief valve and said by-pass valve to urge the latter into closed position.

7. In a relief valve assembly, an open ended valve housing having coaxial cylindrical bores extending from said ends and an intermediate annular portion providing a relief valve seat, a hollow, ported relief valve having a closed end slidably fitting in one of said bores and an open end slidably fitting in the other of said bores, a ported annular member interfitted within said relief valve to provide a by-pass valve seat, a by-pass valve therefor, a spring confined under compression between the closed end of said relief valve and said by-pass valve to urge the latter

into closed position, and a peripherally held diaphragm closing the other of said bores and responsive to pressures exerted thereon to transmit forces through said annular member tending to hold said relief valve against its seat.

8. In a valve assembly, a hollow valve open at one end and closed at the other and provided with an external seating surface, said ends having cylindrical external guiding surfaces, said valve being further provided with an intermediate internal annular shoulder and the wall of said valve being ported above and below said shoulder, a ported member seated on said shoulder and formed with a dependent cylindrical portion providing an annular by-pass valve seat, a by-pass valve for said seat and a spring positioned in said closed valve end for urging said by-pass valve against its seat.

9. A valve element comprising a cup-shaped member having external cylindrical guiding surfaces adjacent the upper open end and the lower closed end thereof and having an intermediate external seating surface, the wall of said member being ported above and below said seating surface.

10. A relief and by-pass valve assembly, comprising a cup-shaped relief valve having external guiding surfaces and an intermediate external seating surface, the wall of said valve being ported above and below said seating surface and being internally offset adjacent said seating surface, a ported by-pass valve member resting on said offset wall portion and formed with a dependent portion providing an annular by-pass valve seat, a by-pass valve for said seat and a spring normally held under compression between said by-pass valve and the closed end of said cup-shaped relief valve to urge said by-pass valve into seating relationship.

11. A fluid pressure control device including a valve housing comprising a wall separating a high pressure cavity and a low pressure cavity and providing a relief valve seat bore, a wall on the high pressure side providing a dash-pot cylinder bore, a wall on the low pressure side providing a guide bore, said bores being coaxial, a relief valve fitting said bores and provided with an axial port and with lateral ports setting communication between said high pressure cavity and said low pressure cavity, a by-pass valve seat and a by-pass valve arranged in said axial port, said by-pass valve provided with axial and radial ports setting communication between said high pressure cavity and said low pressure cavity, a spring pressing said by-pass valve against said by-pass valve seat, a flexible diaphragm having effective area substantially equal to said relief valve seat, a cover clamping said diaphragm in fluid tight manner on said valve housing on said low pressure cavity, said cover providing a top wall with an axial bore and a side wall with a vertical groove, an adjusting screw provided with a flange and lodged in said axial bore, a washer provided with a radial finger fitting into said groove to prevent rotation of said washer and provided also with a central threaded hole fitting said adjusting screw and with a concentric spring seat, a spring resting at one end on said spring seat, a second washer provided with a spring seat fitting the other end of said spring and provided also with a central ball seat, a ball in said seat, a third washer also provided with a central ball seat and resting against said ball on one side and against said diaphragm on the other side, whereby the tension of said spring can be adjusted by

means of said adjusting screw, said adjusting screw flange is pressed in a fluid tight manner against said cover top wall by the spring tension, said tension is transmitted to said relief valve through said second and third washers and said ball forming a universal joint, and through said diaphragm and by-pass valve seat, and whereby the spring pressure adjustment is uninfluenced by variations of pressure on the low pressure side.

12. A fluid flow control device including a valve housing comprising a wall separating an upper and a lower chamber and provided with a relatively large circular opening providing communication between said chambers, the margin of said opening constituting a valve seat, a bore associated with said upper chamber and a bore associated with said lower chamber, said bores and said opening being coaxial, a valve body having a pair of concentric cylindrical surfaces each adapted for axial reciprocation in one of said bores and including a valve surface adapted to cooperate with said valve seat whereby to close said opening, said valve body being provided with a relatively large axial opening there-through whereby to permit communication between said chambers when said valve and seat are in engagement, a second valve serving to control communication through the axial opening in said valve body and opening in the opposite direction with reference to said first valve, flexible pressure responsive means and spring means operative to press said first valve against said seat, and a spring tending to keep said second valve closed.

13. A fluid flow control device for a pump including a valve housing providing an upper and a lower chamber, the upper chamber being adapted to be connected to the suction side of the pump and the lower chamber being adapted to be connected to the pressure side of said pump, said chambers being separated by a partition provided with an opening forming a communication between said chambers, the margin of said opening constituting a relief valve seat, a relief valve arranged in said opening and including a valve surface adapted to cooperate with said valve seat to close and open said opening in said partition, said relief valve also including an axial by-pass opening therethrough to permit communication between said chambers, the margin of said latter opening constituting a by-pass valve seat, a by-pass valve arranged to close said by-pass opening or to open said by-pass opening, relief spring means tending to close said relief valve, by-pass spring means tending to close said by-pass valve, and a pressure responsive diaphragm having an effective area substantially equal to the area of said relief valve, said diaphragm being placed between said relief valve and said relief spring, whereby said relief valve is substantially uninfluenced by variations of pressure on the suction side of said pump.

14. The structure recited in claim 13 in combination with means in said valve housing providing a cylindrical bore, and piston-like means on

said relief valve fitting said bore whereby to provide a dash-pot action for dampening axial vibrations of said relief valve.

15. The structure recited in claim 13 in combination with means in said valve housing providing a cylindrical bore, an extension of said relief valve fitting said bore, said extension being provided with an axial dash-pot port providing a valve seat, a dash-pot valve member closing said valve seat, and spring means to keep said dash pot valve closed.

16. A fluid control device including a valve housing having a suction side chamber and a pressure side chamber, a partition separating said chambers and provided with a circular opening setting communication between said chambers, the margin of said opening constituting a valve seat, said housing also providing a suction side cylindrical bore and a pressure side cylindrical bore, said bores and said opening being coaxial, a relief valve member through said opening and said bores, said relief valve member provided with a relief valve surface adapted to collaborate with said relief valve seat whereby to close said opening, a cylindrical surface fitting into said suction side cylindrical bore, a cylindrical surface fitting into said pressure side cylindrical bore, a relatively large axial opening and radially spaced openings between said axial opening and said chambers whereby to permit communication between said chambers when said valve and seat are in engagement, said axial opening provided with a first by-pass valve seat on the suction side, a by-pass valve member lodged in said axial opening and on said first by-pass valve seat, said by-pass valve member also provided with a second axial opening and radial openings whereby to set communication between said suction and pressure side chambers and with a second by-pass valve seat on the pressure side, a second by-pass valve member also lodged in said first axial opening and arranged to collaborate with said second by-pass valve seat in closing said second axial opening, a pressure responsive diaphragm resting on said by-pass valve member, and spring means pressing said diaphragm against said by-pass valve member and thereby said by-pass valve member against said first by-pass valve seat and said relief valve member against said relief valve seat.

17. A fluid flow control device including a valve housing and a valve lodged in said housing and movable with reference thereto, a pressure responsive diaphragm operatively resting on said valve, tension means operatively resting at one end on said valve through said diaphragm, a cover for said tension means, tension adjusting means through said cover, said tension means pressing said adjusting means against the inside surface of said cover, and outside said cover additional means operative in further pressing said adjusting means against said cover whereby to provide a fluid tight closure and prevent relative movement of said adjusting means and said cover.

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