June 1, 1965 L. B. TAPLIN 3,186,353

MEANS FOR STABILIZING PUMP PRESSURES

Original Filed Aug. 30, 1960 2 Sheets-Sheet 1

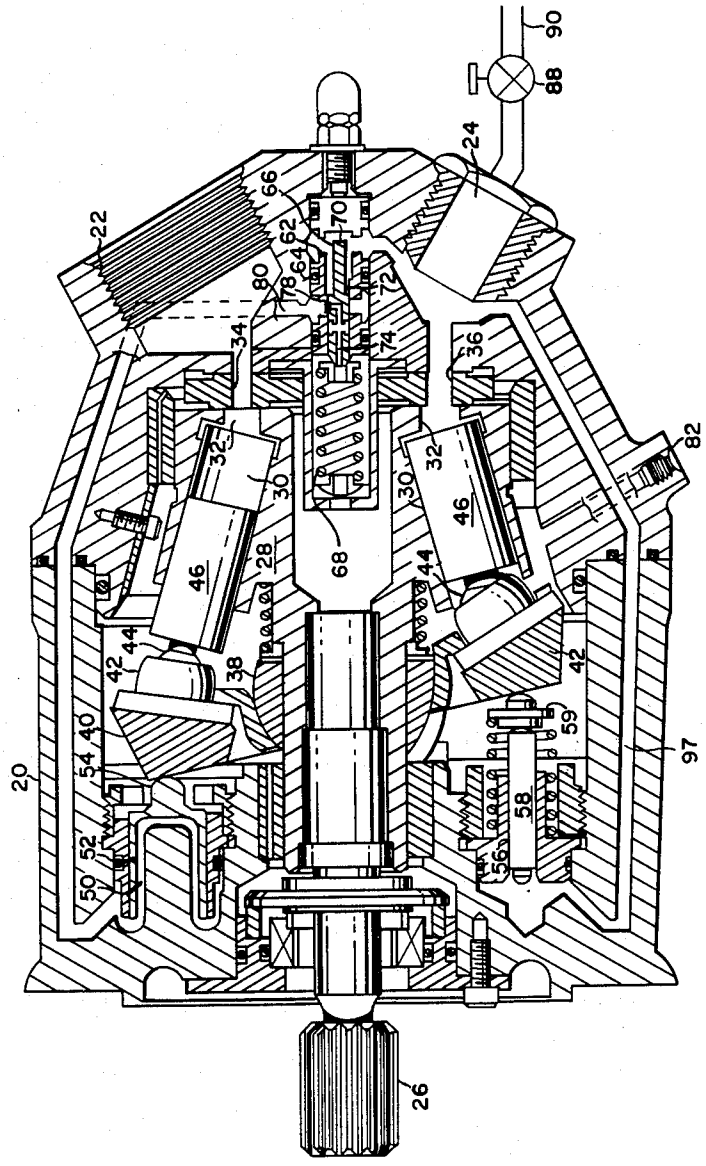

FIG. 1

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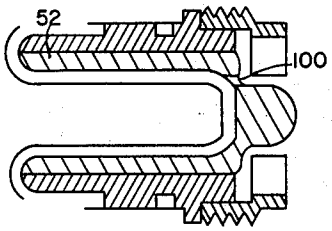

FIG. 3

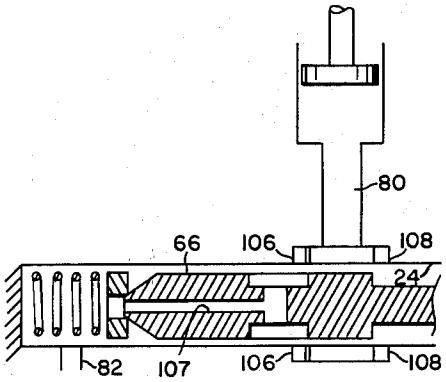

FIG. 6

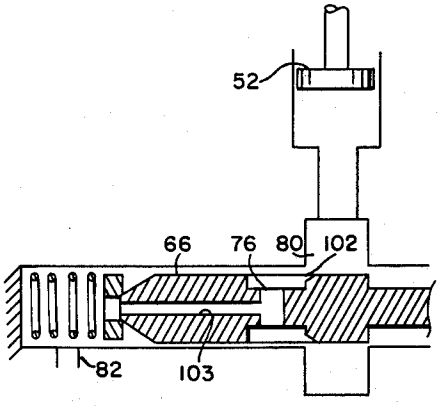

FIG. 4

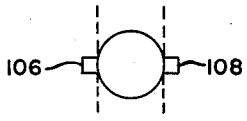

FIG. 7

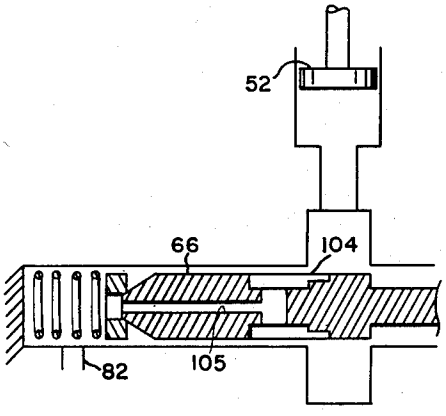

FIG. 5

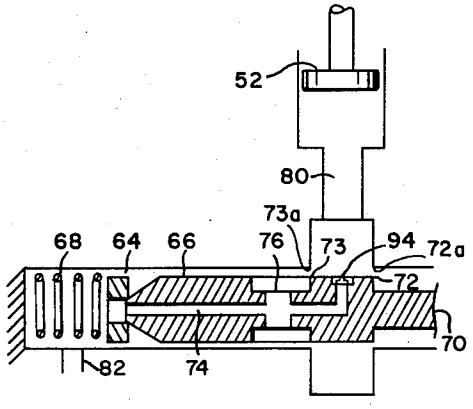

FIG. 2

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MEANS FOR STABILIZING PUMP PRESSURES

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Continuation of application Ser. No. 52,946, Aug. 30, 1960. This application June 22, 1964, Ser. No. 379,429
10 Claims. (Cl. 103—162)

This is a continuation of my copending application Serial No. 52,946, filed August 30, 1960, and entitled "Means for Stabilizing Pump Pressures."

This invention pertains to means for stabilizing pump pressures and, more particularly, to minimizing pressure variations in a variable volume, constant pressure pump.

It is an object of this invention to provide in a constant output pressure pump having a servo valve which is positioned according to any pressure deviation or error from the desired constant pressure to establish a compensating or correcting action to eliminate the error, a minimum of variations and oscillations in the output pressure which are due to servo valve "dead space" or length that the valve must move before the compensating action begins.

It is a further object of this invention to provide means in such a pump that minimizes pressure variations due to the compressibility of the fluid in the pump.

It is a further object of this invention to provide in such a pump having a servo valve operated control piston which acts against a wobble plate to vary the angle of the plate and the delivery of the pump, means for minimizing pressure variations due to the friction between the control piston and the wobble plate.

It is an object of this invention to provide in the pump control circuit, which is that pressure varied by the servo valve to move the control piston, a leakage path to cause said servo valve to remain slightly open at all times to provide a leakage flow, thereby removing "dead space" and to make the pump insensitive to small pressure errors due for example to the friction between the control piston and the wobble plate and to fluid compressibility.

These and other objects and advantages will become more apparent when preferred embodiments of this invention are considered in connection with the drawings in which:

FIGURE 1 shows a schematic, sectioned view of a pump of this invention;

FIGURE 2 shows an enlarged schematic view of the servo valve shown in FIGURE 1;

FIGURE 3 is an enlarged view of a modified control piston;

FIGURE 4 is an enlarged, schematic view of the servo valve of this invention wherein a land of the servo valve is chamfered on its low pressure side to provide a leakage valve;

FIGURE 5 is a view similar to FIGURE 4 wherein the land has an annular groove instead of a chamfer;

FIGURE 6 is an enlarged, schematic view of the servo valve area wherein a leakage path is formed on either side of the port covered by the servo valve land; and

FIGURE 7 is a section taken at 7—7 of FIGURE 6.

In the drawings is shown a pressure compensated wobble plate pump which operates essentially in the manner of wobble plate pumps of the prior art but which has means in the form of a bleed passage in the control pressure portion thereof which minimizes oscillations in the pump due to dead space in the servo valve, friction between the control piston and wobble plate, and fluid compressibility.

A housing 20 has formed therein an input port 22 and an exhaust port 24. Rotatably mounted in housing 20 is a rotor shaft 26 to which is fixed an annular cylinder

block 28 having seven equally spaced cylinders 30 therein. Each cylinder 30 has a passage 32 which registers with an input kidney shaped opening 34 which is fixed in housing 20 and which communicates with input port 22. Passages 32 also register with a kidney shaped port 36 which is in communication with output port 24.

Rotatably mounted to shaft 26 is a half ball joint 38 and fixed against rotation about the longitudinal axis is cam or wobble plate 40 which can wobble about an axis transverse to shaft 26. Placed against one side of and mounted for rotation relative plate 40 are slipper discs 41 to which are attached seven sockets 42 in which are mounted spheres 44 to provide an universal type rotation or movement. Each sphere 44 is fixed to a piston 46, each of which reciprocates in a corresponding cylinder 30.

Formed in housing 20 is a longitudinal cylinder 50 in which reciprocates a control piston 52 having a boss 54 which bears against wobble plate 40 to control its angle about an axis transverse to shaft 26 and hence control the stroke of pistons 46 in cylinders 30. Mounted in housing 20 diametrically opposite from cylinder 50 is a cylinder 56 in which a bias piston 58 is reciprocably mounted with piston 58 acting against a head 59 which is urged by a spring 60 against wobble plate 40 to urge plate 40 against boss 54.

Turning now to the means for supplying fluid pressure to move piston 52 against plate 40, a sleeve 62 is fitted into an axial cylinder 64 formed in housing 20 and a servo valve 66 is mounted for reciprocation in sleeve 62. An enlarged, schematic view of the valve 66 without sleeve 62 is shown in FIGURE 2. Valve 66 is urged in one direction by compression spring 68 and in the opposite direction by the pump output pressure which bears against end 70 and shoulder 72. In the centered position of valve 66, as shown in FIGURE 2, shoulders 72, 73 form metering edges and corners 72*a* and 73*a* form land means with which shoulders 72, 73 register. Formed in valve 66 is an axial passage 74 and a transverse slot 76 intersecting passage 74. When valve 66 moves to the right under pressure of spring 68, slot 76 communicates with an annular passage 78 in sleeve 62 which communicates through passage 80 to control piston 52. In the rightward position of the servo valve 66, the fluid in line 80 goes through slot 76 and passage 74 to the case drain port 82 with the result that the pressure against piston 52 is lessened and the angle of wobble plate 40 increases, thereby increasing the pump output. When servo valve 66 is urged leftwardly until shoulder 72 has been moved past the edge of annular ring 78, the fluid from the pump output is pumped into passage 80 increasing the pressure against piston 52 and reducing the angle of wobble plate 40. This pump is, therefore, a constant pressure pump since as soon as the pressure in output port 24 drops, as would be caused by opening a valve 88 in an output line 90, servo valve 66 is caused to move rightwardly by spring 68 causing fluid to flow from line 80 to the pump reservoir reducing the pressure against piston 54 and increasing the wobble plate angle 40 and increasing pump output. When the quantity of fluid pumped is sufficiently increased, the pressure will again increase in the output of the pump and urge the servo valve 66 in a leftwardly direction against spring 68 until shoulder 72 has cleared the annular groove 78 and output fluid is pumped into line 80 to reduce the angle of the wobble plate 40. Also, there is a passage 97 that is formed between the output port and piston 56 so that the output pressure helps in forcing piston 58 against wobble plate 40.

What has been described to this point is a standard wobble plate constant pressure pump. These type pumps have been subject in the past to oscillations, due to

several causes, which result in pump instability. A first type of instability is caused by the fact that under certain pump conditions there is a tendency for shoulder 72 to move a distance to the right of annular opening 78 when servo valve 66 is in a neutral position so that when the pressure in output port 24 increases there is a "dead space" in the compensating action due to the fact that valve 66 must move a certain distance to the left before there is communication between the output pressure and line 80. Due to this "dead space" the output pressure tends to build to a higher level than is required or desired and therefore control or compensating piston 54 moves too far to the right and then a correction is needed to bring the piston 54 back to where the desired output pressure is reached. The valve 66 then closes or moves to the right until line 80 communicates through annular ring 78 with the case reservoir or drain port 82. This causes unnecessary oscillation of servo valve 66 and pump output pressures.

This instability is compensated for by placement of a small separate bleed passage 94 (FIGURES 1 and 2) in servo valve 66 between line 80 and the case drain or reservoir 82. Since a certain amount of fluid is constantly flowing from passage 94 through passage 74 to the case drain 82, the pressure in line 80 will be slightly lower than it would be normally and this would slightly increase the wobble plate pump output, thereby increasing the pressure against the end of valve 66 until a crack is formed between shoulder 72 and annular groove 78 allowing enough output fluid to flow into line 80 to compensate for the fluid lost through bleed hole 94. In this way, any increase in output pressure results in substantially immediate pressure increase in line 80 and corresponding movement of piston 52. Pressure overshoots are minimized since the "dead space" or overlap is eliminated.

Another cause of pump instability or oscillation is due to the fact that there is a certain amount of frictional resistance to movement of wobble plate 40 by pistons 52 and 58 and trunnion bearings supporting wobble plate 40. This slows movement of the control piston and it will be necessary, therefore, for the servo valve to stay open even after the desired output pressure has been reached. The output pressure changes while the valve is open resulting in overshoots and oscillations. By providing a sufficient leakage path between control pressure line 80 and the fluid reservoir of the pump, the pump becomes less sensitive to minor pressure variations since a small increase in line 80 pressure will result merely in more fluid going through bleed hole 94 with an compensating movement of the piston 52. Therefore, if the leakage path is large enough so that the pump becomes insensitive to the error produced due to the frictional force between the control piston 52 and the wobble plate 40, no oscillations will be introduced.

A further error results in prior art pumps due to compressibility of the fluid. Since the fluid in control passage 80 is compressible there will be a certain lag between an increase in output pressure and a compensating movement of piston 52. During this time, the output pressure will build to a point greater than the desired pressure resulting in a compensating movement of piston 52 in the opposite direction to lower the pressure thereby establishing undesirable oscillations. This too can be eliminated by introduction of a leakage path such as provided by bleed 94 since any increase in pressure due to fluid compressibility will simply cause more fluid to pass through the bleed passage 94.

Bleed passage 94 therefore is designed to be large enough so that any slight variation of pressure in control passage 80 from that pressure which corresponds exactly to the desired pump output pressure will result in increased or decreased flow through the bleed passage rather than a compensating movement of the piston 52. While this increases to a very small extent the variation limits of the desired output pressure, it eliminates undesirable oscillations.

Other methods of introducing the separate leakage path are shown in FIGURES 3–7. FIGURE 3 shows an enlarged view of control piston 52 wherein a small bleed 100 has been formed in the forward wall thereof to provide the communication between the pressure which actuates piston 52 and a lesser pressure such as reservoir 82 pressure. In FIGURE 4 a shoulder on the valve 66 land that is between passages 80 and 76 has been chamfered at 102 so that there is a slight leakage between passage 80 through passage 103 to the case drain or reservoir 82. FIGURE 5 is a view similar to FIGURE 4 except that the chamfer at 102 is replaced by an annular groove 104 in the land of servo valve 66 to provide the leakage between piston 52 through groove 104 and passage 105 to the reservoir 82 pressure. In FIGURES 6 and 7 a leakage is shown between the output pressure passage 24 and the case reservoir 82 by placement of two open areas 106 and 108 on either side of the port to a line 80 permitting passage of fluid around the servo valve 66 when it is in its neutral position through passage 107 to reservoir 82.

The required leakage for any given pump condition varies directly with the desired control pressure in line 80, the desired pump gain, the servo valve gain which is the flow into the control piston line 80 per the pressure change needed to open the valve and the volume of fluid under compression in the control pressure circuit, and varies inversely to the leakage coefficient of the pump, the fluid bulk modulus and the area of the control piston 52.

Although this invention has been disclosed and illustrated with reference to particular applications, the principles involved are susceptible of numerous other applications which will be apparent to persons skilled in the art. The invention is, therefore, to be limited only as indicated by the scope of the appended claims.

Having thus described my invention, I claim:

1. Apparatus comprising
variable volume pumping means having a high pressure region and a low pressure region,
control piston means to vary the volume of said pumping means,
land means,
a metering servo value having at least two edges which are registrable with the land means in a centered position of said valve,
said valve being movable in a first direction so that a first edge moves out of registration with said land means to form a passage between the high pressure region and said control piston,
said valve being movable in a second direction so that a second metering edge moves out of registration with said land means to form a passage between said control piston and the low pressure region,
a separate metering passage for providing a continuous flow between said control piston and said low pressure region thereby keeping said first and second edges in registration with said land means to minimize servo valve dead space and to minimize pump pressure oscillations, pulsing and vibration.

2. The apparatus of claim 1 with said metering passage comprising a metering passage in said servo valve.

3. The apparatus of claim 2 with one end of said metering passage being between said two metering edges and the other end communicating with said low pressure area.

4. The apparatus of claim 1 with said metering passage being in said control piston.

5. The apparatus of claim 2 with one end of said metering passage between at least one of said two metering edges of said servo valve and said land means.

6. An apparatus comprising a housing,
a shaft being rotatable in said housing,

a wobble plate being mounted to wobble axially of said shaft,

an annular cylinder block having a plurality of cylinders formed therein being connected with said shaft,

a multiplicity of pistons being universally connected to a slipper disc and cooperating with corresponding cylinders,

said disc being rotatable against said plates,

an input port,

an output port,

said cylinders alternately communicating with said output and input ports,

control piston means being reciprocably mounted in said housing and engageable with the sides of said wobble plate axially opposite to the side said pistons are connected to,

reservoir means,

a servo valve being reciprocably mounted in said housing,

said valve being spring urged at one end to a position causing communication between said control piston and said reservoir means and the other end of said valve communicating with the output pressure which output pressure urges said valve to a position causing communication between said control piston and the output pressure,

whereby a decrease in output pressure which signifies a decrease in output flow would cause said servo valve to move to a position connecting said control piston means to said reservoir means thereby increasing output flow and whereby an increase in output pressure which signifies an increase in output flow will cause said servo valve to move to a position connecting said control piston means to output pressure thereby decreasing output flow,

said servo valve having a neutral position between the positions of connecting said control piston means to the reservoir means and the output pressure,

separate bleed means causing communication between said control piston means and said reservoir means when said servo valve is in said neutral position to minimize servo valve oscillations,

said bleed means causing a continuous communication between said control piston and said reservoir means.

7. The apparatus of claim **6** with said metering passage comprising a metering passage in said servo valve.

8. The apparatus of claim **7** with one end of said metering passage between said two metering edges and the other end communicating with said low pressure area.

9. The apparatus of claim **6** with said metering passage being in said control piston.

10. The apparatus of claim **7** with one end of said metering passage between at least one of said two metering edges of said servo valve and said land means.

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