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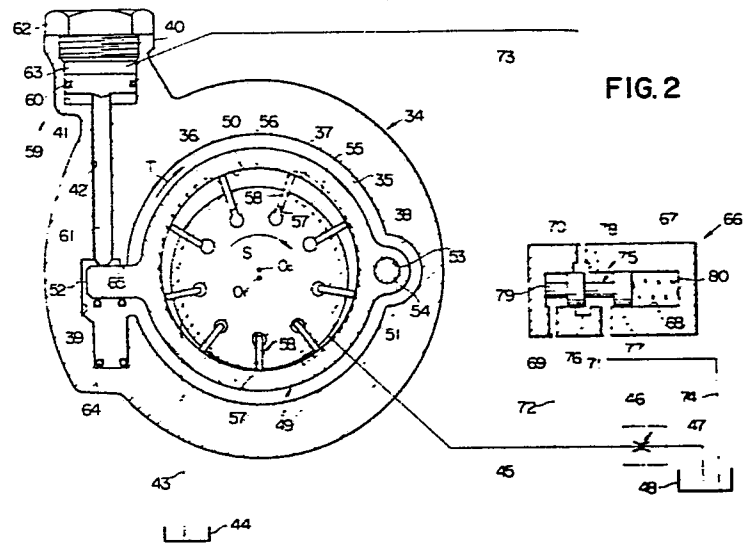
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⑤④ **Variable-displacement sliding-vane pump.**

⑤⑦ A variable-displacement sliding vane pump in which a generally cylindrical cam member (49) formed with an axial bore (50) is positioned in a cavity (35) in a pump housing (34) and is rockable about an axis (54) fixed with respect to the pump housing (34) and in which a cylindrical rotor (55) is rotatable in the axial bore (50) in the cam member (49) about an axis (Or) fixed with respect to the pump housing (34) and offset from the center axis (Oc) of the axial bore (50) in the cam member (49) and is formed with radial slots (57) having vanes (58) slidably received in the slots (57), respectively, wherein a pressure-acting member (60) responsive to variation in the pressure of the fluid delivered from the pump chamber (56) formed between the rotor (55) and the cam member (49) is operative to drive the cam member (49) to turn about the axis (54) of rocking motions of the cam member (49) in a direction to reduce the amount of eccentricity between the axis (Or) of rotation of the rotor (55) and the center axis (Oc) of the cam member (49) in response to an increase in the fluid pressure, the pressure-acting member (60) being constructed separately of the pump housing (34) and being urged to turn the cam member (49) about the axis (54) of rocking motions thereof in a direction to increase the aforesaid amount of eccentricity.

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DESCRIPTION

The present invention relates to a variable-displacement sliding-vane pump for use with, for example, a fluid-operated actuator of a hydraulic control system of an automatic power transmission for an automotive vehicle.

In accordance with the present invention, there is provided a variable-displacement sliding-vane pump which comprises in combination a hollow pump housing formed with a cavity, a fluid inlet port and a fluid discharge port, a hollow cam member positioned in the cavity in the pump housing and rockable in the cavity about an axis fixed with respect to the pump housing, the cam member being formed with an axial bore having a center axis substantially parallel with the axis of rocking motions of the cam member, a cylindrical rotor positioned in the axial bore in the cam member and rotatable in the axial bore about an axis substantially parallel with and offset from the center axis of the bore in the cam member, the amount of eccentricity between the center axis of the axial bore and the axis of rotation of the rotor being variable depending upon the angular position of the cam member with respect to the rotor about the axis of rocking motions of the cam member, the rotor forming between the inner peripheral surface of the cam member and the outer peripheral surface of the rotor a variable-volume pump chamber being formed with a plurality of radial slots which are open radially outwardly of the rotor,

vanes respectively slidable in the slots in the rotor and movable toward and away from the inner peripheral surface of the cam member, and displacement adjusting means for varying the aforesaid amount of eccentricity in response to a change
5 in the pressure of the fluid delivered from the pump chamber through the fluid discharge port, the displacement adjusting means comprising a pressure-acting member which is responsive to variation in the pressure of the fluid delivered from the pump chamber through the fluid discharge port and which is
10 operative to drive the cam member to turn with respect to the pump housing and to the rotor about the axis of rocking motions of the cam member in a direction to reduce the aforesaid amount of eccentricity, the pressure-acting member being constructed separately of the cam member, and biasing
15 means urging the cam member to turn with respect to the pump housing and to the rotor about the axis of rocking motions of the cam member in a direction to increase the aforesaid amount of eccentricity.

20 Problems encountered in prior-art variable-displacement sliding-vane pump and detailed features and advantages of a variable-displacement sliding-vane pump according to the present invention will be understood from the following description taken in conjunction with the accompanying
25 drawings, in which:

Fig. 1 is a view showing, partly in cross section and partly in longitudinal section, an example of a prior-art

variable-displacement sliding-vane pump of the type to which the present invention appertains;

Fig. 2 is a view similar to Fig. 1 but shows a preferred embodiment of a variable-displacement sliding-vane pump
5 according to the present invention;

Fig. 3 is a cross sectional view showing part of a modification of the embodiment illustrated in Fig. 2; and

Fig. 4 is a fragmentary cross sectional view showing part of another modification of the embodiment illustrated in
10 Fig. 2.

A representative example of a conventional variable-displacement sliding-vane pump of the nature to which the present invention appertains is shown in Fig. 1 of
15 the drawings. The prior-art pump herein shown is taught in Japanese Preliminary Publication of Patent No. 55-17696.

The variable-displacement sliding-vane pump comprises a pump housing 1 having defined therein a generally cylindrical cavity 2 which is axially elongated between opposite end wall
20 portions (not shown) of the housing 1. The housing 1 is further formed with a fluid inlet port 3 and a fluid discharge port 4. The fluid inlet port 3 communicates with a fluid reservoir 5 through a fluid feed passageway 6 while the fluid discharge port 4 communicated with a suitable
25 fluid-operated device 7 through a fluid delivery passageway 8. In the cavity 2 of the pump housing 1 is positioned a generally cylindrical eccentric cam member 9 formed with an

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axial bore 10 and pivotally retained to an internal surface portion of the housing 1 by means of a pivot pin 11. The pivot pin 11 is fitted to the outer peripheral wall of the cam member 9 in such a manner that the cam member 9 is
5 rockable within the cavity 2 in the housing 1 about the center axis of the pivot pin 11. The cam member 9 has formed in its outer peripheral wall an axial groove diametrically opposite to the pivot pin 11 and has a sealing strip 12 fitted in the groove. The sealing strip 12 is held in
10 slidable contact with an internal surface portion of the pump housing 1 and defines, together with the pivot pin 11, a control fluid chamber 13 between a generally semicylindrical inner peripheral surface portion of the pump housing 1 and a generally semicylindrical outer surface portion of the cam
15 member 9. The cam member 9 further has a lug portion 14 radially projecting from the outer peripheral wall of the cam member 9 opposite to the control fluid chamber 13. A helical compression spring 15 is seated between the lug portion 14 and an internal surface portion of the pump housing 1 and
20 urges the cam member 9 to turn about the center axis of the pivot pin 11 in a direction to reduce the volume of the control fluid chamber 13. The outer peripheral wall of the cam member 9 further has a radial protrusion 16 protruding into the control fluid chamber 13. The protrusion 16 is
25 forced against an internal surface portion of the pump housing 1 by the force of the compression spring 15.

Within the axial bore 10 in the eccentric cam member 9

is positioned a cylindrical rotor 17 which is rotatable with respect to the cam member 9 about axis fixed with respect to the pump housing 1 and offset from the center axis of the bore 10 in the cam member 9. The amount of eccentricity
5 between the axis of rotation of the rotor 17 and the center axis of the bore 10 in the cam member 9 is variable depending upon the angular position of the cam member 9 with respect to the housing 1 about the center axis of the pivot pin 11. The rotor 17 thus rotatable with respect to the cam member 9
10 forms a variable-volume pump chamber 18 between the inner peripheral surface of the cam member 9 and the outer peripheral surface of the rotor 17. The pump chamber 18 is variable in volume depending upon the above mentioned angular position of the cam member 9 with respect to the housing 1.
15 The pump chamber 18 communicates on one hand with the fluid inlet port 3 and on the other hand with the fluid discharge port 4. The rotor 17 is formed with a multiplicity of radial slots 19 which are arranged symmetrically about the axis of rotation of the rotor 17 and which are open radially
20 outwardly of the rotor 17. A vane 20 is slidably fitted in each of the slots 19 and projects radially outwardly of the rotor 17 into the pump chamber 18.

The rotor 17 is operatively connected to suitable drive means (not shown) and is driven for rotation about the center
25 axis thereof in a direction indicated by arrow P. As the rotor 17 is thus driven to rotate within the cam member 9, the fluid admitted into the pump chamber 18 through the fluid

inlet port 3 is pressurized in the pump chamber 18 and is delivered through the fluid discharge port 4 to the fluid delivery passageway 8 and by way of the passageway 8 to the fluid-operated device 7.

5 The prior-art variable-displacement sliding-vane pump shown in Fig. 1 further comprises a pressure regulator valve 21 having a valve housing 22 formed with an axial bore 23, a fluid inlet port 24, a control fluid outlet port 25 and a fluid discharge port 26. The fluid inlet port 24
10 communicates through a passageway 27 with the fluid delivery passageway 8 leading from the fluid discharge port 4 in the pump housing 1, while the control fluid outlet port 25 communicates through a passageway 28 with the control fluid chamber 13 formed between the pump housing 1 and the cam
15 member 9. The fluid discharge port 26 communicates with a sump 29 through a passageway 30. A valve spool 31 having a cylindrical land portion 32 is axially slidable in the axial bore 23 in the valve housing 22 and is urged by a spring 33 to stay in an axial position isolating the ports 24 and 25
20 from each other by the land portion 32 thereof, as shown.

When the rotor 17 of the pump constructed and arranged as described hereinbefore is being driven for rotation at an increasing speed, the pressure of the fluid delivered to the fluid-operated device 7 through the fluid delivery passageway
25 8 increases accordingly. The increasing fluid pressure is directed through the passageway 27 to the fluid inlet port 24 of the regulator valve 21 and acts on the land portion 32 of

the valve spool 31. The valve spool 31 is thus urged to axially move in the bore 23 in the valve housing 22 against the force of the spring 33 as indicated by arrow Q. When the fluid pressure acting on the land portion 32 of the valve

5 spool 31 is increased beyond a certain value, the valve spool 31 is caused to axially move in the direction of the arrow Q in the axial bore 23 of the valve housing 22 and provides communication between the fluid inlet port 24 and the control fluid outlet port 25 while blocking the communication between

10 the control fluid outlet port 25 and the fluid discharge port 26. The fluid delivered from the pump is, thus, passed not only to the fluid-operated device 7 but through the passageway 27, pressure regulator valve 21 and passageway 28 to the control fluid chamber 13. The fluid pressure directed

15 into the control fluid chamber 13 acts on the semicylindrical outer peripheral surface portion of the cam member 9 and forces the cam member to turn about the center axis of the pivot pin 11 in a direction indicated by arrow R against the force of the spring 15. The cam member 9 being thus caused

20 to turn with respect to the rotor 17 positioned therewithin, there is caused a change in the amount of eccentricity between the center axis of the axial bore 10 in the cam member 9 and the axis of rotation of the rotor 17. The change in the eccentricity results in a change in the volume

25 of the pump chamber 18 and accordingly in a change in the amount of fluid displacement of the pump. the increase in the fluid pressure as caused by an increase in the revolution

speed of the rotor 17 is in this manner compensated by the change in the amount of fluid displacement of the pump and, thus, the pressure of the fluid delivered from the pump is maintained at a practically fixed value.

5 In order that the fluid pressure directed into the control fluid chamber 13 be capable of properly acting on the semicylindrical outer peripheral surface portion of the cam member 9, it is important that the control fluid chamber 13 be effectively sealed to preclude leakage of fluid across the
10 pivot pin 11 and the sealing strip 12. Since, in this instance, both of the pivot pin 11 and the sealing strip 12 are subject to sliding motion, the sealing performance to be achieved by each of the pivot pin 11 and the sealing strip 12 is primarily affected by the dimensional accuracy of the
15 internal surfaces of the pump housing 1 and the external surfaces of the cam member 9. There may thus be a case where satisfactory sealing is achieved at the pivot pin 11 and the sealing strip 12 but excessive sliding friction is produced between the pivot pin 11 and the cam member 9 or between the
20 sealing strip 12 and the pump housing 1. The excessive sliding friction produced therebetween would result in deterioration in the ability of the cam member 9 to move in response to a change in the fluid pressure in the control fluid chamber 13 and accordingly in the ability of the pump
25 to adjust its fluid displacement in response to a change in the revolution speed of the rotor 17. There may otherwise be a case where the fluid displacement characteristics of the

pump can be adjusted satisfactorily in response to a change in the fluid pressure in the control fluid chamber 13 but the sealing performance achieved by the pivot pin 11 and the sealing strip 12 is unsatisfactory. To avoid these
5 contradictory problems, it is of paramount importance to have the pump housing 1 and the cam member 9 machined with utmost accuracy during manufacture thereof.

Another problem encountered in a prior-art pump of the described nature results from the fact that the pivot pin 11
10 is received in part in a groove formed in the internal wall of the pump housing 1 and in part in a groove formed in the outer wall of the cam member 9. Extra precision machining is thus required for the formation of these grooves in the pump housing 1 and the cam member 9 during manufacture of these
15 members.

The present invention contemplates solution of these and other problems which have thus far been encountered in a variable-displacement sliding-vane pump of the described nature.

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Referring to Fig. 2 of the drawings, a variable-displacement sliding-vane pump embodying the present invention comprises a hollow pump housing 34 having defined
25 therein a generally cylindrical cavity 35 having a center axis therethrough and axially elongated between opposite end wall portions (not shown) of the housing 34. One of the end

wall portions of the pump housing 34 is formed with fluid inlet and discharge ports 36 and 37 extending arcuately about the center axis of the cavity 35 in diametrically opposite relationship to each other across the center axis of the

5 cavity 35 as indicated by broken lines. The pump housing 34 is further formed with a generally semicylindrical groove 38 protruding radially outwardly from the cavity 35 and extending in parallel with the center axis of the cavity 35 and a recess 39 partially protruding radially outwardly from

10 the cavity 35 in diametrically opposite relationship to the groove 38 and partially elongated in a direction tangential to the cavity 35. The fluid inlet port 36 is located closer to the recess 39 and the fluid discharge port 37 is located closer to the groove 38. The pump housing 34 further has a

15 lug portion 40 located in conjunction with the recess 39 and is formed with a generally cylindrical concavity 41 which is open outwardly of the pump housing 34 and an elongated bore 42 axially extending between the recess 39 and the concavity 41 and open at one end thereof to the recess 39 and at the

20 other end thereof to the concavity 41. The fluid inlet port 36 communicates through a fluid feed passageway 43 with a suitable source of fluid such as a fluid reservoir 44, while the fluid discharge port 37 communicates through a fluid delivery passageway 45 with a suitable fluid-operated device

25 such as a hydraulic actuator 46 for use in, for example, the hydraulic control system of an automatic power transmission for an automotive vehicle. The hydraulic actuator 46 has a

fluid discharge port communicating through a passageway 47 with a sump 48.

In the cavity 35 of the pump housing 34 is positioned a generally cylindrical eccentric cam member 49 formed with an axial bore 50 having a center axis O_c parallel with the center axis of the cavity 35 in the pump housing 34. The cam member 49 has radial protrusions 51 and 52 radially outwardly protruding from the outer peripheral wall of the cam member 39 in diametrically opposite relationship to each other across the center axis O_c of the axial bore 50. One radial protrusion 51 projects into the semicylindrical groove 38 in the pump housing 34 and is formed with a cylindrical hole 53 having a center axis parallel with the center axis of the bore 50. A pivot pin 54 is slidably passed through the cylindrical hole 53 and is connected at the opposite axial ends thereof to the opposite end wall portions of the pump housing 34. The cam member 49 is thus pivotally connected to the end wall portions of the pump housing 34 and is rockable about the center axis of the pivot pin 54 with respect to the pump housing 34. The other radial protrusion 52 of the cam member 49 projects into the recess 39 in the pump housing 34 and is movable in an arc in the recess 39. The previously mentioned axial bore 42 in the pump housing 34 extends in a direction perpendicular in non-intersecting relationship to the center axis of the pivot pin 54, viz., the axis about which the cam member 49 is rockable with respect to the pump housing 34.

In the axial bore 50 of the cam member 49 is positioned a cylindrical rotor 55 which has a center axis O_r parallel with the center axis O_c of the axial bore 50 in the cam member 49 and fixed with respect to the pump housing 34 and
5 which is rotatable about the center axis O_r thereof with respect to the pump housing 34 and the cam member 49. The center axis O_r of the rotor 55 is offset from the center axis of the axial bore 50 in the cam member 49 so that a variable-volume pump chamber 56 having a crescent-shaped
10 cross section is formed between the inner peripheral surface of the cam member 49 and the outer peripheral surface of the rotor 55. The amount of eccentricity between the center axis O_c of the cam member 49 and the center axis O_r of the rotor 55 and accordingly the volume of the pump chamber 56 thus
15 formed between the cam member 49 and the rotor 55 are variable depending upon the angular position of the cam member 49 about the center axis of the pivot pin 54 with respect to the rotor 55. The pump chamber 50 communicates on one hand with the fluid inlet port 36 and on the other hand
20 with the fluid discharge port 37 in the pump housing 34. The rotor 55 is formed with a multiplicity of radial slots 57 which are arranged symmetrically about the center axis O_r of the rotor 55 and which are open radially outwardly of the rotor 55. Vanes 58 are respectively fitted in these slots 57
25 and project radially outwardly from the slots 57 into the pump chamber 56.

The rotor 55 is operatively connected to suitable drive

means such as, for example, the output shaft of an engine (not shown) installed on an automotive vehicle and is driven for rotation about the center axis O_r thereof in a direction indicated by arrow S. As the rotor 55 is thus driven to

5 rotate in the axial bore 50 of the cam member 49, the individual vanes 58 respectively received in the radial slots 57 in the rotor 55 are forced to slide radially outwardly in the slots 57 by a centrifugal force exerted on each of the vanes 58 and are caused to slide at their respective outer

10 ends on the inner peripheral surface of the cam member 49. As a consequence, the pump chamber 56 formed between the inner peripheral surface of the cam member 49 and the outer peripheral surface of the rotor 55 is divided into a multiplicity of sections by the vanes 58 thus forced against

15 the inner peripheral surface of the cam member 49. As the rotor 55 is rotated about the center axis O_r thereof with respect to the cam member 49, each of the vanes 58 is caused to alternately move radially inwardly and outwardly of the rotor 55 due to the eccentricity between the center axis O_c

20 of the bore 50 in the cam member 49 and the center axis O_r of the rotor 55, with the result that each section of the pump chamber 56 is alternately expanded and contracted. The fluid supplied from the fluid reservoir 44 through the fluid feed passageway 43 enters expanding sections of the pump chamber

25 56 through the fluid inlet port 36 in the pump housing 34 and is pressurized progressively as the particular sections of the pump chamber 56 are reduced in volume. The fluid trapped

in each of the sections of the pump chamber 56 is in this fashion pressurized as the rotor 55 is rotated and the individual sections are expanded and contracted. The pressurized fluid is delivered to the fluid delivery
5 passageway 45 through the fluid discharge port 37 in the pump housing 34 and by way of the passageway 45 to the hydraulic actuator 46.

The variable-displacement sliding-vane pump thus constructed and operative in accordance with the present
10 invention further comprises displacement adjusting means 59 responsive to the fluid pressure delivered from the fluid discharge port 37 and operative to automatically adjust the angular position of the cam member 49 about the center axis of the pivot pin 54 with respect to the pump housing 34 and
15 accordingly to the rotor 55 during operation of the pump.

In the embodiment of the variable-displacement sliding-vane pump illustrated in Fig. 2, the previously described radial protrusion 52 of the cam member 49 forms part of such displacement adjusting means 59 which further
20 comprises a piston 60 axially slidable in the concavity 41 in the lug portion 40 of the pump housing 34 and a plunger 61 axially slidable through the axial bore 42 in the pump housing 34. The plunger 61 is securely connected at one end thereof to the piston 60 and is engaged at the other end
25 thereof by the above mentioned radial protrusion 52 of the cam member 49 through the recess 39 in the pump housing 34. The lug portion 40 of the pump housing 34 is in part

internally threaded and is capped by an externally threaded plug member 62 which thus closes the concavity 41 in the lug portion 40. Between the outer end face of the piston 60 and the inner end face of the plug member 62 is thus formed a
5 control fluid chamber 63 which forms part of the concavity 41 in the lug portion 40.

The displacement adjusting means 59 further comprises biasing element adapted to urge the cam member 49 to turn about the center axis of the pivot pin 54 in a predetermined
10 direction indicated by arrow T and to urge the piston 60 to axially move in a direction to contract the control fluid chamber 63. In the embodiment illustrated in Fig. 2, such a biasing element is constituted by a preloaded helical compression spring 64 which is seated at one end thereof on
15 the radial protrusion 52 of the cam member 49 and at the other end thereof an internal surface portion defining part of the recess 39 in the pump housing 34 as shown. To limit the angular movement of the cam member 49 in the direction of the arrow T about the center axis of the pivot pin 54 with
20 respect to the pump housing 34 and accordingly to the rotor 55, the pump housing 34 has an internal projection 65 projecting into the recess 39 and located so that the radial protrusion 52 of the cam member 49 is engageable therewith when turned in the direction of the arrow T by the force of
25 the spring 64. When the cam member 49 is held in the angular position having the protrusion 52 engaged by the projection 65, the amount of eccentricity between the center axis O_c of

the cam member 49 and the center axis O_r of the rotor 55 becomes maximum so that the volume of the pump chamber 56 and accordingly the amount of fluid displacement achieved by the pump also become maximum.

5 The variable-displacement sliding-vane pump embodying the present invention further comprises a pressure regulator valve 66 responsive to variation in the pressure of the pressurized fluid delivered from the fluid discharge port 37 to the fluid delivery passageway 45 and operative to pass the
10 pressurized fluid from the passageway 45 to the above mentioned control fluid chamber 63 when the fluid pressure is higher than a predetermined value. In the embodiment illustrated in Fig. 2, such a pressure regulator valve 66 is constructed similarly to its counterpart in the prior-art
15 pump illustrated in Fig. 1 and, thus, comprises a valve housing 67 formed with an axial bore 68, a fluid inlet port 69, a control fluid outlet port 70, and a fluid discharge port 71. The fluid inlet port 69 communicates through a passageway 72 with the fluid delivery passageway 45, while
20 the control fluid outlet port 70 communicates through a passageway 73 with the control fluid chamber 63 of the displacement adjusting means 59. The fluid discharge port 71 of the valve 66 communicates with the previously mentioned sump 48 through a passageway 74.

25 The pressure regulator valve 66 further comprises a valve spool 75 axially slidable in the axial bore 68 of the valve housing 67 and having first and second cylindrical land

portions 76 and 77 axially spaced apart from each other so as to form a circumferential groove 78 therebetween. The valve spool 75 further has a stem portion 79 axially projecting from the first land portion 76 and engageable at its leading end with the inner face of one end wall portion of the valve housing 67. The fluid inlet port 69 is located adjacent the inner face of the above mentioned end wall portion of the valve housing 67, while the control fluid outlet port 70 is located intermediate between the fluid inlet port 69 and the fluid discharge port 71 as shown. The valve spool 75 is urged by suitable biasing means to stay in a predetermined axial position having the stem portion 79 held in abutting engagement at the leading end thereof with the inner face of the particular end wall portion of the valve housing 67. In the arrangement of Fig. 2, such biasing means is shown comprising a preloaded helical compression spring 80 seated between the outer end face of the second land portion 77 of the valve spool 75 and the inner face of the other end wall portion of the valve housing 67. When the valve spool 75 is held in the predetermined axial position in the axial bore 68, the first land portion 76 of the valve spool 75 is located to isolate the fluid inlet port 69 and the control fluid outlet port 70 from each other and provide communication between the control fluid outlet port 70 and the fluid discharge port 71. The valve spool 75 is movable from the above mentioned predetermined axial position against the force of the compression spring 80 into an axial position

providing communication between the fluid inlet port 69 and the control fluid outlet port 70 and isolating the control fluid outlet port 70 and the fluid discharge port 71 from each other by the first land portion 76 thereof.

5 In operation, the pressurized fluid delivered from the fluid discharge port 37 in the pump housing 34 to the fluid delivery passageway 45 is passed not only to the hydraulic actuator 46 but through the passageway 72 to the fluid inlet port 69 of the pressure regulator valve 66. The fluid
10 pressure thus directed into the axial bore 68 of the valve housing 67 acts on the end face of the stem portion 79 and the annular outer end face of the first land portion 76 of the valve spool 75 and urges the valve spool 75 to move from the above mentioned predetermined axial position thereof
15 against the opposing force of the spring 80. If, in this instance, the fluid pressure thus acting on the valve spool 75 is lower than a predetermined value, the valve spool 75 is held in the predetermined axial position in the axial bore 68 in the valve housing 67 by the force of the spring 80. The
20 predetermined value of the fluid pressure is dictated by the cross sectional area of the land portion 76 of the valve spool 75 and the spring constant of the spring 80. Under these conditions, the pressurized fluid delivered to the fluid delivery passageway 45 is totally passed to the
25 hydraulic actuator 46 and, as a consequence, there is no fluid pressure developed in the control fluid chamber 63 of the displacement adjusting means 59. The piston 60 is, thus

not operative to exert an action on the radial protrusion 52 of the cam member 49, which is therefore held by the force of the spring 64 in the previously mentioned angular position having the radial protrusion 52 engaged by the internal
5 projection 65 of the pump housing 34. A maximum amount of eccentricity is thus achieved between the center axis O_c of the cam member 49 and the center axis O_r of the rotor 55 so that the pump delivers pressurized fluid to the hydraulic actuator 46 with a maximum fluid displacement rate. The
10 maximum fluid displacement rate is achieved when the rotor 55 is being driven for rotation at speeds lower than a predetermined value. If, thus, the rotor 55 is driven by an internal combustion engine installed on an automotive vehicle, such speeds are produced during idling of the
15 engine.

As the speed of rotation of the rotor 55 increases, the fluid displacement rate of the pump also increases and, as a consequence, the pressure of the pressurized fluid delivered through the fluid discharge port 37 of the pump to the fluid
20 delivery passageway 45 rises. When the fluid pressure in the fluid delivery passageway 45 reaches the previously mentioned predetermined value thereof, the valve spool 75 of the pressure regulator valve 66 is caused to move out of the predetermined axial position thereof against the opposing
25 force of the spring 80, providing communication from the fluid inlet port 69 to the control fluid outlet port 70 and isolating the control fluid outlet port 70 from the fluid

discharge port 71. The increased fluid pressure is thus directed through the passageway 72, the ports 69 and 70 of the pressure regulator valve 66 and the passageway 73 to the control fluid chamber 63 of the displacement adjusting means
5 59 and acts on the outer end face of the piston 60. The piston 60 is consequently urged to move in a direction to expand the control fluid chamber 63 and the plunger 61 is forced at its leading end against the radial protrusion 52 of the cam member 49, urging the radial protrusion 52 to move
10 away from the piston 60 against the opposing force of the spring 64 and thereby urging the cam member 49 to turn about the center axis of the pivot pin 54 in a direction opposite to the direction of the arrow T. If, in this instance, the fluid pressure directed into the control fluid chamber 63 is
15 lower than a predetermined value which is dictated by the cross sectional area of the piston 60 and the spring constant of the spring 64, the cam member 49 is maintained in the angular position having the radial protrusion 52 engaged by the internal projection 65 of the pump housing 34 so that the
20 pump remains in a condition delivering pressurized fluid with the maximum fluid displacement rate. If, however, the fluid pressure in the control fluid chamber 63 is higher than the above mentioned predetermined value, the piston 60 and accordingly the plunger 61 are caused to axially move away
25 from the plug member 62 so that the radial protrusion 52 of the cam member 49 is moved in the recess 39 away from the internal projection 65 of the pump housing 34 against the

opposing force of the spring 64. It therefore follows that the cam member 49 is caused to turn about the center axis of the pivot pin 54 in the direction opposite to the direction of the arrow T from the initial angular position thereof.

5 This produces reduction in the amount of eccentricity between the respective center axes O_c and O_r of the cam member 49 and the rotor 55 and accordingly reduction in the fluid displacement rate of the pump. The decrease in the fluid displacement rate of the pump in turn results in decrease in
10 the pressure of the fluid delivered to the hydraulic actuator 46 and passed through the pressure regulator valve 66 to the control fluid chamber 63 of the displacement adjusting means 59.

When the fluid displacement rate of the pump is thus
15 reduced and as a consequence the fluid pressure acting on the valve spool 75 of the pressure regulator valve 66 drops below the previously mentioned predetermined value thereof, the valve spool 75 is caused to move toward the initial predetermined axial position in the axial bore 68 of the
20 valve housing 67 by the force of the spring 80 and restricts the passage of fluid from the fluid inlet port 69 to the control fluid outlet port 70 and accordingly from the passageway 72 to the control fluid chamber 63 of the displacement adjusting means 59 through the passageway 73.
25 This causes reduction in the fluid pressure in the control fluid chamber 63 and accordingly reduction in the force exerted on the radial protrusion 52 of the cam member 49 by

the piston 60 and the plunger 61, with the result that the cam member 49 is caused by the force of the spring 64 to turn about the center axis of the pivot pin 54 in the direction of the arrow T toward the initial angular position providing the maximum fluid displacement rate of the pump.

If the delivery rate of fluid through the fluid discharge port 37 to the fluid delivery passageway 45 and accordingly the pressure of the fluid passed to the hydraulic actuator 46 and the pressure regulator valve 66 are reduced excessively, the valve spool 75 of the pressure regulator valve 66 is caused to move into the initial predetermined axial position in the axial bore 68 by the force of the spring 80. The passage of fluid from the fluid inlet port 69 to the control fluid outlet port 70 of the pressure regulator valve 66 is thus further diminished and as a consequence the cam member 49 is further turned about the center axis of the pivot pin 54 toward the initial angular position providing the maximum fluid displacement rate of the pump.

The amount of eccentricity between the respective center axes O_c and O_r of the cam member 49 and the rotor 55 is in these manners continuously regulated so that the force exerted on the cam member 49 by the fluid pressure acting on the piston 60 is at all times balanced with the force of the spring 64 and accordingly the pressure of the fluid delivered from the fluid discharge port 37 to the fluid delivery passageway 45 is constantly maintained at a practically fixed value.

Fig. 3 of the drawings show a modification of the hereinbefore described embodiment of the present invention. In the modified embodiment illustrated in Fig. 3, two displacement adjusting means 81 and 81' are provided in lieu of the single displacement adjusting means 59 in the
5 embodiment of Fig. 2. For this purpose, the pump housing 34 in the embodiment of Fig. 3 is formed with two recesses 82 and 82' in addition to the generally cylindrical cavity 35, fluid inlet and discharge ports 36 and 37 and generally
10 semicylindrical groove 38. The recesses 82 and 82' protrude radially outwardly from the cavity 35 and are located at an angular spacing of approximately 90 degrees about the center axis of the cavity 35. The pump housing 34 is further formed with cylindrical bores 83 and 83' located in conjunction with
15 these two recesses 82 and 82', respectively. Plungers 84 and 84' are axially slidable each partially in these bores 83 and 83' and project into the recesses 82 and 82', respectively, in the pump housing 34. The plungers 84 and 84' axially extend and are movable with respect to the pump housing 34 in
20 directions substantially tangential to the outer peripheral surface of the cam member 49. Each of the plungers 84 and 84' forms in each of the bores 83 and 83' a control fluid chamber communicating with the control fluid outlet port of a pressure regulator valve (not shown) similar to the valve 66
25 in the embodiment of Fig. 2.

In the embodiment illustrated in Fig. 3, furthermore, the generally cylindrical cam member 49 has two radial

protrusions 85 and 85' radially projecting outwardly from the outer peripheral wall of the cam member 49. The radial protrusions 85 and 85' are angularly spaced apart through the angle of approximately 90 degrees about the center axis of the cylindrical bore 50 in the cam member 49 and project into the above mentioned recesses 82 and 82', respectively, in the pump housing 34. More specifically, one radial protrusion 85 is located approximately in diametrically opposite relationship to the pivot pin 54 across the center axis of the axial bore 50 in the cam member 49, and the other radial protrusion 85' is angularly spaced apart through approximately 45 degrees from each of the pivot pin 54 and the former radial protrusion 85 as shown. The above mentioned plungers 84 and 84' are engaged at their respective leading ends by these radial protrusions 85 and 85', respectively.

As in the embodiment describes with reference to Fig. 2, the cam member 49 of the embodiment illustrated in Fig. 3 is urged by suitable biasing means to turn about the center axis of the pivot pin 54 in a direction to provide a maximum amount of eccentricity between the center axis O_c of the axial bore 50 in the cam member 49 and the center axis O_r of the rotor 55. In the embodiment of Fig. 3, such biasing means comprises a torsion spring 86 located adjacent one axial end of the cam member 49 and having a helical intermediate portion wound on an end portion of the pivot pin 54, an arm portion merging in one direction out of the

helical intermediate portion and seated on a spring seat element 87 secured to one end wall portion (not shown) of the pump housing 34, and an arm portion merging in another direction out of the helical intermediate portion and seated on a projection 88 of the cam member 49. The projection 87 of the cam member 49 projects from one end wall portion 89 of the cam member 49. The cam member 49 is, thus, urged to turn with respect to the pump housing 34 and accordingly to the rotor 55 about the center axis of the pivot pin 54 in a direction (indicated by arrow U) to have its radial protrusions 85 and 85' forced against the leading end faces of the plungers 84 and 84', respectively. The plungers 84 and 84' and accordingly the radial protrusions 85 and 85' of the cam member 49 are thus located so that the two perpendicular components of the force exerted on the cam member 49 by the torsion spring 86 are respectively imparted from the radial protrusions 85 and 85' of the cam member 49 to the plungers 84 and 84' in the axial directions of the plungers. Thus, the cam member 49 is at all times urged to turn with respect to the rotor 55 about the center axis of the pivot pin 54 in the direction of the arrow U to enlarge the amount of eccentricity between the center axis O_c of the axial bore 50 in the cam member 49 and the center axis O_r of the rotor 55.

When a fluid pressure is directed into the control fluid chamber formed in each of the bores 83 and 83' in the pump housing 34, the plungers 84 and 84' are forced at their

leading end faces against the radial protrusions 85 and 85', respectively, of the cam member 49. If the resultant of the forces thus exerted on the radial protrusions 85 and 85' by the plungers 84 and 84', respectively, is larger than the
5 force exerted on the cam member 49 by the torsion spring 86, the cam member 49 is caused to turn about the center axis of the pivot pin 54 in a direction opposite to the direction of the arrow U with respect to the rotor 55 and produces reduction in the amount of eccentricity between the center
10 axis O_c of the axial bore 50 in the cam member 49 and the center axis O_r of the rotor 55. The reduction in the amount of eccentricity between the cam member 49 and the rotor 55 results in reduction in the volume of the pump chamber 56 between the inner peripheral surface of the cam member 49 and
15 the outer peripheral surface of the rotor 55 and accordingly in reduction in the fluid displacement rate of the pump, as is the case with the embodiment of Fig. 2.

The two displacement adjusting means 81 and 81' being provided in lieu of the single displacement adjusting means
20 59 in the embodiment of Fig. 2, each of the two displacement adjusting means 81 and 81' can be constructed with reduced dimensions as compared with the dimensions of the displacement adjusting means 59 in the embodiment of Fig. 2. For this reason and further because of the fact that there is
25 no need of providing an extra space for the accommodation of the torsion spring 86, the pump housing 34 of the embodiment of Fig. 3 can be constructed simply with the cylindrical wall

portion and end wall portions without forming any protrusion or lug portion on the cylindrical wall portion. Since, furthermore, each of the plungers 84 and 84' provided in the pump housing 34 can be constructed with reduced dimensions as compared with the piston 60 and the plunger 61 in the embodiment of Fig. 2, the pump housing 34 in the embodiment of Fig. 3 can be constructed with a reduced axial length.

Fig. 4 of the drawings shows portions of another modification of the embodiment hereinbefore described with reference to Fig. 2. In the arrangement herein shown, the cylindrical hole 53 formed in the radial protrusion 51 of the cam member 49 has a diameter larger than the diameter of the pivot pin 54 and has received therein an elastic tubular element 90 of, for example, rubber. The pivot pin 54 is thus slidably received in the axial bore in the tubular element 90 so that the cam member 49 is elastically supported by the pivot pin 54. The tubular element 90 serves not only to dampen the vibration transferred from the vaned rotor 55 to the cam member 49 but to minimize the vibrations to be transferred from the cam member 49 to the pump housing 34. The vibrations transferred to the cam member 49 being dampened, the fluctuations in the pressure of the fluid pumped from the pump chamber can be reduced significantly. The reduction in the fluctuations in the pumped fluid pressure will in turn result in reduction in the vibrations of the cam member 49 and will further contribute to the reduction of the fluctuations in the pumped fluid pressure.

It will have been understood from the foregoing description that each of the hereinbefore described embodiments of the present invention is characterized by the arrangement in which the piston 60 forming part of the

5 displacement adjustment means 59 in the embodiment of Fig. 2 or each of the plungers 84 and 84' of the displacement adjusting means 81 and 81' in the embodiment of Fig. 3 is constructed separately of the cam member 49. The piston 60 or each of the plunger 84 and 84' being thus constructed

10 separately of the cam member 49, the cam member 49 need not be provided with sealing elements between the pump housing 34 and the cam member 49 and, furthermore, no precision machining is required for the fabrication of the piston 60 or the plungers 84 and 84' and for the formation of the

15 concavity 41 in the pump housing 34 of the embodiment of Fig. 2 or the bores 83 and 83' in the pump housing 34 of the embodiment of Fig. 3. In the embodiment of Fig. 2, the piston 60 can thus be easily and reliably sealed to the internal peripheral surface of the lug portion 40 of the pump

20 housing 34 and, for this reason, the sliding friction to be produced between the piston 60 and the internal peripheral surface of the lug portion 40 can be controlled easily without giving serious consideration to the sealing performance of the piston 60. Furthermore, the pressure

25 acting area of the piston 60 can be selected without respect to the geometry of the cam member 49 since the piston 60 can be designed independently of the cam member 49. These

advantages are also achieved in the displacement adjusting means 81 and 81' of the embodiment of Fig. 3.

Because, furthermore, the cam member 49 in each of the embodiments hereinbefore described is pivotally mounted on the pivot pin 54 by means of its radial protrusion 51 salient from the cylindrical wall portion of the cam member 49 and protruding into the semicylindrical groove 38 in the pump housing 34, the axis of rocking motions of the cam member 49 is located outside the cylindrical wall portion of the cam member 49 so that the cam member 49 is permitted to oscillate through reduced central angles with respect to the pump housing 34 and to the rotor 55 when required to turn a certain distance about the center axis of the pivot pin 54. This means that the cam member 49 is faithfully responsive to forces imparted thereto from the piston 60 and the spring 64 in the embodiment of Fig. 2 or from the plungers 84 and 84' in the embodiment of Fig. 3. For this reason, not only the cam member 49 is prevented from being moved excessively with respect to the rotor 55 but the spring constant of the compression spring 64 in the embodiment of Fig. 2 or of the torsion spring 86 in the embodiment of Fig. 3 can be selected without strict consideration of the responsiveness of the cam member 49 to the forces exerted and to be exerted thereon. The axis of rocking motions of the cam member 49 being located outside the cylindrical wall portion of the cam member 49, furthermore, the plunger 61 in the embodiment of Fig. 2 or each of the plungers 84 and 84' in the embodiment

of Fig. 3 can be designed to have a relatively short maximum distance of strokes.

In each of the embodiments of Figs. 2 and 3, the pivot pin 54 is located closer to the fluid discharge port 37 than to the fluid inlet port 36 in the pump housing 34 as previously mentioned in connection with the embodiment of Fig. 2. In order to exploit the hereinbefore described advantages of the embodiments of the present invention, the cylindrical hole 53 to receive the pivot pin 54 in the radial protrusion 51 of the cam member 49 is preferably located outside a cylindrical plane defined by the outer peripheral surface of the cylindrical wall portion of the cam member 49. Furthermore, the pivot pin 54 located in conjunction with the fluid discharge port 37 as above mentioned is preferably located in such a manner that the center axis of the pivot pin 54 is located within the central angle of $360/2n$ degrees across a plane passing through the center axis of the rotor 55 and the center of arc of the fluid discharge port 37, wherein n denotes the number of the vanes 58 on the rotor 55. When the pivot pin 54 is located in this fashion with respect to the fluid discharge port 37 in the pump housing 34, the components of the fluid pressure acting on the inner peripheral surface of the cam member 49 are cancelled by each other so that the cam member 49 is subjected to no such biasing force that would otherwise be imparted to the cam member 49 by the pressurized fluid flowing from the pump chamber 56 into the fluid discharge port 37. This will

significantly facilitate the designing of the displacement
adjusting means.

CLAIMS

1. A variable-displacement sliding-vane pump comprising,
in combination,

a hollow pump housing formed with a cavity, a fluid
5 inlet port and a fluid discharge port,

a hollow cam member positioned in said cavity and
rockable in the cavity about an axis fixed with respect
to the pump housing, the cam member being formed with an
axial bore having a center axis substantially parallel with
10 the axis of rocking motions of the cam member,

a cylindrical rotor disposed in said axial bore and
rotatable in the axial bore about an axis substantially
parallel with an offset from the center axis of the bore
in the cam member, the amount of eccentricity between the
15 center axis of said bore and the axis of rotation of the
rotor being variable depending upon the angular position
of the cam member with respect to said rotor about the axis
of rocking motions of the cam member, the rotor forming
between the inner peripheral surface of said cam member
20 and the outer peripheral surface of the rotor a variable-
volume pump chamber and being formed with a plurality of
radial slots which are open radially outwardly of the rotor,

vanes respectively slidable in said slots and movable
toward and away from the inner peripheral surface of said
25 cam member,

displacement adjusting means for varying said mount of
eccentricity in response to a change in the pressure of the
fluid delivered from said pump chamber to said fluid
discharge port,

c h a r a c t e r i z e d in that

the displacement adjusting means (59,81) comprise a pressure-acting member (60,84) which is responsive to variation in the pressure of the fluid delivered from said pump chamber (56) through said fluid discharge port (37) and which is operative to drive said cam member (49) to turn with respect to said pump housing (34) and to said rotor (55) about said axis of rocking motions of the cam member (49) in a direction to reduce said amount of eccentricity, said pressure-acting member (60,84) being constructed separately of said cam member (49), and biasing means (64,86) urging said cam member (49) to turn with respect to said pump housing (34) and said rotor (55) about said axis of rocking motions of a cam member (49) in a direction to increase said amount of eccentricity.

2. A variable-displacement sliding-vane pump as set forth in claim 1,

c h a r a c t e r i z e d in that

said cam member (49) has a cylindrical wall portion defining said axial bore (50) therein and a radial protrusion (51) radially salient from said cylindrical wall portion and formed with a cylindrical hole (53) having a center axis coincident with said axis of working motions of the cam member (49) and is rockably supported on said pump housing (34) by means of an elongated pivot element (54) connected at the opposite axial ends thereof to said pump housing (34) and passed through said cylindrical hole (53) in the radial protrusion of the cam member (49) so that the

cam member (49) is rockable about the center axis of the pivot element (54).

3. A variable-displacement sliding-vane pump as set forth
5 in claim 1 or 2,
c h a r a c t e r i z e d in that
said pump housing (34) is formed with a control fluid
chamber (63,83) communicable with said fluid discharge
port (37), said pressure-acting member (60) being slidable
10 in said control fluid chamber (63) in response to variation
in the fluid pressure developed in the control fluid
chamber (63).

4. A variable-displacement sliding-vane pump as set forth
15 in claim 3,
c h a r a c t e r i z e d in that
said radial protrusion (51) of the cam member (49) consti-
tutes a first radial protrusion of the cam member (49) which
further comprises a second radial protrusion (52) radially
20 salient from said cylindrical wall portion of the cam mem-
ber (49) substantially in diametrically opposite relation-
ship to said first radial protrusion (51) across the
center axis of said axial bore (50) in the cam member
(49), said pressure-acting member (60) being engageable
25 with said second radial protrusion (52) of the cam member
(49) and operative to force the second radial protrusion
(52) to move in a direction to turn the cam member (49) in
said direction to reduce said amount of eccentricity against
the force of said biasing means (64).

5. A variable-displacement sliding-vane pump as set forth in claim 4,

characterized in that

said fluid inlet port (36) and said fluid discharge port

5 (37) extend arcuately about the axis of rotation of said rotor (55) in diametrically opposite relationship to each other across the axis of rotation (Or) of the rotor (55),

the first radial protrusion (51) of the cam member (49) being located closer to the first discharge port (37) than

10 to the fluid inlet port (36) and the second radial protrusion (52) of the cam member (49) being located closer to the fluid inlet port (36) than to the fluid discharge port (37).

15 6. A variable-displacement sliding-vane pump as set forth in any of the claims 2 to 5,

characterized in that the cylindrical hole (53) in said radial protrusion (51) of the cam member (49) is

located outside the cylindrical plane defined by the outer

20 peripheral surface of said cylindrical wall portion of the cam member (49).

7. A variable-displacement sliding-vane pump as set forth in claims 5 and 6,

25 characterized in that

the center axis of said pivot element (54) is located within the central angle of $360/2n$ degrees across a plane passing through the axis of rotation (Or) of said rotor (55) and the center of arc of said fluid discharge port

(37) wherein n is the number of said vanes (58).

8. A variable-displacement sliding-vane pump as set forth in claim 3,

5 c h a r a c t e r i z e d in that
said displacement adjusting means (81) which is arranged substantially in diametrically opposite relationship to said axis of rockable motions of said cam member (49) across the center axis (Oc) of the axial bore (50) in
10 the cam member (49), constitutes one of two such means (81,81') the other of the two displacement adjusting means (81') being arranged at an angular spacing of approx. 90 degrees from each of said first displacement adjusting means (81) and said axis of rocking motions
15 of the cam member (49).

9. A variable-displacement sliding-vane pump as set forth in claim 2,

c h a r a c t e r i z e d in that
20 said biasing means comprise a torsion spring (86) located adjacent one axial end of said cam member (49) and having a helical intermediate portion wound on said pivot element (54), an arm portion merging in one direction from the helical intermediate portion and engaged by said pump
25 housing (34) and an arm portion merging in another direction out of said helical intermediate portion and engaged by said cam member (49) for urging the cam member (49) to turn about the center axis of said pivot element (54) in

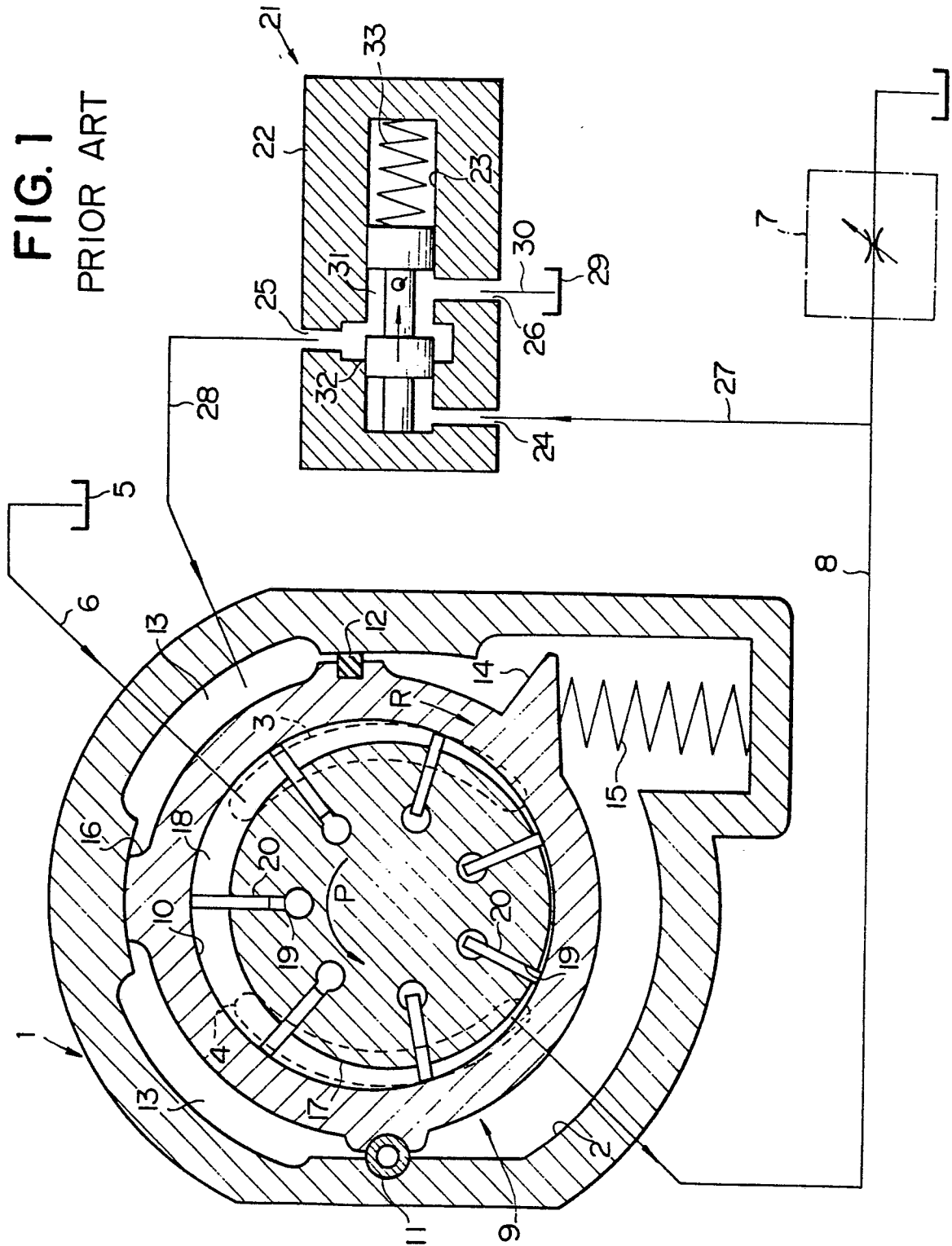
a direction to reduce said amount of eccentricity.

10. A variable-displacement sliding-vane pump as set forth
in any of the claims 2 to 9,
5 c h a r a c t e r i z e d by
an elastic tubular element (90) fitted in said cylindrical
hole (53) in said radial protrusion (51) of the cam member
(49), said pivot element (54) being axially passed through
an axial bore in said tubular element (90).

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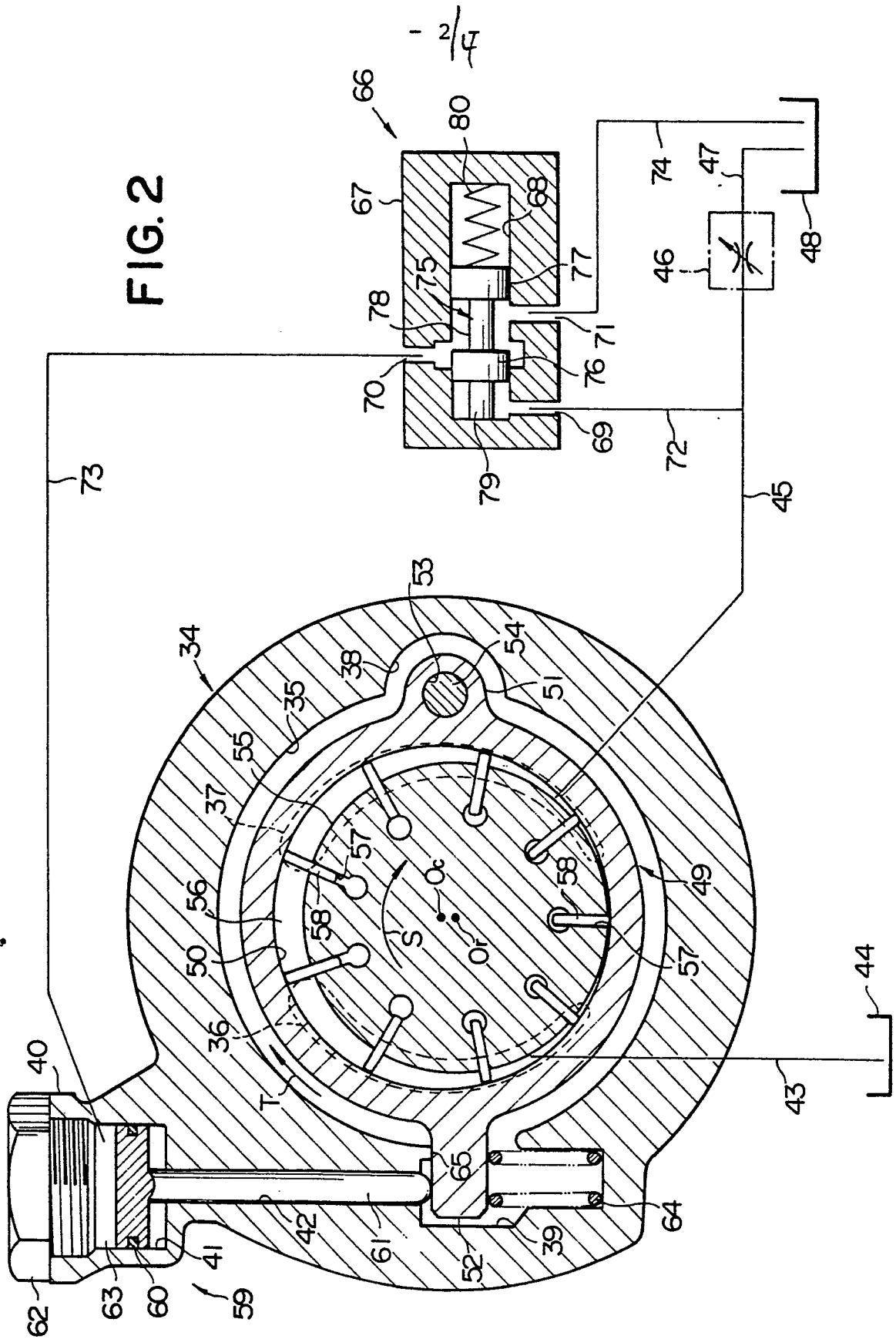
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FIG. 1
PRIOR ART



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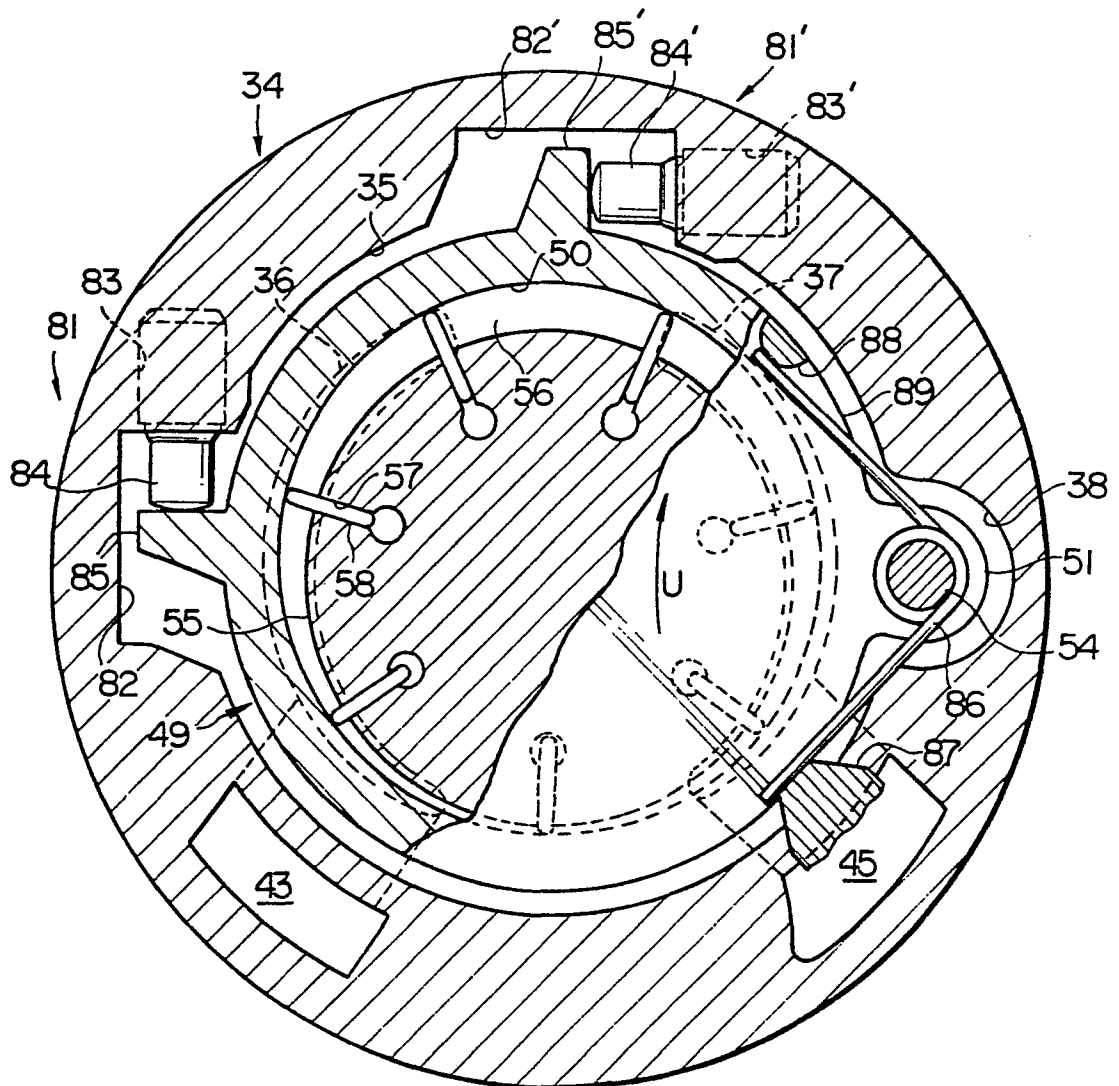
FIG. 2



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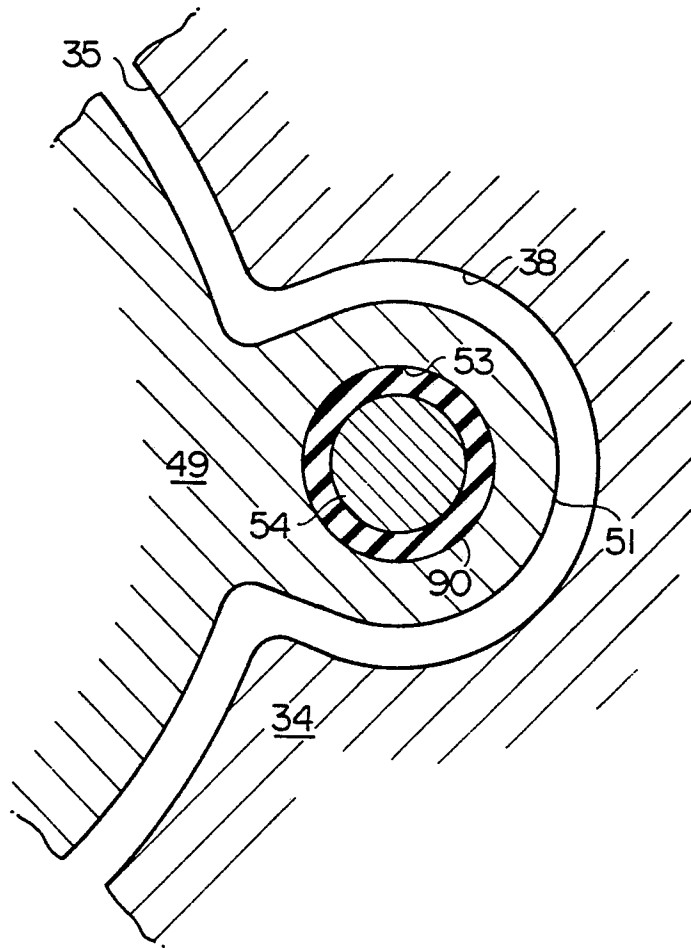
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FIG. 3



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FIG. 4





European Patent
Office

EUROPEAN SEARCH REPORT

0049838

Application number

EP 81 10 7873

DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl. 3)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
X	<u>FR - A - 981 234 (VROLIX)</u> * Page 1, left-hand column, two last paragraphs and right-hand column, two first paragraphs; figure 1 * -- <u>DE - A - 1 403 890 (PACIFIC PRESS)</u> * Page 6, last paragraph; page 7, two last paragraphs; figures 1,2; page 9, second and third paragraph; page 11, two last paragraphs; figure 3; page 12, two first paragraphs; page 19, last paragraph; page 20; page 21, first paragraph * --	1-5 1-4,6	F 04 C 15/04
A	<u>US - A - 3 334 590 (WADDINGTON)</u> * Column 1, three last paragraphs; column 2, figures 1,2,4,5 * --	1,3-6, 10	TECHNICAL FIELDS SEARCHED (Int. Cl. 3) F 04 C F 01 C
A	<u>DE - C - 513 440 (STRECKERT)</u> * Lines 31-34; 59-65; figure * ----	9	CATEGORY OF CITED DOCUMENTS X: particularly relevant if taken alone Y: particularly relevant if combined with another document of the same category A: technological background O: non-written disclosure P: intermediate document T: theory or principle underlying the invention E: earlier patent document, but published on, or after the filing date D: document cited in the application L: document cited for other reasons
☒ The present search report has been drawn up for all claims			&: member of the same patent family, corresponding document
Place of search		Date of completion of the search	Examiner
The Hague		11-01-1982	KAPOULAS