FLOW CONTROL ARRANGEMENT FOR AN AXIAL PISTON PUMP

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Filed: Feb. 15, 1973

Appl. No.: 332,749

Foreign Application Priority Data
Feb. 25, 1972 Germany................................. 2208890

U.S. Cl. ................................................. 91/499

Int. Cl. ................................................. F04b 13/04

Field of Search .................................... 91/475, 6.5, 487, 499; 417/269

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ABSTRACT
An axial piston pump has stationary and rotary control faces formed with inlet and outlet ports, and with cylinder ports, respectively, and a swash plate determining retracted and advanced dead center positions of the pistons in an axial plane. The inlet port has one end located closer to the plane in the region of the retracted dead center position than half the circumferential extension of each cylinder port so that each cylinder port communicates with the inlet port also after the respective piston has passed through the retracted dead center position and begins to move toward the advanced dead center position. In this manner, the respective cylinder is completely filled while the center of the respective cylinder port moves an angle between 5° and 20° beyond the plane of the dead centers. The other end of the inlet port may also be closer spaced from the plane of the dead center positions than half the circumferential extension of each cylinder port.

10 Claims, 5 Drawing Figures
FLOW CONTROL ARRANGEMENT FOR AN AXIAL PISTON PUMP

BACKGROUND OF THE INVENTION

The present invention relates to axial piston pumps of the type provided with a rotary cylinder body in which pistons are mounted for axial reciprocation under the control of a swash plate. In axial piston machines of this type, a stationary control face with partial circular inlet and outlet ports, cooperates with a rotary control face provided with cylinder ports which sweep the part-circular inlet and outlet ports to fill the cylinders with fluid, and to discharge the fluid again from the filled cylinders.

An axial piston pump of this type is disclosed in the U.S. Pat. No. 2,299,933. In this pump, the control ports are asymmetrically arranged to an axial plane passing through the dead center positions of the pistons. The cylinder ports are connected with the low pressure port only when the advanced dead center position has been passed by the cylinder ports, and the piston has begun to move toward the retracted dead center position. The moment at which the cylinder ports are provided with fluid from the inlet port, is delayed relative to the moment at which the respective piston is in its advanced dead center position. The separation of a cylinder port from the inlet port takes place exactly in the retracted dead center position.

The arrangement serves the purpose of preventing pressure oscillation and mechanical oscillations in the pump by permitting afterexpansion and precompression of the cylinders before the cylinder ports are provided with fluid. However, it has been found that at very high rotary speeds, using small flow cross-section, it is not possible to fill a cylinder in the available time during which each of the cylinder ports communicates with the low pressure inlet port. When the cylinders are then filled with high pressure fluid, the efficiency is reduced, noise is produced, and cavitation takes place at the control faces.

SUMMARY OF THE INVENTION

It is one object of the invention to provide an axial piston pump in which even at very high rotary speed, the cylinders are well filled due to the construction of the control faces so that upon connection of the high pressure port, the backflow into the cylinder is limited to the volume required for compressing the fluid in the cylinders to the desired pumping pressure. This result is obtained by the invention by providing an opening angle in which each of the cylinder ports which communicate with the low pressure control port, extends at least 5° beyond the angular position determined by the retracted dead center position.

This has the advantage that the cylinder chamber is already reduced before the disconnection of the low pressure control port from the already advancing piston, so that all cylinder spaces which are still empty, are filled. Furthermore, the increased opening angle has the result that an equalizing flow can take place which obtains a better filling of the cylinder space by using the kinetic energy of the fluid medium entering the cylinder chamber.

In a preferred embodiment, the radial control edges of the cylinder ports and of the low pressure port, coincide at the moment of separation, and extend over the full radial width of the respective ports in the moment of separation. This has the result that the greatest possible flow cross-section is available until the respective cylinder port separates from the low pressure inlet port.

The novel features which are considered as characteristic for the invention are set forth in particular in the appended claims. The invention itself, however, both as to its construction and its method of operation, together with additional objects and advantages thereof, will be best understood from the following description of specific embodiments when read in connection with the accompanying drawing.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a longitudinal axial sectional view illustrating an axial piston pump;
FIG. 2 is a cross-sectional view taken along line II—I in FIG. 1;
FIG. 3 is a fragmentary end view illustrating a modified construction of the control ports;
FIG. 4 is a fragmentary end view illustrating another modified construction of the control ports; and
FIG. 5 is an end view illustrating a control face according to FIG. 2, but modified.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to FIG. 1, an axial piston pump has a pot-shaped housing 1 with an interior 2 closed by a cover 3 at one end while the other end is closed by the bottom wall of the pot-shaped housing 1. Bearings 4 and 5 are mounted in the interior 2 of the housing, bearing 4 being located in the bottom region, and bearing 5 mounted in the cover 3. A shaft 6 is mounted in the bearings 4 and 5 and has one end projecting through cover 3 to the outside for connection with a drive motor, if the machine is operated as a pump.

The interior of housing 1 is sealed by sealing means 7. Shaft 6 has near bearing 5, a flange projection with peripheral grooves and recesses 9 engaging corresponding grooves and recesses in a collar portion 12 of the rotary cylinder body 10, so that cylinder body 10 is connected for rotation with shaft 6.

Cylinder body 10 has a central bore 11 in which the drive shaft 6 is located. The rotary cylinder body 10 and shaft 6 rotate about an axis to which a circular row of cylinders 14 is concentric. Each cylinder has a port 31 opening on the rotary control face 26 of the cylinder body 10, as best seen in FIG. 2.

A piston 15 is guided in each cylinder 14 for reciprocating movement, and the ends of each piston 14 are provided with a ball head 16 mounted in a spherical seat 17 of a slide shoe 18 at whose end face, a hydrostatic bearing 19 is provided which receives pressure fluid through a passage 16a in the head 16. The slide shoes 18 slidingly engage on rotation of the cylinder body, a swash plate 20 which is fixedly mounted in the housing and has a central opening through which the drive shaft 6 passes. A holding ring 21, holds the slide shoes 18 in sliding contact with the surface of the swash plate 20.

A pressure ring 22 which has a part-spherical outer surface engages with the same an inner part-spherical surface of the holding ring 21. Pressure ring 22 has a planar surface confronting the cylinder body 10, and abutting an axial face on the collar 12 of the cylinder body 10. A spring 23 is mounted in an annular chamber formed by bore 11 and shaft 6, and abuts by means of
The rotary control face 26 of cylinder body 10 is provided at the end of the cylinder body 10 remote from the swash plate 20, and slingly abuts a control face 27 of a valve plate 28 which is secured to the bottom wall of housing 1, and may be integral with the same. The control valve plate 28 and its control face 27 is provided with part-circular diametrically arranged inlet and outlet ports 29 and 30 which are connected with the inlet 129 and outlet 130, respectively.

During rotation of the rotary cylinder body 10, the fluid medium enters some of the cylinders 14 through the inlet port 29 and cylinder ports 31, and is discharged through other cylinder ports 31 and outlet port 30.

As best seen in FIG. 2, the cylinder ports 31 have the same radial width as the low pressure inlet port 29, and the high pressure outlet port 30, and are spaced the same radial distances from the axis of shaft 6. It is apparent that the cylinder port 31, shown in chain lines in FIG. 2, will, during rotation of the cylinder body 10, first sweep the end 29b and latter the end 29c of the inlet port 29. Each cylinder port 31 has inner and outer edges forming part of circles whose center is located in the axis of shaft 6, and circumferential ends 32 and 33 which are small circles having a diameter corresponding to the radial width of the control ports 29 and 30.

As shown in FIG. 2, the inlet port 29 and the outlet port 30 are diametrically arranged, but not symmetrical to the axial plane 34 in which the retracted dead center position 34a and the advanced dead center position 34b are located. The angular region in which each of the cylinder ports 31 is connected with the low pressure port 29, is displaced between 5° and 20° beyond the retracted dead center position 34a. The cylinder port 31 is shown in chain lines in FIG. 2 in the position in which the communication between the cylinder port and the low pressure inlet port 29 is interrupted. In this position, the center of the cylinder port 33 is not located in the plane 34, but angularly spaced therefrom, and when the cylinder port 33 was located with a center in the plane 34, the control edges 32 and 29c overlapped to establish communication which is maintained until the position shown in FIG. 2 is reached during further clockwise rotation.

The end and control edge 29a is located closer to the plane 34 in the region of the retracted dead center position 34a than half the circumferential extension of each cylinder port 31, so that each cylinder port 31 communicates with the end 29c of the inlet port 29 also after the respective piston has passed through the retracted dead center position 34a and begins to move toward the advanced dead center position 34b. In this manner, the respective cylinder 15 is completely filled while the center of the respective cylinder port 34 moves a predetermined angle A beyond the plane 34.

Due to the angular displacement of the low pressure inlet port 29 relative to the plane of symmetry 34, the cylinder space 14 is connected by cylinder port 31 so long with the low pressure side of the pump until piston 15 has moved beyond the retracted dead center position an angle between 5° and 20°, and has started its discharging motion. The result is that insufficiently filled empty spaces in cylinder 14 are filled by fluid medium displaced by the pistons after moving through plane 34. Furthermore, a time period remains in which the kinetic energy of the fluid flowing from the low pressure inlet port into the pump can be used for better filling the cylinders.

The arrangement according to the invention described above considerably improves the suction efficiency of the pump, assures an extension of the span of life of the machine by avoiding cavitation damage, and has a favorable influence on the suppression of noise.

In accordance with the further improvement of the invention shown in FIG. 3, the cylinder ports 31 have control edges 32 which coincide with the control edge 29a of the control port 29, along the entire radial width of the control ports 29 and 31. The control edge 32 is part circular and inward concave, and the control edge 29c of the low pressure port 29 is convex, and has the same curvature so that when cylinder port 31 turns further to the right during rotation in clockwise direction, the control edges 32 and 29c fully coincide, and then sudden separation of the communication between the cylinder port 31 and the low pressure port 29 takes place. Shortly before the interruption of the flow through control ports 29, 31, the entire cross section of low pressure control port 29 is available, contrary to the arrangement shown in FIG. 2 in which the flow cross section of both control ports 31 and 29 are reduced before separation takes place communication between control ports 29 and 31 is cut off.

The large flow cross section until the moment of separation takes place, is favorable for the filling of the cylinders, since it has been found that irrespective of low piston speeds, the flow speed and thereby also the suction resistance are great in the construction of the control port 31, 29 shown in FIG. 2.

FIG. 4 illustrates another modification in which the cylinder port 31a has a straight radially extending control edge 32′, cooperating with a straight radially extending control edge 29a′ of the low pressure port 29′. The control edges 29a′ and 32′ coincide at the moment of separation of cylinder port 31′ from low pressure port 29′, so that the complete flow cross section is available for communication before this moment.

FIG. 5 shows a modification of FIG. 2, the cooperation between the cylinder ports 31 and the low pressure port 29 being the same as in FIG. 2. It will be noted that the end 49a of the inlet port 49 is more closely adjacent the plane 34 in the region of the retracted dead center position than half the circumferential width of the cylinder port 31, not shown in FIG. 5 at the end 49a. The ends 50a and 50b of the high pressure ports 50 are spaced the same circumferential distances from the plane 34, but the end 49b of the inlet port 49 is located closer to the plane 34 in the region of the advanced dead center position 34b. The angle in which the cylinder port 31 communicates with the low pressure port 49 is increased, and the moment of communication between the cylinder port 31 and the end 49b of the low pressure port 49 starts about 5° earlier. The end 50b of the high pressure port 50 is correspondingly shortened. Due to the extension of the end 49b toward the plane 34, the angular region in which the cylinder ports communicate with the low pressure port is substantially increased as compared with the arrangement of FIG. 2. The extension of the end 49b toward the plane 34 in the region of the advanced dead center position obtains at
the low inlet pressures in the region of high rotary speeds, good filling of the cylinders 14, although the acceleration rate cannot be fully obtained in the region of the highest piston speed.

Evidently, the constructions of the control edges shown in FIG. 3 and FIG. 4 can be applied to the arrangement of FIG. 5.

It will be understood that each of the elements described above, or two or more together, may also find a useful application in other types of flow control arrangement for an axial piston pump differing from the types described above.

While the invention has been illustrated and described as embodied in an arrangement in which the low pressure inlet port of an axial piston machine is extended toward the plane in which the dead center positions of the pistons are located in order to obtain better filling of the cylinder chambers, it is not intended to be limited to the details shown, since various modifications and structural changes may be made without departing in any way from the spirit of the present invention.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others can by applying current knowledge readily adapt it for various applications without omitting features that, from the standpoint of prior art, fairly constitute essential characteristics of the generic or specific aspects of this invention and, therefore, such adaptations should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be protected by Letters Patent is set forth in the appended claims.

We claim:

1. Flow control arrangement for an axial piston pump, comprising cylinder body means having an axis, cylinders arranged in a circle about said axis, and pistons in said cylinders; housing means including a swash plate engaged by said pistons, one of said means being rotatable relative to the other means about said axis so that each piston is axially advanced and retracted by said swash plate between advanced and retracted dead center positions located in an axial plane; said cylinder body means having a first control face formed with a circular row of cylinder ports communicating with said cylinders, respectively, and said housing means having a second control face formed with a part-circular continuous inlet port and a part-circular continuous outlet port, respectively located on opposite sides of said plane, said inlet and said outlet port being the only ports formed in said second control face, said first and said second control faces cooperating during rotation so that said inlet and outlet ports are swept by said cylinder ports, said inlet port being longer than said outlet port and having one end located closer to said plane in the region of said retracted dead center position than half of the circumferential extension of each cylinder port so that each cylinder port communicates with said one end of said inlet port also after the respective piston has passed through said retracted dead center position and begins to move towards said advanced dead center position whereby the respective cylinder is completely filled while the center of the respective cylinder port moves a predetermined angle beyond said plane.

2. Flow control arrangement as claimed in claim 1 wherein said predetermined angle is between 5° and 20°.

3. Flow control arrangement as claimed in claim 1 wherein said inlet port has the same radius and radial width as said cylinder ports; wherein said inlet port has at said one end a control edge extending across the full radial width of said one end of said inlet port; wherein each cylinder port has a trailing control edge substantially coinciding with said control edge over the radial width of said one end of said inlet port when the respective cylinder port has moved through said predetermined angle and separates from said inlet port.

4. Flow control arrangement as claimed in claim 3 wherein said control edge at said one end of said inlet port is convex, and said trailing control edges of said cylinder ports are concave and coincide with said control edge of said inlet port when the respective cylinder port separates from said inlet port.

5. Flow control arrangement as claimed in claim 3 wherein said predetermined angle is between 5° and 20°.

6. Flow control arrangement as claimed in claim 3 wherein said control edge at said one end of said inlet port is straight and extends in radial direction.

7. Flow control arrangement as claimed in claim 4 wherein said trailing control edges of said cylinder ports are straight and extend in radial direction.

8. Flow control arrangement as claimed in claim 1 wherein said inlet port has an other end located closer to said plane in the region of said advanced dead center position than half the circumferential extension of each cylinder port so that each cylinder port communicates with said other end of said inlet port while moving an other predetermined angle before the respective piston passes through said advanced dead center position.

9. Flow control arrangement as claimed in claim 8 wherein said other predetermined angle is between 5° and 10°.

10. Flow control arrangement as claimed in claim 9 wherein said predetermined angle in the region of said retracted dead center position is between 5° and 20°.

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