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GB 1244640
GB 1108434
GB 1069671
GB 0721559
GB 0542334
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(54) Axial piston pumps

(57) In order to simply, manufacture, improve sealing and fluid flow and reduce fluid pressure loadings, an axial piston pump for high viscosity fluid eg grease comprises a rotatable cylinder block 6 in which cylinders of constant cross section converge from one end face towards the other 24. A ported member 35 permits suction of fluid into the cylinders from inlet 29 and discharge into outlet passage 30. The rotational axis of the cylinder block 6 is offset from the axis of outlet 30 and the fluid pressure differential across it urges the member 35 against end face 24 of the block 6 leakage passed the pistons is collected and used for lubrication (see Fig. 3 not shown).

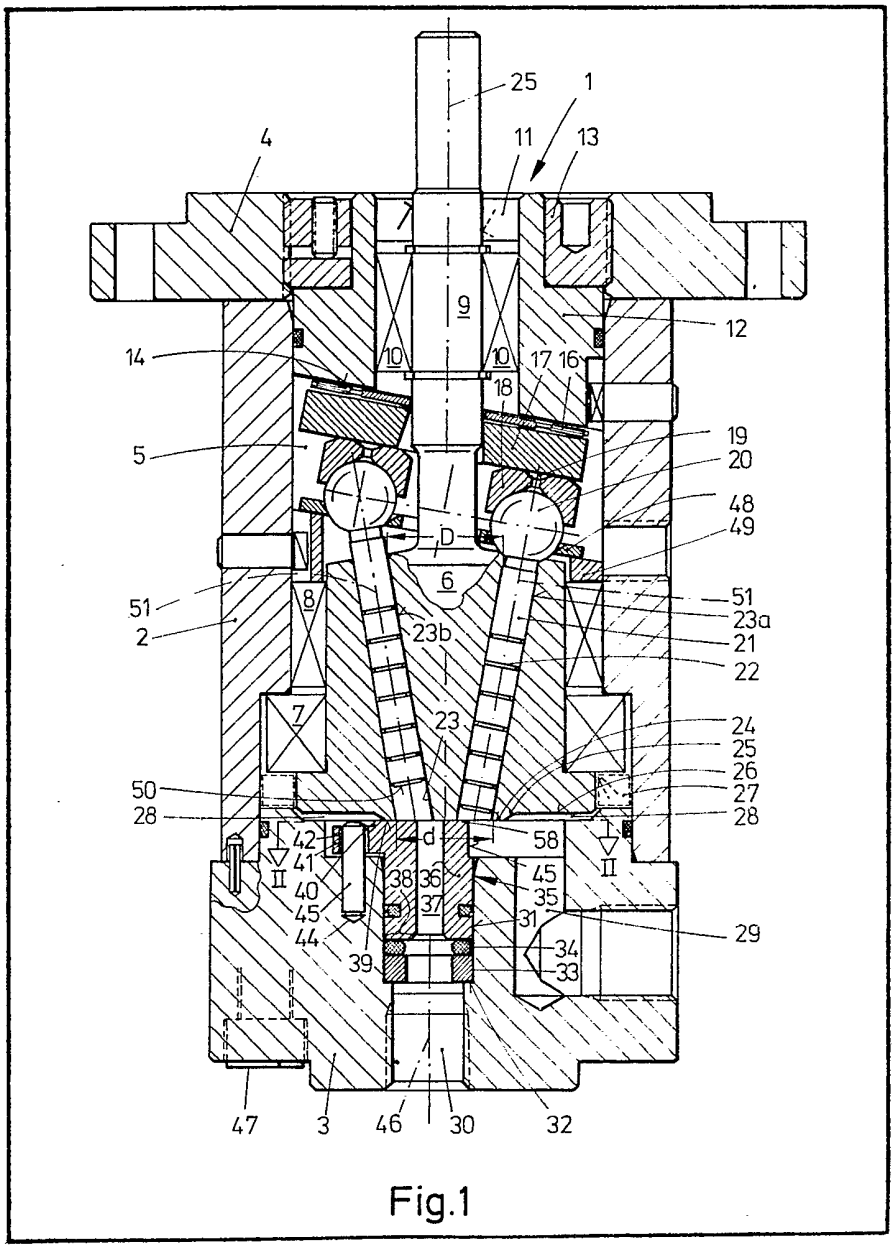

Fig.1

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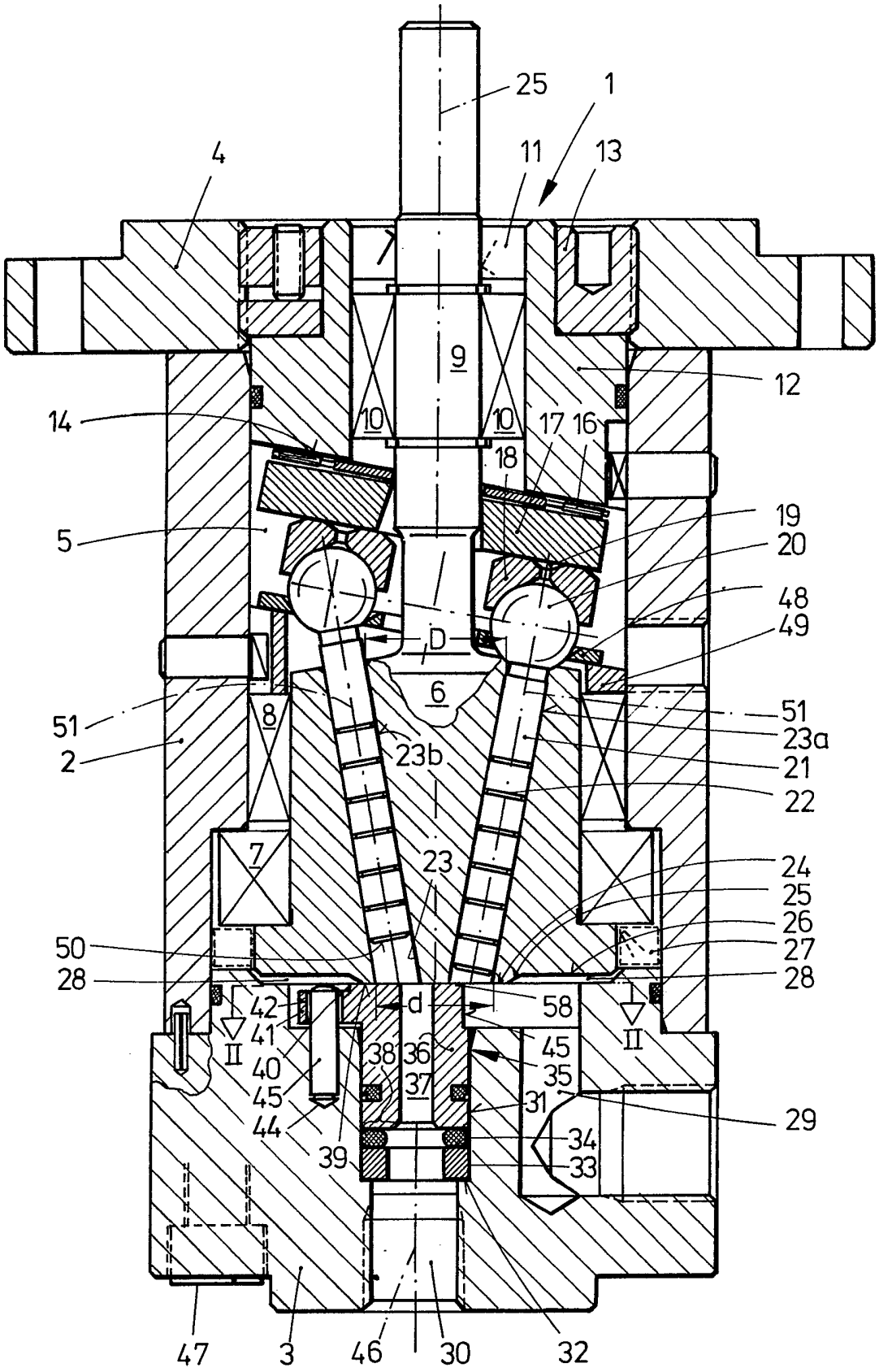
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Fig.1

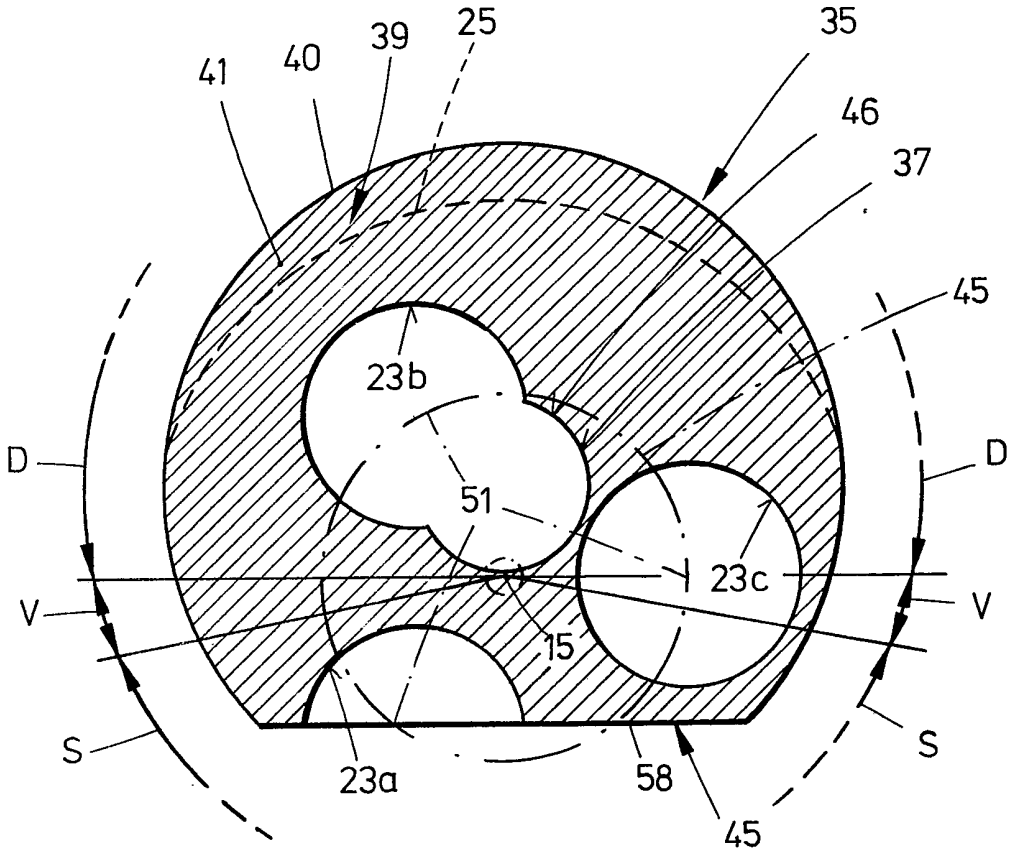


Fig. 2

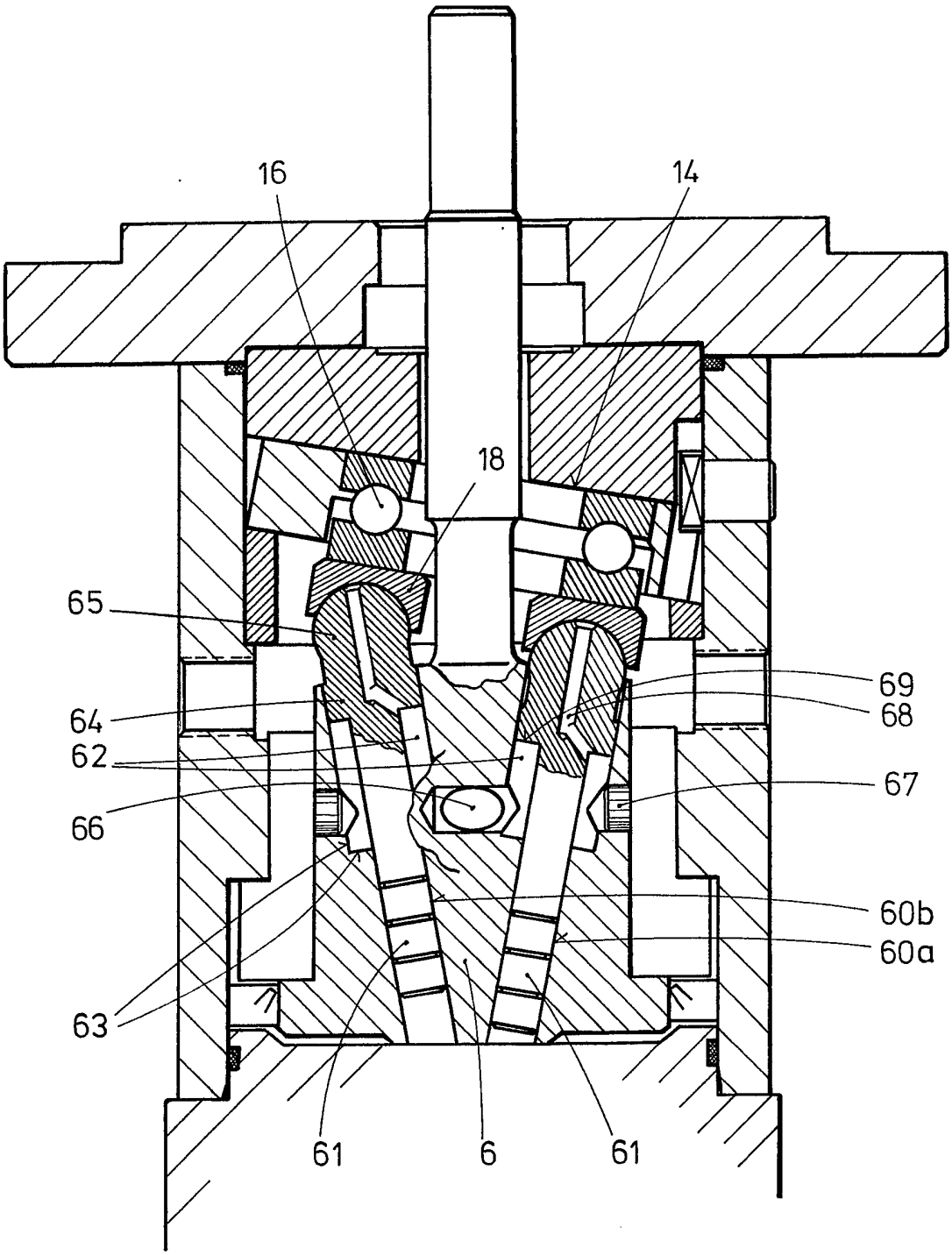


Fig. 3

Fig.4

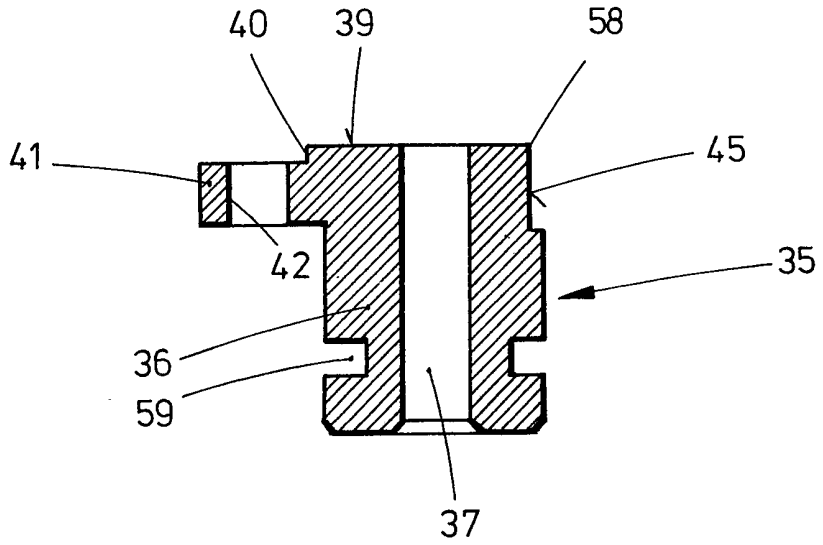
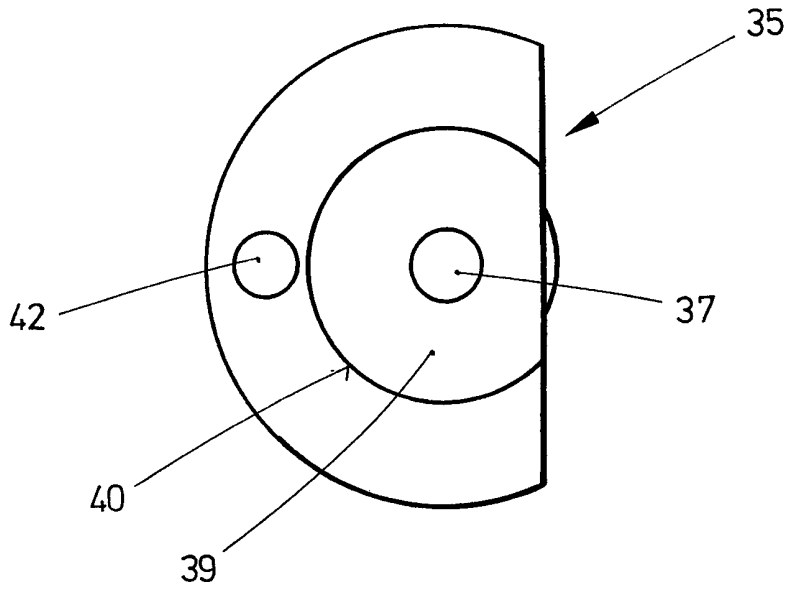


Fig.5



SPECIFICATION

Piston pumps

The invention relates to an axial piston pump of the type indicated in the preamble to the Main Claim.

Already known from the magazine "Olhydraulik und Pneumatik", 7/1975, page 489, is an axial piston pump in which the cylinder bores slightly converge in the direction of the control face of the rotor. From the ends of the cylinder bores, then, more sharply converging and narrowed communicating passages extend to the concave control face which rests on a convex mating control face of a mussel-like control part. The control part comprises kidney-shaped control orifices, pressure and suction passages. The constrictions in and reversal points in the communicating passages adversely affect not only the flow of the medium but also form considerable dead spaces which cannot be utilised for the suction and pressure strokes of the piston. The area of contact between the control face and the mating control face extends quite widely in a radial direction from the axis of rotation of the rotor. As a result of the considerable radial distances from the axis of rotation, high relative speeds occur between the contacting faces and the intersecting mouths between communicating passages and the control apertures and high loadings for the medium which is being conveyed. Furthermore, an asymmetrical load distribution results between the control face and the mating control face and increasing leakage losses with increasing working pressure. Manufacture of the control aperture is difficult, as is also the curved form of the surfaces.

In the case of axial piston pumps in which the cylinder bores lie exactly parallel with the axis of rotation of the rotor, these disadvantages are even more serious, so that such axial piston pumps cannot as a rule be used for conveying lubricating grease or high viscosity oil.

The invention is based on the problem of making an axial piston pump of the type mentioned at the outset usable for conveying also high viscosity media, e.g. grease, and of making such a pump substantially more simple structurally, avoiding disadvantageous reversal points, constrictions or dead spaces in the flow of medium, and making the faces (control face and mating control face) which are movable in relation to one another as small as possible in order to achieve low friction losses and low surface and bearing loadings.

According to the invention, the problem posed is resolved by the features indicated in the characterising part of the Main Claim.

Since the full cross-section of the cylinder bore extends into the control face, there are no constrictions or reversal points. There are virtually no more dead spaces since the piston works directly up to the control face. As a result of the marked oblique position of the cylinder bore, the mouth is situated very close to the axis of rotation so that the necessary area of contact between the control and the mating control faces can be kept optimally small. Pressure forces required to ensure a seal between these faces act ideally independently of the working pressure in terms of magnitude. The loading for the flow of medium being conveyed is also low since the intersecting mouths lie at a small distance from the axis of rotation of the rotor. For practical purposes, there is only a radial shift of the flow of conveying medium out of the suction passage into the pressure passage. As a result of the elimination of communicating passages, the rotor becomes very short in overall length so that the overall length of the axial piston pump is also shortened.

Its individual components are of very simple construction and can be economically produced. Thus, for the first time, an axial piston pump can be used for conveying grease or high viscosity oil, working pressures of 350 bars or more being feasible.

With an eye to simplified manufacture and functionally more reliable inter action between the control face and the mating control face, the measures are proposed which are indicated in Claim 2. The control sleeve is a component which can be easily manufactured and which can with minimal industrial complication be easily and finely precision machined on those parts of its surface area which are important to control of the axial piston pump.

In connection with this, the measures explained in Claim 3 are also important. Thus, the control sleeve is a simple rotary part with an end flange which, for construction of the control edge, is provided with a milled or cut-off flattened portion.

An expedient embodiment of the invention is referred to in Claim 4. If there are an odd number of pistons, a slight degree of non-uniformity of flow can be achieved.

Claim 5 explains a further important measure. This embodiment takes into account the demand for the smallest possible mating control face. Furthermore, with this arrangement, the pressure passage may be virtually a straight extension of whichever cylinder bore connects with it so that the pressurised medium can be ejected against low resistance. The relative speed which arises between the mouth of the pressure passage and the mouth of whichever cylinder bore is connected to it is very low which results in a minor squeezing loading of the medium being conveyed. Since the mutually contacting faces pass each other in this area at only a low relative speed, only negligible friction losses occur.

An axial piston pump according to the invention is, at the control end, simple in design if it has the features according to Claim 6. The suction passage necessitates no particular manufacturing conditions but is simply defined by the housing and the end face of the control sleeve or is constituted by the entire space in the housing surrounding the control face. The control edge has thereby the task of controlling the

induction via that portion of the rotary movement of each cylinder bore through which the relevant piston is in the process of performing its induction stroke.

The securing of the control sleeve against rotation while at the same time ensuring its axial mobility can be achieved particularly easily in the manner indicated in Claim 7. In practice, it has been proven successful, according to Claim 8, to use a fitting bolt which secures the control sleeve against rotation and which fixes it in the predetermined location in the housing and in relation to the rotor. A positive secondary effect is the automatic readjustment capability of the control sleeve in the event of wear between the control face and the mating control face.

Where axial piston pumps are concerned, the sealing of passages which convey medium into the interior of the housing is particularly important, care having at the same time to be taken to see that the pressure applied between the control face and the mating control face must not result in seizure of the contacting faces. This object is achieved with minimum technical complication by the features of Claim 9. The spring element presses the control sleeve against the control face with a preselected degree of initial tension.

A particularly important idea underlying the invention furthermore emerges from Claim 10. This matching of the surfaces which are exposed to the pressure of the medium being conveyed makes it possible to adjust an intentional additional pressure of application of the control sleeve against the rotor, the force of application being increased with increasing working pressure, a factor which is particularly expedient, so that despite rising working pressure, it is possible to maintain a constantly good sealing effect.

A further important feature of the invention is indicated in Claim 11. In the case of axial piston pumps of the type concerned here, the pistons work in the cylinder bores with leakage losses which can be used for lubricating the pivot bearings of the rotor and of the drive used for the pistons, if the medium being conveyed is a grease or a high viscosity oil.

Pursuing this thought, an embodiment such as emerges from Claim 12 is also expedient. In this way, the medium which leaks through becomes at the same time a working medium with which each piston can be pressed against the working surface, so that it becomes possible to save on additional means which are normally provided for the purpose.

According to Claim 13, the medium which is leaking through can, in addition to being used to press the pistons against the working surface, also be used for lubricating the heavily loaded bearings between the piston ends and the ball sockets.

Finally, the measure according to Claim 14 is also important since then the grease which is to be conveyed does not find its way into the transmission space but is deflected directly out of the open suction passage.

Examples of the invention will be described in detail hereinafter with reference to the accompanying drawings, in which:—

Fig. 1 is an axial section through a first embodiment of axial piston pump;

Fig. 2 is a section taken on the line II—II in Fig. 1;

Fig. 3 is a part of an axial section through a further embodiment of axial piston pump according to the invention;

Fig. 4 is an axial section through a detail of Fig. 1, and

Fig. 5 is a plan view of the detail in Fig. 4.

According to Fig. 1, a first embodiment of axial piston pump 1 comprises a central housing part 2 in the bottom of which is inserted a housing part 3 which is fixed by means of fixing screws 47, while the top end of the housing part 2 is covered with a cover plate 4. In this way, there is defined inside the housing a drive chamber 5 in which there is, rotatably mounted in thrust bearings 7, 8 a rotor 6 which is cylindrical in outline. From the rotor 6, a drive shaft 9 extends out of the housing and is sealed on the outside by a gasket 11. The mounting 10 is braced on an insert part 12 which is held by a screwed-in retaining ring 13 and which carries at the inner end a working surface 14 which is oblique in relation to the axis of rotation 15 of the shaft 9 and of the rotor 6.

Rolling bearings 16 rotatably mount on the annular body 17 on the working surface 14; set upon the annular body 17 are bearing cups 18 with lubricating bores 19 in which pivot balls 20 are seated. The pivot balls 20 are held in the cups by braces 48, 49.

The pivot balls 20 are provided at the upper ends of working pistons 21 which are guided for sealed displacement in cylinder bores 23 through peripheral sealing slots 22. Offset by in each case 120° in the peripheral direction there are in the rotor three cylinder bores 23a, b and c, of which the cylinder bores 23a, b are shown as displaced into the plane in which the section is drawn. The cylinder axes of the cylinder bores are designated 51. Provided at that end of the rotor 6 which is remote from the working surface 14 is a circular control face 24. The cylinder bores 23a—c converge in the direction of the control face 24 so sharply that the points at which they emerge at the control face 24 lie in a circle (diameter d) which is smaller than a circle of diameter D which can be circumscribed at that end of the rotor which is towards the working surface, in the openings to the cylinder bores which are located on that surface.

The control face 24 is surrounded by a set back circular face 26 through a circular shoulder 25.

A part of the control face 24 and the circular annular face 26 bound a suction passage 28 from which a bore 29 extends out of the housing. The suction passage 28 is separated from the interior 5 of the housing by an annular gasket 27.

Formed in the housing part 3 and parallel with the axis of rotation 15 but offset eccentrically in

relation thereto is a bore 30 having axis 46. The bore 30 widens via a shoulder 32 to form an enlarged-diameter guide bore 31 leading to the suction passage 28.

Seated on the shoulder 32 is an abutment ring 33 on which there is an O-ring 34 which provides a spring loading on a control sleeve 35 in the direction of the control face 24. The control sleeve 35 has a cylindrical stem 36 by which it is sealed and displaceably guided in the bore 31. Adjacent to the stem 36 is an enlarged-diameter annular flange 41 in which, in a fitting bore 42, there engages a fitting key 43 which is a sliding fit in a blind bore 44 in the housing part 3. The fitting key 43 secures the control sleeve 35 against rotation about its axis 46; it does however permit its axial mobility in the direction of the axis 46. On the side opposite the fitting key 43 there is provided in the annular flange 41 a flattened part 45 which forms a control edge.

In a continuation of the bore 46, the control sleeve 36 is traversed by a pressure passage 37, the cross-sectional area of which is slightly smaller than the cross-sectional area of each cylinder bore 23a—c. The pressure passage 37 emerges directly in a mating control face 39 on the end face of the control sleeve 35 which, under the action of the O-ring 34, is pressed against the control face 24 of the rotor 6. The mating control face is defined by an encircling step 40.

The annular face 38 of the stem 36 of the control sleeve 35 which is subject to the pressure of the medium in the bore 46 and the pressure passage 37 is so dimensioned that it is slightly greater than the portion of the surface of the mating control face 39 on which the working pressure and a partial pressure arising in the lubricating gap between the faces provides a loading. In this way, in addition to the initial tensioning force of the O-ring 34 (or of an equivalent thrust spring), it is possible to derive from the difference between the effective surfaces a force at which, as a function of the particular working pressure at any given time, the axial piston pump presses the control sleeve 35 against the control face 24. Then, with increasing working pressure, a firmer application and even sealing effect can be achieved. The axial piston pump according to Fig. 1 operates as follows:—

Upon a rotation of the rotor 6, and at every say 360° revolution, the pistons 21 are pushed once into the cylinder bore and extracted once therefrom so that they perform a pressure and a suction stroke. The control edge 58 is thereby and in relation to the working surface 14, so disposed that the mouth of each cylinder bore 23 is exposed when the piston located in the cylinder bore is performing its suction stroke. In doing so, it draws in medium until it has reached its top dead centre position. Afterwards, the mouth of the cylinder bore in the control face 24 cuts across the control edge and moves downwards until such time as it meets the pressure passage 37. During the subsequent rotary movement of the rotor, the piston performs its pressure stroke and pushes the indrawn quantity of medium into the pressure passage 37. Naturally, the size of opening for the medium being conveyed will vary via the angle of rotation associated with the suction and pressure stroke. It is intended to explain these kinetics hereinafter with reference to Fig. 2.

Fig. 2 shows the mating control face 39 in solid lines. The control edge 58 extends in a straight line and separates from the full circle a segment of a circle. The pressure passage 37 contains the longitudinal axis 46 of the control sleeve 35 and its outer periphery is adjacent the axis of rotation 15 of the rotor. The mouths of the cylinder bores 23a—c indicated by full circles have their central axes 51 which rotate around the axis of rotation 15 on a circle U_{51}. The broken line 25 denotes the annular shoulder 25 of the rotor, which bounds the control face 24. In the position shown in the drawing, there is thus a contacting face between the mating control face 39 and the control face 24, the size of which is defined by the contour of the mating control face 39, except for the sickle shaped portion outside the broken line 35. After deducting the cross-section of the pressure passage 37 and of the mouths of the cylinder bores 23a—23c located above the mating control face 39, it is possible to determine the area of contact between the control face 24 and the mating control face 39 in which friction occurs and which is used for outwards sealing of the pressurised passages 37, 23b. It is clearly recognisable that particularly the pushing out of medium through the piston which at any given time intersects the pressure passage 37 or the cylinder bore mouth (in this case 23b) which is associated with the said piston occurs very close to the axis of rotation 15 so that there the relative rotary speed is low and even at high pressure it is possible to achieve perfect sealing. The reaction forces of the medium which is pressurised act likewise very close to the axis of rotation 15 so that they can be easily absorbed by the rotor mountings.

On condition that in Fig. 2 the direction of rotation of the rotor is clockwise, the cylinder bore 23a has almost completed its suction stroke, since it has already travelled over the control edge 58 by approximately half its cross-section. The piston in the cylinder bore 23b is still its pressure stroke and forces medium into the pressure passage 37. The maximum overlap between cylinder bore 22b and the pressure passage 37 is still not reached. The mouth of the cylinder bore 23c in which the piston is just short of its bottom dead centre position, it completely closed by the mating control face 39. There is no overlap with either the pressure passage 37 or the suction passage 28.

Upon further rotation of the rotor 6, firstly the mouth of the cylinder bore 23a is completely closed by the mating control face 39 before it arrives at the pressure passage 37. The mouth of the cylinder bore 23b is then still connected to the pressure passage 37, so that here there is an

overlap since then the pistons in the cylinder bores 23a and 23b simultaneously deliver into the pressure passage 37. The mouth of the cylinder bore 23c then has at least partially already travelled over the control edge 58 so that the piston disposed therein can draw in the medium which is to be conveyed. Related to the central axis 51 of each cylinder bore 23, the mouth thereof so co-operates with the mating control face 39 during a 360° rotation that there is at any given time a narrow sector V between the suction and the pressure stroke or vice versa in which the mouth is completely masked (V). Over an approximate range of 180° D, the pressure stroke occurs during which the mouth is in flowing communication with the pressure passage 37. Over an arc S of less than 180°, finally, the medium being conveyed is drawn in. By corresponding matching of the cross-sections of the cylinder bores, of the pressure passage and the construction of the location of the control edge (it might have a form other then a straight line), these control characteristics can be varied as desired.

Fig. 3 shows part of a further embodiment of axial piston pump in longitudinal section. Since with this embodiment the bottom part of the housing generally corresponds substantially to that in Fig. 1, there is no need to explain it in greater detail. In contrast, a difference is the manner in which the pistons are driven.

In the rotor 6, three pistons 61 are displaceably and in sealing-tight fashion guided in the cylinder bores 60a—60c, of which only cylinder bores 60a and 60b are shown in the plane of the sectional drawing. Close to the upper end of each cylinder bore 60a to 60c is an enlarged diameter (stepped bore 63) collecting chamber 62, all collecting chambers 62 being connected to one another in pressure-compensating fashion via communicating passages 66. Closure plugs 67 occlude the manufacture-necessitated mouths of the communicating passage 66 towards the outside ambient. Each piston 61 has at its end which is towards the working surface 14 a head 64 which is of thickened diameter and which has an annular shoulder 69 projecting into the collecting chamber 62 and being displaceably and in sealing-tight fashion guided therein. Also provided in the head 64 is a lubricating passage 68 which extends from the collecting chamber 62 to the head end 65 disposed in the cup 18. Conveyed medium which necessarily leaks through along the piston periphery collects in the chamber 62 and builds up a certain pressure. This pressure acts on the annular face 69 and pressed the piston head 64 with their ball faces into the cups 18 and presses these latter against the bearings 16, so that there is no need for any special means to hold the pistons on the working surface 14. The lifting movement of each piston is not influenced by the pressure in the collecting chamber since the collecting chambers communicate with one another and during rotation of the rotor the quantity of medium contained in all the collecting chambers remains constant. The pressure in the collecting chambers is furthermore utilised in order to employ the medium for lubricating the ball faces and cups 18, which is particularly advantageous if the medium being conveyed is a grease or a high viscosity oil.

The control sleeve 35 can be clearly seen in Figs. 4 and 5. It is a simple rotary part with annular flange 41 which has on one side a recess 42 while on the other side it has a flattened portion 45 which forms the control edge 58. The pressure passage 37 is a smooth bore. Provided in the stem 36 there is also an encircling slot 59 which accommodates a further sealing element which can be seen in Fig. 1. Furthermore, additional oil control grooves or slots can be provided on the outside of the stem 36. The particular advantage of this control part which is required to control the pump function, namely the control sleeve 35, lies in the benefit of simple and competitive manufacture. Particularly fine surface machining is required only for the mating control face 39 which co-operates with the control face 24, which must likewise be finely machined, on the rotor. In comparison with the concavely or convexly shaped control and mating control faces in the prior art, which must similarly be finely machined, extraordinary advantages are provided here from the manufacturing point of view.

Claims

1. Axial piston pump for conveying a hydraulic medium, particularly for grease or high viscosity oil, and having a rotor which can be driven to rotate in a housing and which has at the end a control face and which, divided in the peripheral direction, has at least one cylinder bore located obliquely of the axis of rotation and in the direction of the control face, for a piston which is displaceable in the bore and which, at the end remote from the control face, is supported on a fixed working surface which is inclined in relation to the axis of rotation, and having furthermore, extending out of the cylinder bore and towards the control face, a connecting passage and having, bearing on the control face and non-rotatably disposed in the housing, a mating control face at which terminate suction and pressure passages with which the communicating passage is alternately in flowing communication, characterised in that the cylinder bore (23a—23c; 60a—60c) extends with unvaried diameter as far as the control face (24) forming thereby at the same time a communicating passage, and in that the cylinder bore extends so sharply obliquely to the axis of rotation (15) that its mouth lies in the central portion of the control face and within a circle, the diameter (d) of which is smaller than the diameter (d) of a circle which can be circumscribed at the opposite end of the rotor (6) within the other cylinder bore mouth, and in that the mating control face (39) is constructed on a control sleeve which can be pressed against the control face (24) with a force which is expediently

dependent upon the working pressure in the pressure passage (37, 46).

2. Axial piston pump according to Claim 1, characterised in that the mating control face (39) is the end face of the control sleeve (35) which is rotationally rigidly disposed in the housing (3) eccentrically of the axis of rotation (15) and which has a central bore forming the pressure passage (37) and on the outside a segment-like flattened portion (45) which forms the control edge (58).

3. Axial piston pump according to one of Claims 1 or 2, characterised in that the control sleeve (35) has a cylindrical stem (36) and a circular end flange (41) on the side of which is located the flattened portion (45).

4. Axial piston pump according to Claim 1, characterised in that an odd number of converging cylinder bores (23a—23c, 60a—60c) which are staggered evenly in a peripheral direction are provided in the rotor (6).

5. Axial piston pump according to one of Claims 1 to 4, characterised in that the pressure passage connection (37) is disposed in a central zone of the mating control face (39) and borders on or comprises the axis of rotation (15) of the rotor (6).

6. Axial piston pump according to one of Claims 1 to 5, characterised in that the suction passage (28) is bounded by the circumference of the end flange (41) and the inner wall of the housing (in the housing part 3).

7. Axial piston pump according to Claim 3, characterised in that a raised part disposed in the housing engages the annular flange (41) to secure the control sleeve (35) against rotation.

8. Axial piston pump according to Claim 7, characterised in that the raised portion is, rigid with the housing, a fitting bolt (43) which is a sliding fit in a bore (42) in the annular flange (41).

9. Axial piston pump according to one of Claims 1 to 8, characterised in that the control sleeve (35) has its stem (36) mounted for sealing-tight displacement in a housing bore (31) and — in the direction opposite to the rotor (6) — is braced on an abutment (32, 33) through an interposed spring element (34).

10. Axial piston pump according to one of Claims 1 to 9, characterised in that the annular face (38) of the stem (36) which is subject to the pressure in the pressure passage (37) corresponds approximately to the average proportion of the area of the mating control face (39) on which, during rotation of the rotor (6), the working pressure emerging from the cylinder bores and a partial pressure arising from the lubricating gap between the faces acts, so that the control sleeve (35) is in an axial direction subject to a force which is directed towards the rotor and which is directly proportional to the working pressure.

11. Axial piston pump according to one of Claims 1 to 10, characterised in that provided in each cylinder bore (60a—c) is a collecting chamber (62) for a medium leaking through to the end of the piston (61) and from which lubricating passages extend to the pivot bearings of the rotor and/or the upper support of the pistons.

12. Axial piston pump according to Claim 11, characterised in that each cylinder bore (60b) is widened out in its diameter (step bore 63) at its end which is towards the working surface (14) in order to provide the collecting chamber (62), and in that the piston (61) guided in the cylinder bore has a head (64) which is of thicker diameter and which is guided in sealing-tight fashion in the collecting chamber and in that the collecting chambers (62) of all cylinder bores are in flowing communication with one another through alternative channels (66) in the rotor (6).

13. Axial piston pump according to Claim 12, characterised in that in the head (64) of each piston (61) there is lubricating passage (68) which has a constriction in it, the passage starting at the collecting chamber (62) and extending as far as the free head end which engages into a ball socket.

14. Axial piston pump according to one of Claims 1 to 13, characterised in that the housing part (3) containing the suction passage (28) is separated from transmission housing space (5) by a seal (27).

15. Axial piston pump substantially as hereinbefore described with reference to the accompanying drawings.

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