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[54] AXIAL PISTON UNIT

9 Claims, 6 Drawing Figs.

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[51] Int. Cl. F04b 1/26, F01b 13/04, F01b 31/10
[50] Field of Search 103/162, 162V, 162B; 230/178; 92/57, 156; 91/198, 199

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ABSTRACT: An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis. Support means serve to support the drive shaft, such support means incorporating a substantially ring-shaped hydrostatic axial sliding bearing means acted upon by the pressure of the fluid medium, typically oil for instance. The hydrostatic axial sliding bearing means comprise rotor means and stator means. At least one throttle means is provided for transmitting the pressure of the fluid medium to the hydrostatic sliding bearing means, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium.

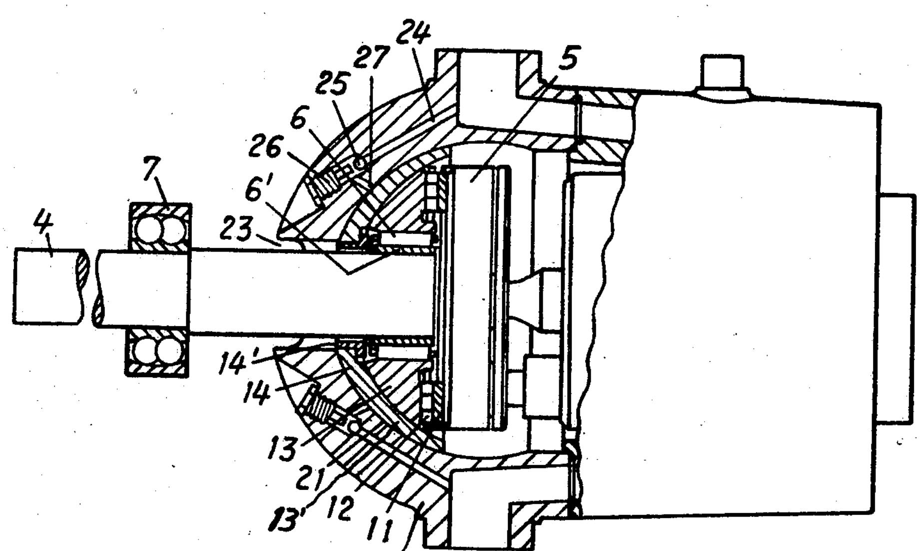

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Fig. 1

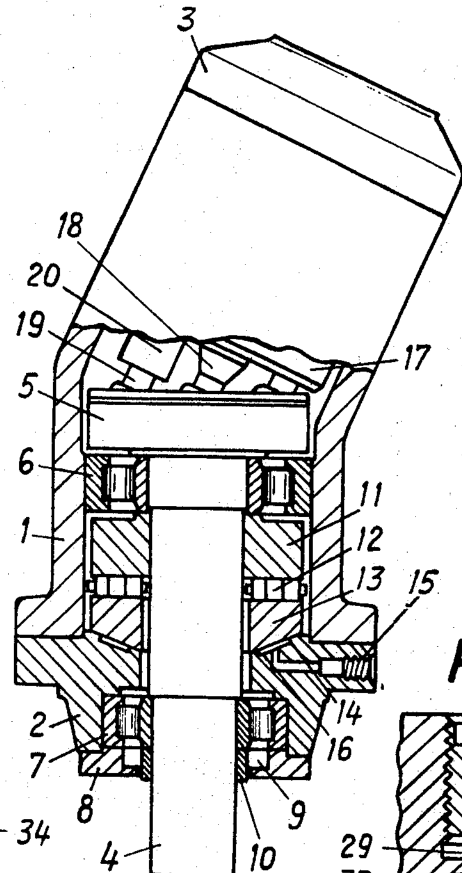

Fig. 5

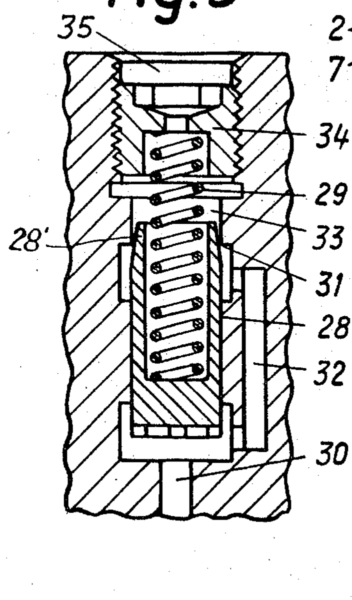

Fig. 6

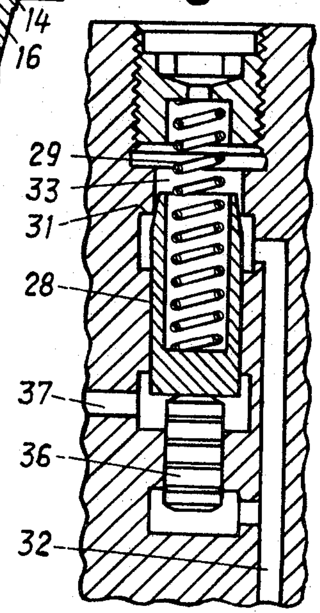

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Fig. 2

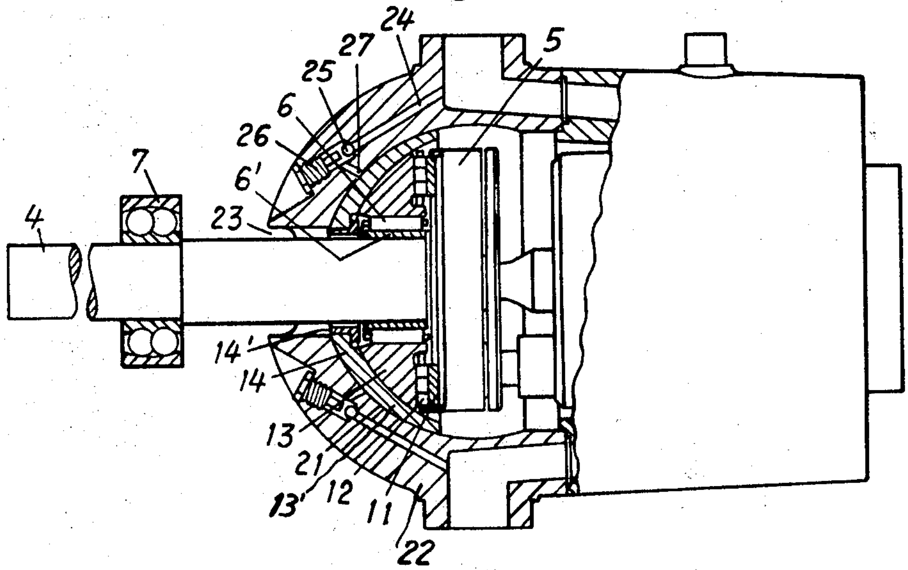

Fig. 3

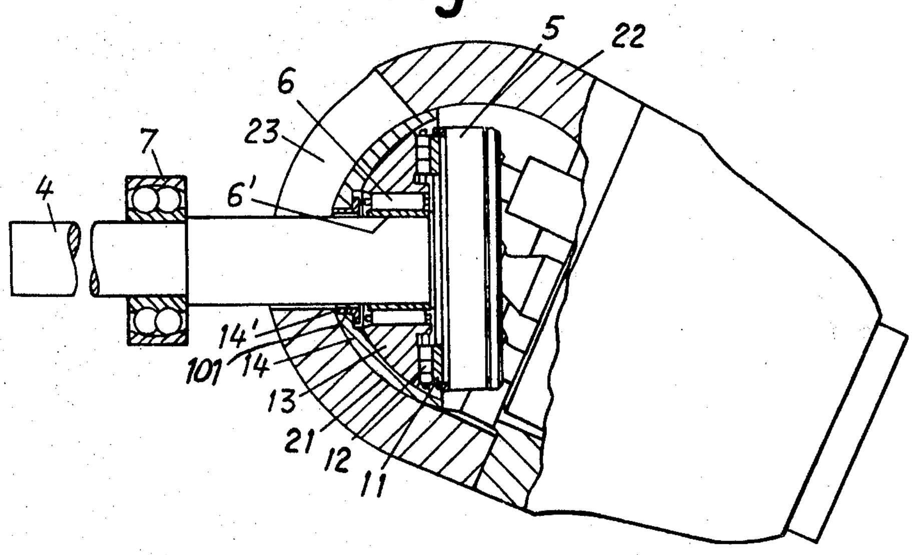

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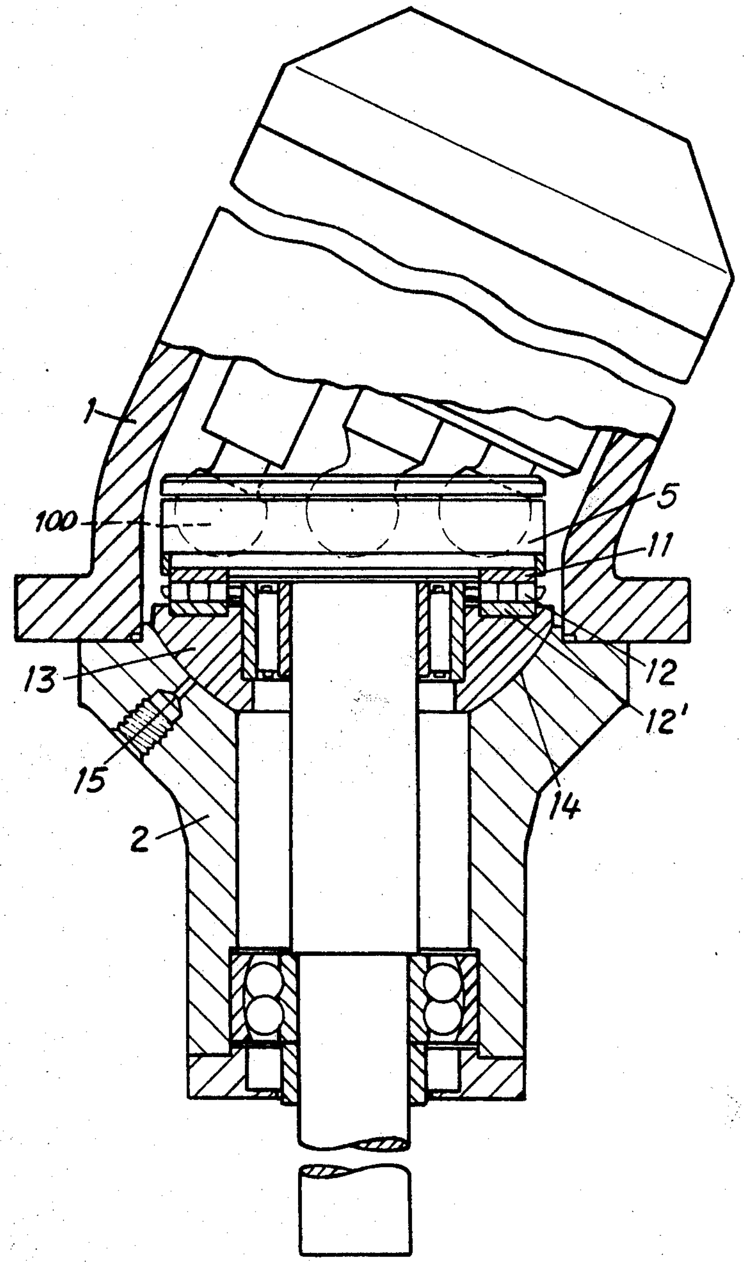
Fig. 4



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AXIAL PISTON UNIT

BACKGROUND OF THE INVENTION

The present invention relates to an improved axial piston unit, such as is employed in the hydraulic oil art both as a pump as well as also a motor.

Axial piston units are known in which the axis of a rotating cylinder block is inclined with respect to the axis of a drive shaft and drive plate or flange rigidly connected with such drive shaft, in order to be able to regulate or adjust for a predetermined stroke. Such type axial piston units are therefore commonly referred to as units having an inclined axis.

The pistons, articulated by means of their piston rods and ball-and-socket joints to the drive plate, are impinged at their respective front faces by the pressurized medium, always one-half of the pistons being subjected to such pressure. Consequently, considerable forces are exerted upon the drive plate or flange, sometimes referred to as a wobble or swash plate. Thus, for instance, with a seven-piston unit having a piston diameter of 25 millimeters, wherein alternatingly three or four pistons are loaded by an operating pressure of 200 atmospheres absolute pressure, with a small stroke, that is to say, when the axis of the cylinder and the axis of the drive shaft are approximately aligned or in coincidence with one another, the thrust or axial force amounts to approximately 3 tons (for three loaded pistons) or approximately 4 tons (for four loaded pistons).

These considerable forces must be transmitted through the agency of suitable bearings to the housing. Because of the compact construction of these units only very little place is available for accommodating such bearings. Thus, special types of roller or antifriction bearings have been developed, especially axial cylindrical roller bearings which are presently used quite extensively. Still, it was not possible to satisfy all the requirements by means of these bearings. This was especially so if they were used in an environment where, on account of the size of the installation, even small standstill periods for the exchange of the bearings do not have to be accepted, or where because of reasons of space greater loaded units had to be employed where the longevity or lifespan of the aforementioned bearings is not sufficient. It was therefore apparent to use sliding or friction bearings instead of the roller or antifriction bearings. Hydrostatic sliding bearings, especially hydrostatic axial sliding bearings are known to the art; likewise their use in axial piston units while making use of the operating pressure. They are capable of satisfying increased operating conditions. A hydrostatic axial sliding bearing, consisting of an annular region or zone or a number of ring or annular segmentlike regions or zones, is supplied throughout its entire periphery with a throttled high pressure which, however, at the unit itself only appears at the pressure side, that is to say, at half of the pistons. During idling when the supply pressure prevails at both pressure sides, the loading of the bearing is relatively twice as great than if a pronounced high pressure would prevail, since then the throttled pressure which is delivered to the axial sliding bearing, depending upon the size of the bearing, proportionally corresponds to the pressure prevailing at both pressure sides, that is to say, it appears at all piston members. Measurements taken of the bearing temperature during idling confirms this characteristic of hydrostatic bearings operated in this manner, in that the bearing temperature during idling exceeds that during the loaded condition.

A further feature or characteristic of the hydrostatic bearing resides in the fact that it is only operationally reliable during the presence of a pressure. When starting, initially there is no pressure, which first then begins to build up during the start of the stroke movement of the piston members. The resulting metallic contact of the bearing surfaces can be only then prevented if, prior to the start of the operation, a special pressure source delivers pressurized oil to the hydrostatic bearings and thus overcomes the phenomena of contact of the bearing surfaces already during standstill.

Accordingly, it is necessary to differentiate between three operating conditions for the hydrostatic bearings or support of a hydraulic unit, namely: The starting condition in which the pressure at the bearing has not yet built up, the idling condition where the bearing is loaded relatively great because of the same or equal pressure which prevails at both pressure sides, and the actual operation where the high pressure prevailing only at one pressure side is available for the entire bearing.

Failure to observe these factors or conditions can lead to considerable difficulties. It is quite possible that because of these reasons there existed a rather limited use of hydrostatic bearings, notwithstanding their unequivocal superiority with regard to their loading and longevity in contrast to roller or antifriction bearings.

SUMMARY OF THE INVENTION

Accordingly, it is a primary object of the present invention to provide an improved axial piston unit which effectively overcomes the aforementioned drawbacks of the prior art structures.

Another, more specific object of the present invention relates to a solution for effectively controlling the characteristics of hydrostatic sliding bearings used in hydraulic units, especially the features associated with axial sliding bearings, so that they can be employed as operationally reliable and maintenance-free structural elements.

Now, in order to implement these and still further objects of the invention, which will become more readily apparent as the description proceeds, the invention contemplates delivering the oil pressure through the agency of at least one throttling device to the hydrostatic sliding bearing, the throttling action of which is at least partially suppressed during low oil pressure and becomes more pronounced with increasing oil pressure.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be better understood, and objects other than those set forth above, will become apparent, when consideration is given to the following detailed description thereof. Such description makes reference to the annexed drawings wherein:

FIG. **1** is a partial sectional view through an axial piston-constant unit employing a first embodiment of the inventive thrust or axial bearing;

FIG. **2** is a partial sectional view through a second embodiment of thrust or axial bearing employed with an axial piston-adjustable unit;

FIG. **3** is a partial sectional view of the arrangement of FIG. **2**, taken substantially perpendicular to the sectional view shown in FIG. **2**;

FIG. **4** is a partial sectional view through an embodiment of thrust or axial bearing unit substantially of the type according to FIG. **2**, but here used with an axial piston-constant unit;

FIG. **5** is a fragmentary cross-sectional view showing details of a throttling device; and

FIG. **6** is a similar fragmentary sectional view of a second embodiment of throttling device.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Describing now the drawings, in FIG. **1** reference numeral **1** designates the housing of a constant or invariable hydraulic unit. This housing **1** is closed by the drive plate cover member **2** and the control level cover member **3**. The nonillustrated bores for the infeed and outfeed of the pressurized fluid medium, typically oil, are arranged at the control level cover member **3**, and with which bores there are connected appropriate conduits. The drive shaft **4**, advantageously formed of one piece or integral with the drive plate **5**, is mounted by means of two radial bearings, namely the inner bearing **6** and the outer bearing **7**, both shown as cylindrical roller bearings, at the housing **1** and at the drive plate cover member **2**, respectively. By means of the end plate or further cover member **8**, having a recess **9** in which there is mounted a nonillustrated standard lip-sealing device bearing by means of its sealing lips at the traveling ring **10**, the housing **1** is sealed against loss of oil.

The axial or thrust bearing unit, consists of a first rotor ring **11**, the cylindrical roller bearing means **12**, the second rotor ring **13**, and the stator plate **14** which is defined by the portion of the drive plate cover member **2** directed towards the interior of the housing **1**. By means of the connection or duct **15**, the throttled operating medium flows into the annular or ring zone **16** of the hydrostatic bearing. This ring zone **16** can also be subdivided into individual ring segmentlike areas or zones wherein each possesses a separate infeed for the throttled operating medium beginning at the connection **15**.

By means of suitable ball-and-socket joints **100** (FIG. **4**) it is possible to hingedly connect the support rod **18** mounted at the cover member **3** and which guides the cylinder **17** as well as the piston rods **19** and their associated piston members **20**, with the drive plate **5**.

The sliding thrust or friction bearing functions in the following manner: The operating medium flows throttled through the connection **15**, in a manner which will be described more fully hereinafter, into the ring zone or area **16**, raising the second rotor ring **13** to such an extent until there appears at this ring zone **16** that pressure which is in equilibrium with the thrust or axial force exerted by the piston members **20**. This ring member **13** can therefore rotate without contacting the a stator **14**, so that for the frictional force there is only operative the shearing force of the operating medium which is generated by such operating medium.

The axial force or thrust of the piston members **20** is transmitted by the first rotor ring **11**, fixedly seated upon the drive shaft **4**, to the roller bearing unit **12** and from this location to the second rotor ring **13**.

The frictional force of the roller bearing **12** is dependent upon the axial force or thrust of the piston members **20** and, accordingly, during idling is very small, in fact, smaller than the frictional force of the hydrostatic sliding bearing **13, 14**. This means that during this operating condition the second rotor ring **13** acts as a stator, in other words is at standstill or only rotates with a very small rotational speed. Now, if the operating pressure increases, then also the friction of the roller bearing **12** increases and, at a certain operating pressure, it becomes greater than the shear force of the operating medium at the hydrostatic slide bearing **13, 14**, this shearing force essentially only being dependent upon the viscosity and the size or height of the gap. It therefore increases considerably slower or hardly with the operating pressure, so that at a certain operating pressure the second rotor ring **13** begins to rotate and the roller bearing remains at standstill or only rotates very slowly with respect to both rotors.

The change from the one to the other of both-mentioned operating conditions occurs when going from idle to load. Initially, the axial force or thrust is small, so that only the roller bearing is effective. Then, when during the operation the pressure builds up, the moment occurs where the hydrostatic sliding or friction bearing becomes completely effective and the roller bearing practically is at standstill. Theoretically, the situation could occur where each of both bearings assumes one-half of the rotational speed. However, in practical operation always one or both operating conditions exists, where at one time the roller bearing and at one time the hydrostatic slide bearing has imparted to it almost the entire rotational speed.

In the arrangement shown in FIGS. **2** and **3** there is again depicted a drive shaft **4**, a drive plate or flange **5**, an inner radial bearing **6** and an outer radial bearing **7**. The construction of the axial or thrust bearing arrangement is similar to that of FIG. **1**, however, the individual elements are arranged somewhat differently. Reference numeral **11** designates the first rotor ring which in this case bears directly against the drive flange or plate **5**, reference numeral **12** the cylindrical roller bearing, and reference numeral **13** the second rotor ring which is mounted together with the inner radial bearing **6** upon the drive shaft **4**. The inner radial bearing **6** has an inner ring or race **6'**, but no external ring or race, and therefore travels in the second rotor ring **13** which, for this reason, must be manufactured of ball-bearing steel or a similar material.

The second rotor ring **13**, the bearing or support surface **13'** of which is spherical as in FIG. **1**, which, however, is not in any way an absolute necessity for its functionality although advantageous, travels in the stator **14**. This stator **14** possesses the form of a bowl having a spherical convex external surface **21**. This stator **14** is stationary with respect to the rotor ring **13**, yet it is pivotable with respect to the housing **22**. For this purpose, a slot or recess **23** is formed in the housing **22**, as best shown by referring to FIG. **3**, and in which the shaft **4** can move from the illustrated extreme position for maximum stroke through the null-stroke position until reaching the other extreme position of maximum stroke.

A flange sleeve **14'** is located at the stator **14** which forms a gap or slot **101** with the shaft member **4** through which the leakage oil of the hydrostatic sliding bearing, arranged between the rotor ring **13** and the stator **14**, can flow away.

The hydrostatic sliding bearing is impinged by the momentary high pressure through the agency of the bore **24**, the check or nonreturn valve **25**, here shown in the form of a ball, and the displacement of which can be limited or regulated by the screw **26**, and the channel **27**. One or a number of nonillustrated channels leads from the outside region **21** of the stator **14** to the hydrostatic bearing and supply such throughout the entire annular circular region. In so doing, the momentary low-pressure side is blocked by the check valve **25** from the high-pressure side.

In FIG. **4** there is depicted a hydrostatic axial slide bearing which has been mounted at a constant unit. The arrangement is essentially similar to that of FIGS. **2** and **3**, but it is possible to dispense with the stator bowl **14** of FIG. **2** because of the fixed pivot angle. Reference numeral **11** designates the first rotor ring upon which travels the cylindrical roller bearing **12**. A further bearing ring **12'** is situated in the second rotor ring **13**, the rear portion of which together with the stator surface **14** forms the hydrostatic slide bearing. The stator surface **14**, similar to FIG. **1**, is arranged in the drive plate cover member **2** at the inside thereof, which provides a portion of the housing **1** of the unit. The hydrostatic slide bearing **13, 14** is supplied via the conduit **15**, and the pressurized medium is distributed by means of nonillustrated branch conduits into the spherical bearing ring surface arranged and distributed about the drive plate **5**. The admission of the momentary high pressure to the hydrostatic slide bearing takes place externally of the bearing by means of a nonillustrated conventional reversing valve.

The stability of the hydrostatic sliding bearing, that is to say, the maintaining of a gap suitable for the function of the bearing, even when the load changes, is insured in known manner in that a stationary or fixed prethrottling device is connected before the hydrostatic slide bearing, whereas the bearing gap provides a variable throttle. If, for example, the bearing gap is too large with regard to the equilibrium condition, then the pressure in the bearing drops and the gap becomes momentarily smaller. On the other hand, in reverse fashion with a gap which is too small the pressure in the bearing increases and the rotor ring is raised until the equilibrium condition has been maintained.

The prethrottling valve depicted in FIG. **5** provides a throttling arrangement which is dependent upon pressure, and specifically, in such a manner that with increasing pressure the throttling is increased. This valve consists of a hollow throttling piston **28** having a conically tapered portion **28'** and which is pressed into its cylindrical bore by the spring **29** towards the infeed mouth **30**. With small operating pressure or during idling, the spring **29** presses the piston **28** against the mouth portion **30** and therefore at least partially frees the throttle cross section, so that the operating medium can flow with very little throttling through the channel **32** into the compartment **33** and through the through bored nut member **34**, against which bears the spring **29**, into the compartment **35**, the pressure of which is identical with the pressure in the hydrostatic slide bearing.

The throttle cross section is formed by the conically tapered throttle piston 28 and the edge portion 31. However, this throttle piston 28 can be also cylindrical and possess at least one groove situated at its external or outer surface, the effective length of which increases with increasing operating pressure. The groove could also be provided at the cylindrical wall of the compartment 33, in which case, then, the throttle piston 28 would possess a smooth wall construction.

The entire prethrottling valve can be, as shown in FIG. 5, mounted in a bore which is situated at the stator or in a special housing.

In FIG. 6 there is shown a slight variant embodiment of the prethrottling valve depicted in FIG. 5. In order that the spring member 29 can be dimensioned smaller, a small measuring piston 36 is serially arranged before the throttle piston 28 and the end face of the throttle piston 28 is connected via a channel 37 to a pressureless compartment, so that the pressure acting upon the measuring piston 36, on the one hand, is in equilibrium with the pressure acting in the compartment 33 upon the throttle piston 28 and, on the other hand, with the force of the spring member 29.

The hydrostatic sliding or friction bearing which is impinged by the operating pressure of an axial piston unit and having the prethrottle valve arrangement provides a bearing which is efficient and operationally reliable for the majority of operating conditions. During starting, where the pressure has not yet built up at the slide bearing and during idling, the roller bearing arranged in series with the slide bearing provides a supplementary device by means of which it is possible to prevent the completely insignificant starting wear at the slide bearing, and further, to additionally reduce or, in fact, prevent the temperature increases which occur during idling, which already are effectively reduced by the prethrottling valve due to the taking over of a portion of the rotational speed.

It should be apparent from the foregoing detailed description, that the objects set forth at the outset of the specification have been successfully achieved.

I claim:

1. An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis, means defining an outlet for said axial piston unit, support means for said drive shaft, said support means incorporating a substantially ring-shaped hydrostatic axial sliding bearing means acted upon by the pressure of the fluid medium, said hydrostatic axial sliding bearing means including rotor means and stator means, and at least one throttle means for delivering the pressure of the fluid medium to said hydrostatic sliding bearing means, said throttle means communicating with said outlet and operably communicating with said axial sliding bearing, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium.

2. An axial piston unit as defined in claim 1 wherein said support means for said drive shaft further includes an axial roller bearing means arranged in series with said hydrostatic axial sliding bearing means.

3. An axial piston unit as defined in claim 2, further including a housing, said rotor means including a first rotor element and a second rotor element, said stator means incorporating a pivotably mounted stator bowllike element arranged between said housing and said second rotor element.

4. An axial piston unit as defined in claim 2, wherein said support means further including at least an inner radial bearing means, said rotor means encompassing a first rotor element and a second rotor element, said second rotor element carrying both said axial sliding bearing means as well as said inner radial roller bearing means.

5. An axial piston unit as defined in claim 1, wherein said throttle means includes a throttle piston which is conically tapered at one end region thereof.

6. An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis, support means for said drive shaft, said support means incorporating a substantially ring-shaped hydrostatic axial sliding through the agency of said measuring piston, bearing means acted upon by the pressure of the fluid medium, said hydrostatic axial sliding bearing means including rotor means and stator means, and at least one throttle means for delivering the pressure of the fluid medium to said hydrostatic sliding bearing means, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium, said throttle means including a substantially cylindrical throttle piston and provided with a throttle cross section which is formed by at least one groove means, the effective length of which is dependent upon the position of said cylindrical throttle piston, said position of said cylindrical throttle piston being regulated as a function of the operating pressure of the fluid medium.

7. An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis, support means for said drive shaft, said support means incorporating a substantially ring-shaped hydrostatic axial sliding bearing means acted upon by the pressure of the fluid medium, said hydrostatic axial sliding bearing means including rotor means and stator means, and at least one throttle means for delivering the pressure of the fluid medium to said hydrostatic sliding bearing means, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium, said throttle means incorporating a throttle piston and a measuring piston, said operating pressure acting upon said throttle piston through the agency of said measuring piston.

8. An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis, support means for said drive shaft, said support means incorporating a substantially ring-shaped hydrostatic axial sliding bearing means acted upon by the pressure of the fluid medium, and an axial roller-bearing means arranged in series with said hydrostatic axial sliding bearing means, said hydrostatic axial sliding bearing means including rotor means and stator means, and at least one throttle means for delivering the pressure of the fluid medium to said hydrostatic sliding bearing means, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium, said throttle means including a substantially cylindrical throttle piston and provided with a throttle cross section which is formed by at least one groove means, the effective length of which is dependent upon the position of said cylindrical throttle piston, said position of said cylindrical throttle piston being regulated as a function of the operating pressure of the fluid medium.

9. An axial piston unit of the type comprising a drive shaft and a cylinder having an inclined cylinder axis, support means for said drive shaft, said support means incorporating a substantially ring-shaped hydrostatic axial sliding bearing means acted upon by the pressure of the fluid medium, and an axial roller-bearing means arranged in series with said hydrostatic axial sliding bearing means, said hydrostatic axial sliding bearing means including rotor means and stator means, and at least one throttle means for delivering the pressure of the fluid medium to said hydrostatic sliding bearing means, the throttling action of said throttling means being at least partially suppressed during low pressure of the fluid medium and becoming more pronounced during increasing pressure of the fluid medium, said throttle means incorporating a throttle piston and a measuring piston, said operating pressure acting upon said throttle piston through the agency of said measuring piston.