United States Patent [19]

Molly

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[45] **Dec. 4, 1973**

[54] **HYDROSTATIC DRIVE UNIT**

[76] Inventor: **Hans Molly,** Dr. Eugen-Essig-Strasse 48, Malsch, Germany

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[30] **Foreign Application Priority Data**

Feb. 4, 1971 Germany ..... P 21 05 119.4

[52] **U.S. Cl.** **60/490,** 60/DIG. 10, 60/484

[51] **Int. Cl.** **F16h 39/46**

[58] **Field of Search** ..... 60/53 A, 52 US, 490, 60/487, DIG. 10

[56] **References Cited**

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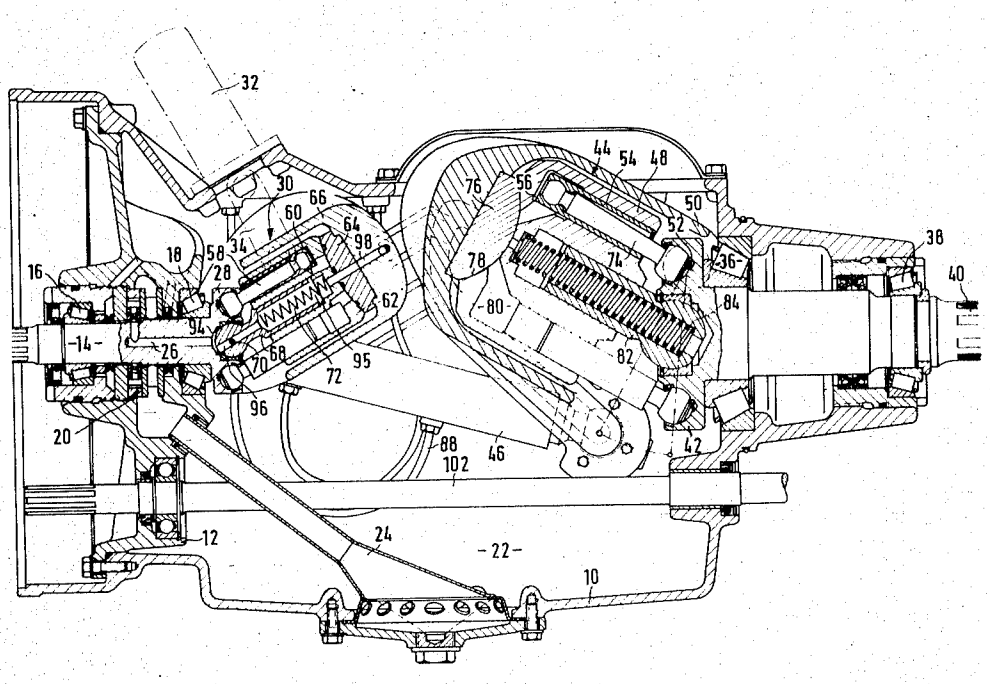
3,175,363 3/1965 Molly ..... 60/53 A

Primary Examiner—Edgar W. Geoghegan
Attorney—Darbo, Robertson & Vandenburgh

[57] **ABSTRACT**

A hydrostatic drive unit comprises a pump driven by a prime mover and having constant delivery during normal operation, and at least one variable stroke axial piston-type motor fed by the pump. The intake volume per revolution of the motor is variable by pivoting the cylinder block about an off-center pivot axis through an angle of more than 30° such that the dead space in the cylinders is kept as small as possible. The intake volume per revolution of the motor at maximum pivot position is a multiple of the delivery volume per revolution of the pump. The cylinder block of the motor is carried along by the drive flange through peripherally arranged teeth in mesh in the region of the pivot axis and permitting the pivoting movement of the cylinder block.

9 Claims, 15 Drawing Figures



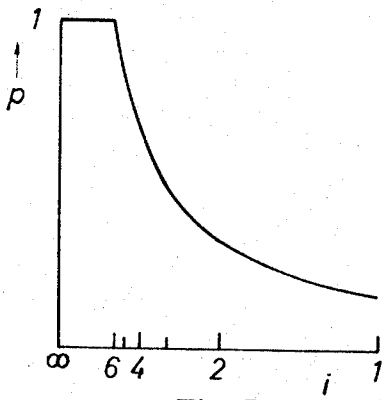


Fig. 5

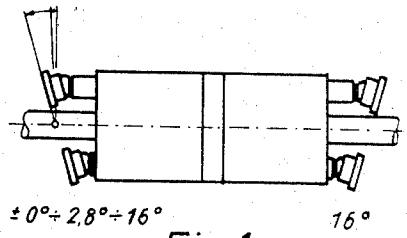


Fig. 1

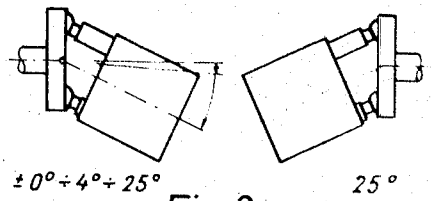


Fig. 2

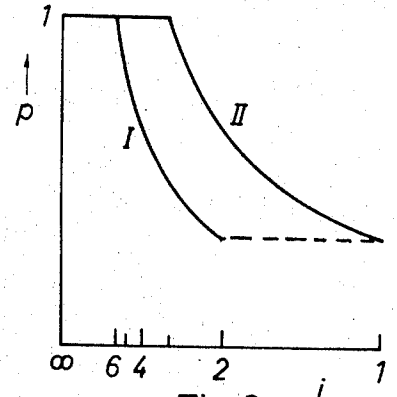


Fig. 6

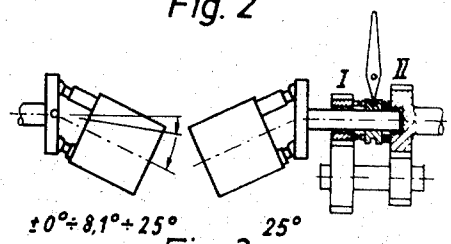


Fig. 3

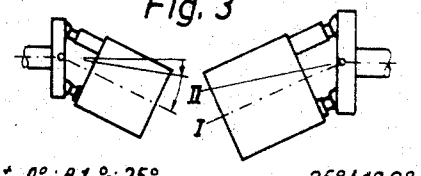


Fig. 4

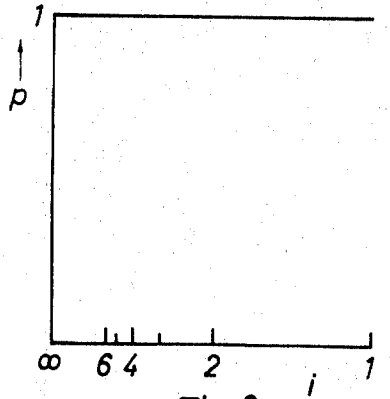


Fig. 8

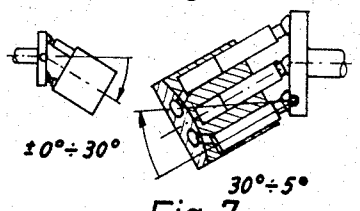


Fig. 7

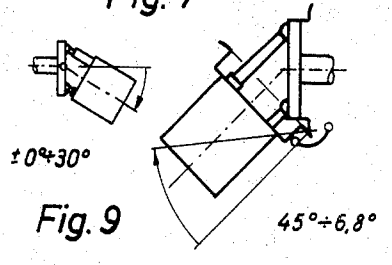


Fig. 9

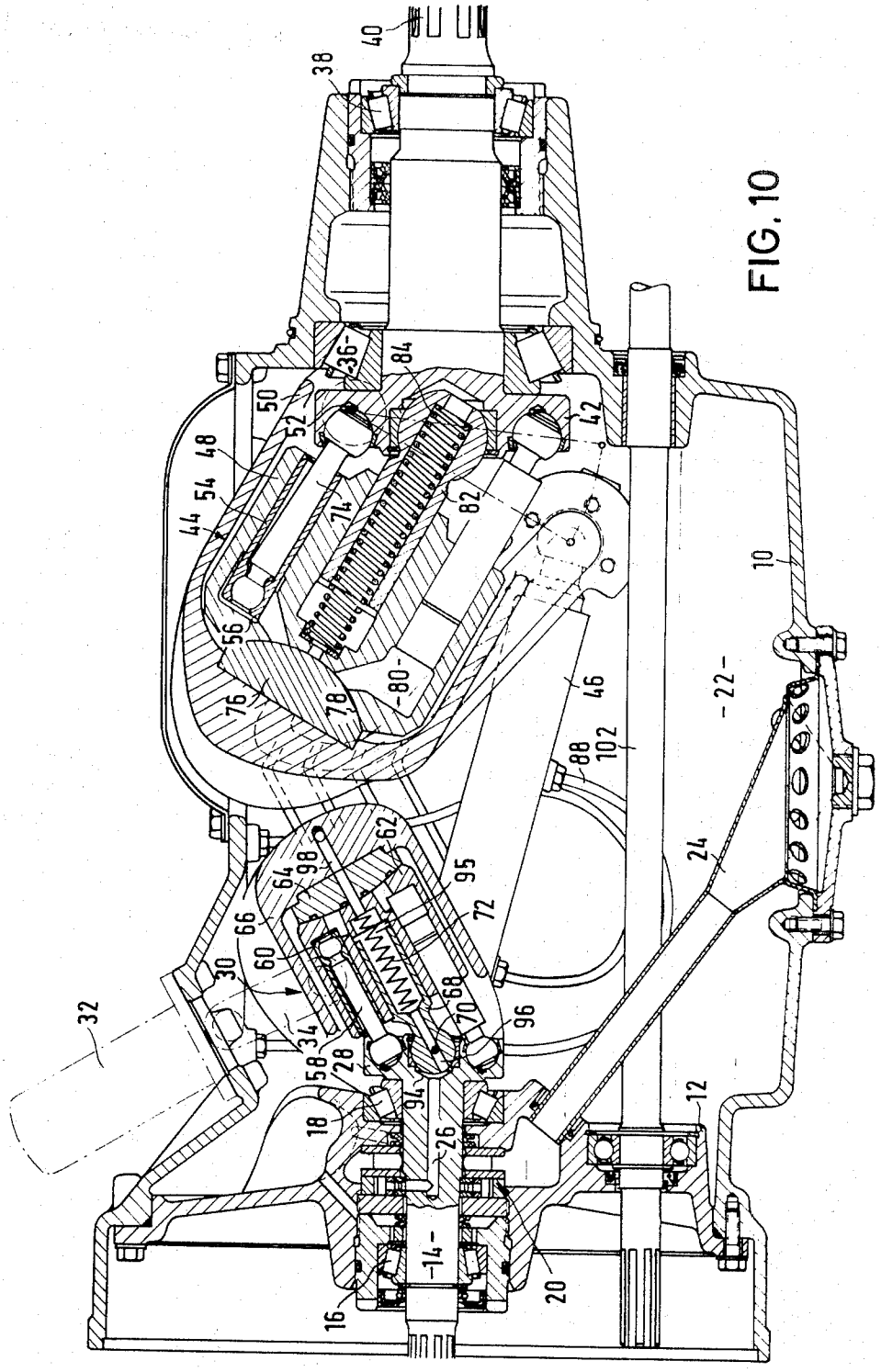

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FIG. 10

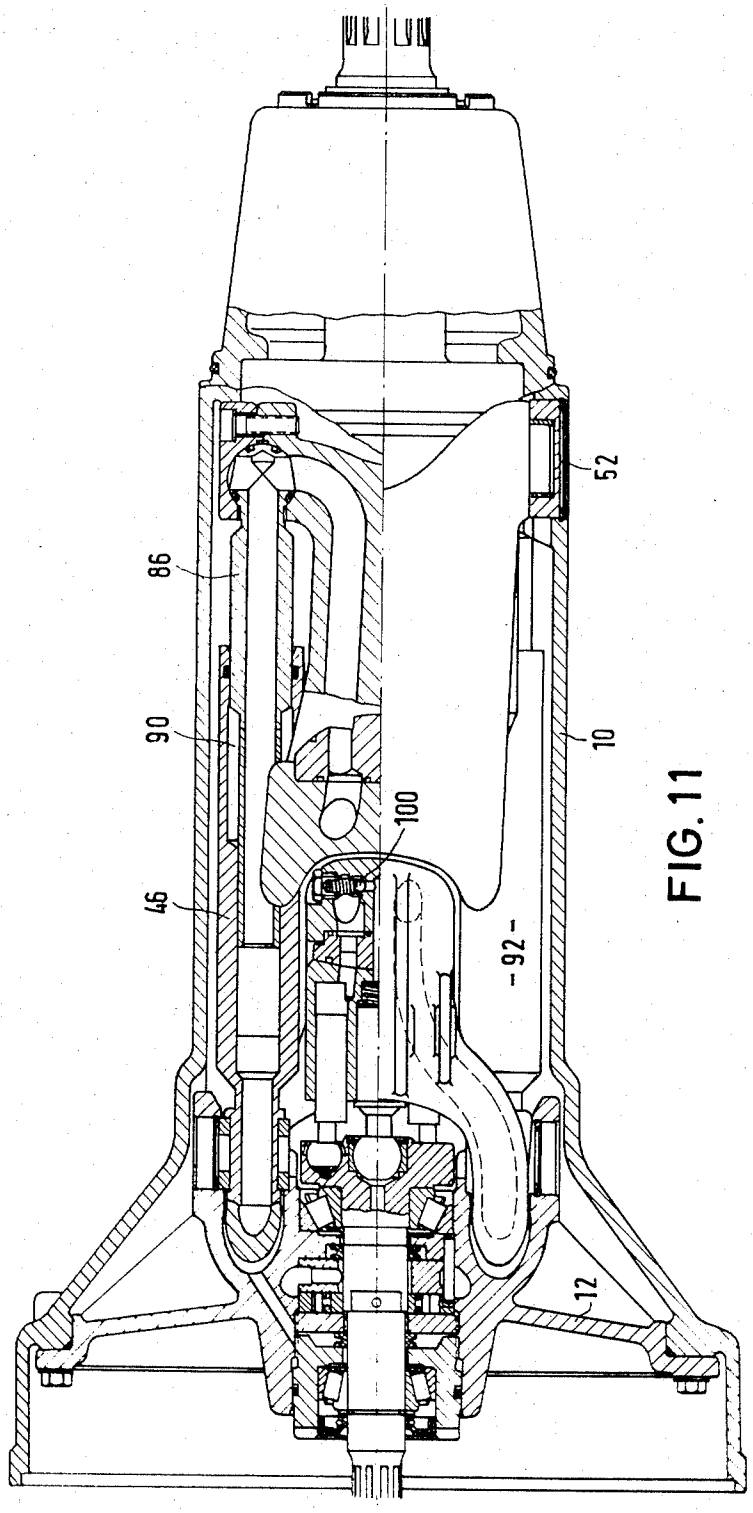


FIG. 11

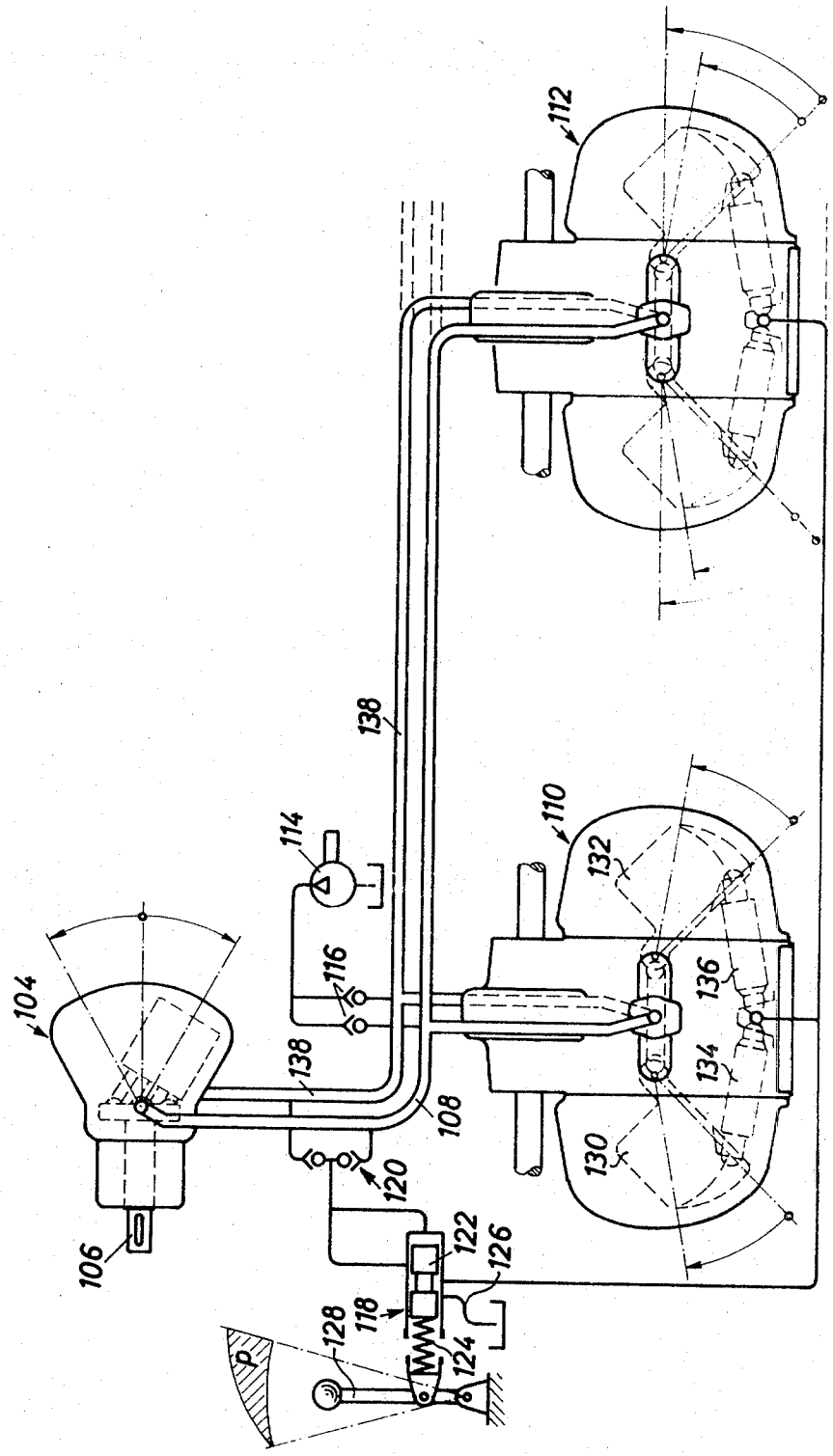

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Fig. 12

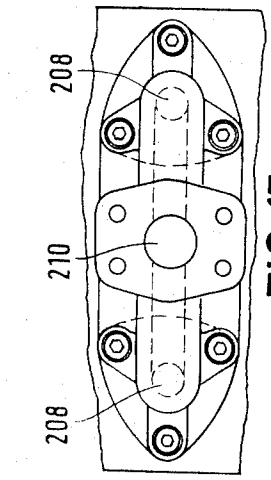


FIG. 15

FIG. 13

FIG. 14

HYDROSTATIC DRIVE UNIT

The invention relates to a hydrostatic drive unit comprising a pump driven by a drive shaft and at least one axial piston-type motor fed by the pump, said motor driving a driven shaft and which has a drive flange connected to the driven shaft, a cylinder block pivotable with respect to the driven shaft in a pivotable frame as well as axial pistons guided in the cylinder block and in articulated connection with the drive flange through piston rods.

In this case, as is well known, the axial piston-type motor is designed as a motor which develops torque on a swash plate, namely the drive flange. The torque produced by the oil pressure originates at the drive flange, while the cylinder block is only carried along by the friction being overcome either by the piston rods or by an additional drive connection. The stroke variation takes place in such a motor by the cylinder block being pivoted in a pivotable frame with respect to the drive flange, with the drive flange having a fixed position on the driven shaft.

Among the known axial piston units are those in which the pistons are supported through cross-head shoes or the like on pivotable slipping discs. The cylinder blocks are arranged coaxially from pistons and motor and keyed to the drive or driven shaft. In this case the torques are produced at the cylinder blocks (FIG. 1). Such drive units have the advantage of a compact construction, however are somewhat larger in size, because the angles of pivot cannot be designed beyond 15°–17° for constructional and functional reasons. In the design of a hydrostatic drive unit the pump must, on the one hand, be able, at full pivot, to supply a stream of hydraulic fluid sufficient to give the hydraulic motor the requisite maximum rotational speed. On the other hand, however, the pump must also be able to yield a high pressure pressur in the regions of the requisite largest transmission, wherein it must be taken into account that with any readjustment of the stroke volume the pressure increases with reciprocal magnitudes. Thus there occurs according to a hyperbolic curve and with large transmission this maximum pressure, which is necessary for constant power transmission, which must be obtainable. The result of this is that in large control ranges in direct operational drive, operation is effected with very low pressures.

In the design of the size of a gear pump one starts from a so-called angular output, the output which results from the product of the high pressure and the large conveying stream. In operation the machine is never stressed with this output, but its size is so large, and in particular in the range of the direct drive that low total efficiency results on account of the low oil pressure. For these reasons the design is generally limited to 3:1 in the control range. If one exceeds this, dimesnions, as FIG. 1 shows, result.

Furthermore drive units are known, which operate with axial piston-type pumps and motors in which the torque — as described — is developed on the swash plate. In the known drive units the motor is embodied as an axial piston-type motor with a fixed angle between the drive flange and the cylinder block, wherein therefore, the adjustment of the drive unit takes place by pivoting the cylinder block. The conveying volume of the pump at full angle of pivot corresponds — as has already been mentioned in the above example FIG. 1 — to approximately the intake volume of the motor so that with full angle of pivot of the pump, there is a direct transmission of rotational speed. With a return pivoting of the pump, a reduction in rotational speed is achieved, FIG. 2. The construction of these machines offers a more favorable mechanical efficiency and achieves a further improvement by the larger angles of pivot which can be used here. However, a similar restriction of the control range — as in the case of the construction in FIG. 1 — also has to be counted on because of the oil pressure which occurs in a hyperbolic form.

For these reasons it is known to combine a continuously regulatable hydrostatic drive unit with a back gear which is reversible in steps (FIG. 3). In this way a sufficiently large regulation range is indeed obtained. This range is made up of two steps. In such devices a smaller overall machine is obtained as a result of the design determined by the angular output. However, such a combined mechanical-hydraulic solution is crude and expensive.

Moreover, a hydrostatic drive unit is known, in which both the pump and the axial piston-type motor, which is substantially larger than the pump, are equipped with a pivotable cylinder block (FIG. 4). Here the control of the transmission ratio also takes place in the normal control operation in the same way as in the drive unit explained with reference to FIG. 3 by the pump, and the additional pivoting of the motor cylinder, taking over the function of the back gear, namely an additional extension of the control range. Now — compared with FIG. 2 — only the adjustable motor must still be kept large, while the pump — thanks to the two-step stepping down — yields a smaller angular output.

FIG. 5 shows the pressure (p)— reduction gear ratio (i)— characteristic curve in drive units of the type according to FIG. 1 and FIG. 2. FIG. 6 shows corresponding characteristic curves I and II for the two change-over positions of the back gear (FIG. 3) or pivoting positions of the motor (FIG. 4) respectively in the drive units of FIGS. 3 and 4. In the known drive units according to FIG. 4 the resettings from I to II change-over positions, which thus replace the drive unit arrangement from FIG. 3, but here adjustments from the position I to II are carried out in a separate adjustment operation, if the pump has assumed full angles of pivot. In the diagram of FIG. 6, in this case, the oil pressure in the dashed line extends from the end point of the hyperbola I to the end point of the hyperbola II. Also mixed adjustments at which the pump and the motor are commonly adjusted according to predetermined mathematical relationships, are in use. In this case the oil pressure runs inside the space enclosed between the two hyperbolas I and II.

The pump illustrated in FIG. 4 is still subjected to a dimensioning by the angular output which indeed as compared with the example shown in FIGS. 1 and 2, has been reduced to half and corresponding to the take-over of the adjustment range through the hydraulic motor. However, still no satisfactory dimensioning is given to the installation and also sufficient account is not taken of the required control range, since any resetting of the hydraulic motor to a still smaller angle leads to intolerable losses in efficiency.

The object of the invention is to design a drive unit of the above-mentioned type in such a manner that it

is space-saving, simple and inexpensive and has a good degree of efficiency in the whole range of operation.

According to the invention, this problem is solved in that in the normal control operation the adjustment takes place with fixed conveying volume of the pump on the driven side by pivoting the cylinder block, in that the intake volume per rotation of the axial piston-type motor at a maximum pivoting position of the motor is a multiple of the conveying volume per roatation of the pump, and in that the cylinder block of the axial piston-type motor is pivotable about an off-centre axis such that the dead space in the cylinders is kept as small as possible (FIG. 7).

So that the adjustment in normal control operation — apart from starting, if necessary — takes place only through the motor, favorable dimensioning of the pump is rendered possible. The large size of the motor, as compared with the pump permits a large range of step-up, e.g. from $i = 1$ to $i = 6$ and more, to be included without any difficulties occurring with respect to the angle of pivot. Though in a motor dimensioned in this way — particularly in direct operation (or smallest angle of pivot) — with the usual cylinder block pivoting about an axis intersecting the drive flange axis, the hydraulic efficiency would be bad. Namely the relatively large dead oil volumes in the motor would be ocntinuously compressed without direction and expanded. Since in a constantly pivoted pump the pressure p remains constant, i.e. very high (FIG. 8), this influence is very large. The further feature of the pivot axis disposed off-center counteracts this, by which the dead space can be kept small.

It is advantageous to make the cylinder block of the axial piston-type motor pivotable to an angle of pivot larger than 30°. This can particularly advantageously be solved constructively in such a manner that the cylinder block is carried along by the drive flange through peripherally arranged teeth which are designed in such a manner as to permit a pivoting movement between the cylinder block and the drive flange, and are in mesh with each other in the region of the axis of pivot of the cylinder block (FIG. 9).

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a diagrammatic side elevation of a prior art hydrostatic transmission.

FIG. 2 is a diagrammatic side elevation of another form of prior art hydrostatic transmission.

FIG. 3 is a diagrammatic side elevation of still another form of prior art hydrostatic transmission.

FIG. 4 is a diagrammatic side elevation of a fourth form of prior art hydrostatic transmission.

FIG. 5 shows the pressure versus drive ratio characteristic of transmissions according to FIGS. 1 and 2.

FIG. 6 shows similar characteristics for the transmission of FIG. 4.

FIG. 7 is a side elevation, partly in section, of a transmission of the invention.

FIG. 8 shows the pressure versus drive ratio characteristic of a transmission of FIG. 7.

FIG. 9 is a diagrammatic side elevation of another embodiment of a transmission of the invention.

FIG. 10 shows, in section, a first embodiment of a drive unit according to the invention.

FIG. 11 is associated plan view with portions broken away.

FIG. 12 schematically illustrates a second embodiment having several hydraulic motors fed by a pump.

FIG. 13 is a section with portions broken away of a double motor shown in FIG. 12.

FIG. 14 shows a section along the line X—X in FIG. 13.

FIG. 15 shows a view of the pressure output for the double motor.

DESCRIPTION OF SPECIFIC EMBODIMENTS

In the embodiment according to FIGS. 10 and 11, 10 designates a drive unit housing which is closed on the drive side by a cover part 12. A drive shaft 14 is mounted in bearing in the cover part 12. The drive shaft is dirven by a prime mover, not shown. A filling pump 20, designed as a cycloid pump, is keyed on the drive shaft. This pump draws in oil from a sump 22 through a conduit 24 and discharges it in a chennel 26 which runs centrally in the drive shaft 14.

The drive flange 28 of an axial piston pump generally 30 is disposed on the drive shaft. The axial piston pump works in the normal control range with constant pivot position and correspondingly constant qunatity of hydraulic fluid being moved per rotation, but for starting purposes it can be angularly adjusted by an adjusting cylinder 32 through a guide rod 34.

On the other side of the housing 10 the driven shaft 40, on which the drive flange 42 of an axial piston-type motor generally 44 is disposed, is mounted in bearings 36 and 38. In the normal control operation the ratio of drive rotational speed to driven rotational speed of the drive unit is effected by adjusting the motor 44. The motor 44 is angularly adjusted by an adjusting cylinder 46.

The intake volume of the motor 44 in the illustrated, fully pivoted position, is a multiple of the discharge volume of the pump. Therefore a relatively large range of driven rotational speeds can be achieved, the angles of pivot of the motor being kept within required limits. On the one hand, the motor is pivoted back to a small angle of pivot not impairing the efficiency, while on the other hand the maximum angle of pivot is determined by the physical characteristics of the machine. Since with a relatively large motor, particularly in the range of direct drive (small angle of pivot), an impairment of the efficiency would normally occur by compression and expansion of the oil volume, the cylinder block 48 is pivotable in a pivotable frame 50 about an off center axis 52 so that the oil volume in the upper dead center of the pistons 54 and 56 can be kept as small as possible in all positions. Thus the dead space of the motor is kept small in a manner known per se by this construction. Also see patent application Ser. No. 199,974, filed Nov. 18, 1971.

The drive unit is constructed as follows in detail:

The axial piston pump 30 includes the drive flange 28. Piston rods 58 are articulated to the drive flange 28. The piston rods are also articulated to axial pistons 60 which are guided in a cylinder block 62. The cylinder block 62 bears against a spherical surface of a stationary valve portion 64 held in a pivotable frame 66. The cylinder block 62 is guided on a trunnion 68 which is articulated at a pivot point 70 on the drive flange. The trunnion 68 is hollow and contains a spring 72 which presses the cylinder block 62 against the stationary valve portion 64.

The motor includes the drive flange 42 to which the piston rods 74 are articulated. The piston rods are ar-

ticulated to the pistons **54** which are guided in the cylinder block **48.** The cylinder block **48** is mounted in a pivotable frame **50** and abuts a stationary valve portion **76** having a spherical control surface **78.** The cylinders in the cylinder block **48** communicate with the stationary valve portion **76** through inclined channels **80** so that a small diameter of the reversing channels results.

The cylinder block **48** is likewise guided on a hollow trunnion **82** which is centrally articulated to the drive flange **42.** A compression spring **84** in trunnion **82** presses the cylinder block against the stationary valve portion **76.** Oil is fed to the motor by the pump through the adjusting cylinder **46** designed as a telescopic tube, the piston **86** (FIG. **11**) in the adjusting cylinder being designed as a differential piston. The oil pressure present in the adjusting cylinder **46** counteracts the torque which is caused by the oil pressure of the cylinder block **48** as a result of the off-center pivoting positioning of the pivotable frame **50.** Pressure oil can be supplied through conduits **88** to the space **90** at the differential steps (FIG. **11**) of the adjusting cylinder **46** in order to effect an adjustment of the angular position of the motor. The return flow of the discharge oil from the motor to the pump takes place through an adjusting cylinder **92** similarly constructed as a telescopic tube, its step being effective in a similar manner.

The filling oil from pump **20** is conducted through the conduit **26,** a pressure field **94** under the articulated head of the pivotable trunnion **68,** a pressure field **95,** a channel **98** and a check valve **100** (FIG. **11**) to the intake side of the stationary valve portion **64.**

A power take-off shaft **102** is mounted in the housing **10.** By this shaft a direct drive from the prime mover to any operating apparatus assembly can be obtained.

In the embodiment of FIGS. **10** and **11** the pump **30** normally works at a fixed pivot position of 30°. Before starting the pump is at an idling position of 0° and to start is moved from that position to ± 30°. Thereafter the rotational speed regulation takes place by pivoting the motor **44** which can be pivoted to a maximum pivot position of from 30° to 50°, the 5° pivot position of the motor corresponding to the direct drive 1:1.

FIG. **12** shows the hydraulic circuit of another embodiment of the invention. There is a pump generally **104** which is driven by a drive shaft **106.** The pump is — as can be seen — pivotable from a zero position about 30° to both sides. The mean zero position in this case corresponds to idling or stop. It can then be started forwards or backwards until the pump has attained its pivot position of +30° or −30°. Oil flows through a conduit **108** to the motor assemblies generally **110, 112,** etc. A filling pump **114** supplies filling oil into the system through check valves **116.** There is a pressure regulator generally **118** which keeps the oil pressure constant in whichever conduit **108** or **138** is serving as the supply conduit, through a check valve arrangement **120.** When conduit **108** serves as the supply conduit, conduit **138** serves as the return conduit and vice versa. The pressure regulator **118** includes a slide **122** which is by a spring **124** and controls the fluid communication between the supply conduit **108** or **138** and an outlet **126.** The force exerted by the spring **124** is adjusted by an control lever **128.** The greater the force applied by the spring the greater will be the pressure in the supply conduit (**108** or **138**) and the less the force the less the pressure.

Each motor assembly includes two motors **130, 132** which are articulated in pivotable frames so as to pivot in an off-center manner. The hydraulic moments acting on the pivotable frames are taken up by the adjusting cylinders **134, 136,** by which an adjustment of the angular position of the motor cylinder blocks is possible.

Conduit **138** leads from the other connection of the pump **104** to the other connections on motor assemblies **110, 112,** etc. When the pump **104** pivots into the other position the conduits **108** and **138** exchange their function.

The motor assembly **110** is shown in section in FIG. **13.** Each of the motors **130** and **132** includes a drive flange **141** and **140** respectively to which are articulated piston rods **142.** Pistons **144** are disposed on the piston rods **142.** The pistons are guided in a cylinder block **146.** The cylinder block **146** is rotatably mounted on a trunnion **148** held in a pivotable frame **150** and abuts against a stationary valve portion **152.**

The concurrent rotation of the cylinder block and the drive flange is effected by a bevel drive with teeth **154, 156** on toric surfaces. The toric surfaces, in this case, have the centers **158, 160** in the plane of engagement and intersection. Pivoting of the pivotable frame and the cylinder block **146** is effected about an off-centre axis in the form of a rolling movement of the toric surfaces on each other, that is such that the dead space in the upper dead denter of the cylinder **144** is kept as small as possible.

The drive flanges **141, 140** of the two motors **130, 132** are mounted with their hubs **162, 163** on a shaft **164.** These hubs are supported in an axial direction through a sliding disc **166** that teat the axial components, acting against each other, of the hydraulic forces acting on the drive flanges **140, 141** can be absorbed at the sliding disc. The dotted lines **168, 169** indicate the transmission of this axis thrust. The dotted lines **170, 171** show how the residual thrust is conducted to the housing.

Teeth **172** and **173** on the hubs **162** and **163** respectively are in mesh with gears **174** and **175.** Gears **174** and **175** are each disposed on a half shaft **176** and **177** respectively which are each in drive connection with a drive wheel of a pair of drive wheels.

FIG. **14,** which illustrates a section along the lines X—X of FIG. **13,** shows how the pivotable frames **150** are mounted on the housing and how this bearing support is utilized for the oil supply. A connecting side bar **178** has two side-by-side bores in which are bearing trunnions **180** and **182.** Trunnion **180** is secured to the housing and trunnion **182** is secured to the pivotable frame. Pressure fields are limited by annular rings **184** and **186** or **188** and **190** respectively seated in grooves on the outer surface of the bearing trunnions. The annular rings **184** and **186** or **188** and **190** respectively are positioned in oppositely inclined planes in the trunnion axes, so that in each case pressure fields enlarging outwards to one side results. These pressure fields relieve the bearing from the hydraulic forces acting thereon at the pivoting frame. The pressure fields on the outer surface of the trunnions **180** and **182** are interconnected through a channel **192** in the connecting side bar **178.** An oil feed channel **194** in the trunnion **180** is connected to the pressure side of the motor **130** or **132** respectively through the channel **192** and through a conduit **196** in the trunnion **182.** This trunnion bearing is thus utilized for the introduction of the

pressure oil and simultaneously produces pressure fields to relieve the bearing of the hydraulic forces. In order to ensure the pivoting of the pivotable frame **150** in the form of a rolling movement of the toric surfaces having the teeth **154** and **156,** there is additionally provided an angle-bisecting gearing which consists of a guide surface **198** on which a projection **200** of the angle-bisector of the curved surface **204** between the drive flange and the cylinder block axis abuts. On surface **204** a projection **206** abuts a correspondingly curved surface on the pivotable frame. This arrangement substantially corresponds to FIG. 3 of my copending patent application for "AXIAL PISTON TYPE MACHINE" filed Nov. 18, 1971 Ser. No. 199,974.

Also in such a drive unit arrangement the conditions with regard to the motor design and pivoting of the motor about an off-center axis described in the other embodiment are realized. Several such motor assemblies **110, 112** can be driven by a pump **104.** In this case there is the possibility of pivoting one of these motor assemblies back to zero if necessary. A differential effect can act through the common pressure input conduits of the motors by the separate drive of the hubs **162, 163,** and thrust accommodation remaining undistrubed.

FIG. **15** shows the composition of the pressure conduit of the motors **130, 132.** The pressure oil entering in the connection **210** is, in this case, conducted through the two openings **208** into the channels **194** (FIG. **14**).

The adjusting cylinders **134, 136** are so arranged that the force effective therein during the return of the motors **130, 132** decreases their spacing to the ideal pivot point **156, 158, 160** so that the adjusting pressure increasing in the adjusting cylinders **134, 136** forces a pivoting of the same magnitude of the motors **130, 132** and so on.

I claim:

1. In a hydrostatic drive unit comprising a driven shaft, a drive shaft, a rotary hydraulic pump connected to the drive shaft and producing an output of a given volume of hydraulic fluid per revolution, at least one axial piston-type motor communicating with the pump to be supplied with hydraulic fluid from the pump, said motor including a cylinder block, pistons in said cylinder block, a drive flange rotatable about an axis, piston rods connecting the pistons and the drive flange, and means supporting the cylinder block for rotation and for pivotal movement about a pivotal axis whereby the hydraulic fluid intake volume of the motor is varied by changing the pivotal position of the cylinder block, there being a dead space in the cylinders when the pistons are fully seated therein, means connecting the drive flange to the driven shaft, the improvement comprising:

in normal operation said pump producing said volume with speed variation being accomplished by pivoting said cylinder block of the motor, said means supporting said cylinder block establishing a maximum angle of pivot of the cylinder block at which the intake volume of the motor is a multiple of said given volume of the pump, said pivotal axis being offset from said rotational axis so that the dead space in the cylinders is kept as small as possible.

2. A hydrostatic drive unit according to claim **1,** characterized in that the cylinder block of the axial piston-type motor is pivotable up to an angle of pivot of greater than 30°.

3. A hydrostatic drive unit as set forth in claim **2,** wherein the cylinder block and the drive flange have mutually engaging teeth so that they rotate concurrently, said point of engagement of said teeth being in the region of the pivotal axis.

4. A hydrostatic drive unit as set forth in claim **3** including a housing, wherein the cylinder block and the drive flange have toric surfaces on which said teeth are mounted, said means supporting the cylinder block includes a frame, a bearing trunnion on the side of the housing, a bearing trunnion on the side of the frame and a connecting side bar having bores within which said trunnions are received, and including angle bisecting gearing means interconnecting the housing and the frame.

5. A hydrostatic drive unit as set forth in claim **4** including a hydraulic fluid conduit between the pump and the motor, characterized by part of said conduit extending through said trunnions and through a channel in the connecting side bar between said bores therein, said trunnions having annular rings thereon at each side of said channel with said rings defining pressure fields about the trunnions.

6. A hydrostatic drive unit as set forth in claim **5,** wherein each trunnion is formed about an axis and the respective rings are in planes inclined in opposite directions with respect to the respective trunnion axis so that the pressure field of a trunnion is larger at one side of the trunnion axis than at the other side, and other side being adjacent the other trunnion.

7. A hydrostatic drive unit as set forth in claim **6,** including a second axial piston type motor communicating with said pump, said second motor including a cylinder block, pistons in said cylinder block, a drive flange rotatable about an axis, piston rods connecting the pistons and the drive flange, and means supporting the cylinder block for rotation and for pivotal movement about a pivotal axis whereby the hydraulic fluid intake volume of the second motor is varied, there being a dead space in the cylinders of the second motor when the pistons are fully seated therein, means connecting the drive flange of the second motor to the driven shaft, said drive flanges being coaxial with respect to each other and rotatable independently of each other, bearing means between the adjacent sides of the flanges, the two cylinder blocks being at the nonadjacent sides of the respective flanges whereby the forces of the piston rods on the flanges act to force the flanges toward each other, said means connecting the drive flanges to the driven shaft including drive wheels secured to the respective flanges.

8. A hydrostatic drive unit as set forth in claim **1,** including a second axial piston type motor communicating with said pump, said second motor including a cylinder block, pistons in said cylinder block, a drive flange rotatable about an axis, piston rods connecting the pistons and the drive flange, and means rotatably supporting the cylinder block for rotation and for pivotal movement about a pivotal axis whereby the hydraulic fluid intake volume of the motor is varied, there being a dead space in the cylinders when the pistons are fully seated therein, means connecting the drive flange of the second motor to the driven shaft, said drive

flanges being coaxial with respect to each other and rotatable independently of each other, bearing means between the adjacent sides of the flanges, the two cylinder blocks being at the non-adjacent sides of the respective flanges whereby the forces of the piston rods on the flanges act to force the flanges toward each other, said means connecting the drive flanges to the driven shaft including drive wheels secured to the respective flanges.

9. A hydrostatic drive unit as set forth in claim **1**, including a housing, said means supporting said cylinder block comprising a frame, a bearing trunnion on the side of the housing, a bearing trunnion on the side of the frame and a connecting side bar having bores within which said trunnions are received, including a hydraulic fluid conduit between the pump and the motor with a part of said conduit extending through said trunnions and through a channel in the connecting side bar between said bores therein, said trunnions having annular rings thereon at each side of said channel with said rings defining pressure fields about the trunnions, each trunnion being formed about an axis and the respective rings being in planes inclined in opposite directions with respect to the respective trunnion axis so that the pressure field of a trunnion is larger at one side of the trunnion axis then at the other side, said other side being adjacent the other trunnion.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,775,981 Dated December 4, 1973

Inventor(s) Hans Molly

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Col. 1, line 38	after "high" insert --oil--; after "pressure", delete "pressur"
Col. 2, line 43	after "from I to II" insert --are sometimes not provided as--
Col. 3, line 9	after "per", "roata-" should be "rota-"
Col. 3, line 27	"ocntinuously" should be "continuously"
Col. 4, line 13	"bearing" should be "bearings"
Col. 4, line 14	"dirven" should be "driven"
Col. 4, line 17	"chennel" should be "channel"
Col. 4, line 22	"qunatity" should be "quantity"
Col. 4, line 47	"off center" should be "off-center"
Col. 5, line 61	after "which is" insert --biased--
Col. 6, line 28	"denter" should be "center"
Col. 6, line 33	"that teat" should be "so that"
Col. 7, line 27	"trubed" should be "turbed"
Col. 8, line 33	"and" should be "said"

Signed and sealed this 30th day of April 1974.

(SEAL)
Attest:

EDWARD M.FLETCHER,JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents

FORM PO-1050 (10-69)

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