

Hydrostatic rotary piston machine

The invention relates to a hydrostatic rotary piston machine of the type stated in the precharacterising clause of Claim 1. These hydrostatic machines can be used both as a fluid power pump and, preferably, as a fluid power motor and are particularly popular as low-speed "torque motors". The working fluids are understood as being liquids and gases. Its advantage is in particular a relatively large absorption volume per revolution and hence a relatively high operating torque. Embodiments according to the precharacterising clause have the advantage that the shaft can be mounted to the left and right of the displacement part and of the control part in roller bearings having large dimensions, so that not only is there exact mounting for the fluid power part but also a large bearing spacing is realised, permitting large radial forces at the drive shaft end or output shaft end - owing to the large lever action of the shaft. Not only does this permit large belt and tooth forces for torque induction and dissipation in these machines but these machines can also be used as a drive hub for hydrostatic wheel drives.

A known machine of this type (DE-A-1,703,573) has a so-called rotor tooth system between the stationary housing and the outer tooth system of the rotary piston. This tooth system operates there as the displacement part. The rotary piston likewise has a rotary tooth system in its inner region, its inner rotor being rotatably connected to the driven shaft or output shaft to form a single piece. In this machine, an attempt is made to ensure supply to and removal from the displacer tooth system via control slots which are arranged in the rotary piston itself. For reasons

relating to design and gear kinematics, it is necessary for the eccentricities of both rotor tooth systems to be the same. The tooth height of the displacer tooth system thus depends on the tooth height of the very much smaller tooth system on the shaft, so that the delivery area, i.e. the specific volume per rotation of the displacer tooth system is still relatively small. Furthermore, the achievable flow cross-sections through the commutator control provided there are disadvantageous, so that high throttle losses occur.

Rotary piston machines which have generic differences have also been disclosed, for example US-A-3,288,078, which describes an output shaft mounted at one end. Such solutions are not possible for high force transmissions since the relevant shaft or its bearing, but in particular the relatively fine tooth system between rotor and drive shaft, is scarcely suitable for transmitting sufficient forces. US-A-3,658,449 has disclosed a hydrostatic rotary piston machine which also dispenses with a central shaft mounted at both ends and instead has a tumbling clutch shaft with spline. Such arrangements have the disadvantage that they must be constructed with a small diameter and therefore can transmit only relatively small forces and are subject to great wear, especially since the tooth profiles of the tumbling shaft each only partly engage the corresponding tooth profiles.

It is the object of the invention to provide a hydrostatic rotary piston machine of the stated type, in which the above-mentioned disadvantages do not occur. In particular, it is intended to increase the absorption volume and to propose a hydrostatic rotary piston machine in which as many parts as possible can be produced by very efficient methods, for example by

the sinter method. Furthermore, the number of parts required is to be kept as small as possible. In particular, the axially shapable parts, for example the control housing 38, are sintered. This is achieved, according to the invention, by the combination of the features of Claim 1.

Expedient embodiments are defined by the features of the subclaims.

According to the invention, twice the tooth height is obtained in the displacer tooth system, which is referred to below as the first inner or outer tooth system, with a difference of 1 in the numbers of teeth if the tooth system provided between the rotary piston and the shaft, which system is referred to below as a second inner and outer tooth system, has a difference in the number of teeth which is greater than 1. However, this consequently achieved substantially larger absorption volume of the displacer tooth system requires a control which is intensive in terms of the time cross-section (in cm^2/sec) for the inflow and outflow of the operating medium, and a separate rotary commutator must therefore be provided. Since the rotary piston transmits its torque to the shaft via very large tooth forces, this shaft must be very stable. Since this thick shaft must be passed through the rotary commutator, a new path must be taken by the rotary piston for driving said commutator. This is probably also the reason why so far those skilled in the art have regarded a larger difference in the number of teeth as impossible.

This object is now achieved in an extremely advantageous manner if the second inner and outer tooth system has a difference in the number of teeth of at least 2 and if the rotary commutator of the control

part is coupled to the rotary piston via an arc gear having a transmission ratio of 1 : 1. The rotary commutator is freely rotatable with respect to the shaft.

5 An inner tooth system having concave tooth flanks which are round and which determine the shape of the tooth flanks of the second outer tooth system on the driven or output shaft by rolling on the second inner tooth system of the rotary piston is suitable as
10 a possible tooth form between the rotary piston and the shaft. Such an inner tooth system has particularly small slide components, owing to a very small angle of engagement.

 The efficiency of the inner gear between the
15 rotary piston and the shaft can be further improved if the second inner tooth system (of the rotary piston) has convex round tooth flanks and the shape of the tooth flanks of the second outer tooth system (of the shaft) is determined by rolling on the second inner
20 tooth system and thus has a concave tooth shape. Consequently, the notch-free cross-section of the rotary piston is also larger than in the variant having concave flanks in the second inner tooth system, which gives greater stability and permits a narrower
25 construction.

 Satisfactory control of the rotary commutator for supply to and removal from the displacement part requires, as is generally known for such machines, that the rotary commutator executes exactly the same number
30 of revolutions as the rotary piston about its own axis. Since the rotary piston performs not only a rotary movement but also an eccentric movement, design difficulties arise in the case of this 1 : 1 transmission. In the case of the rotary piston machine

according to the invention, this 1 : 1 transmission is achieved by the rotary commutator having gear extensions which point toward the displacement part and intermesh directly with the second inner tooth system of the rotary piston. The 1 : 1 transmission arises by virtue of the fact that an arc gear having the transmission 1 : 1 is provided between the rotary commutator and rotary piston, in which arc gear the gear extensions projecting from the rotary commutator are in the form of teeth which have round tooth flanks, are distributed uniformly along an arc and engage round teeth of the rotary piston as a second inner tooth system, the radius of which is smaller, by a factor corresponding to the eccentricity of the rotary piston machine, than the radius of the round tooth flanks of the gear extension, or vice versa. High powers are not transmitted.

Conversely, the tooth system between the rotary piston and the output shaft must be in the form of a rolling tooth system with very small slide components in order to minimise the loss. However, since the rotary commutator operates in a virtually moment-free manner, a gear having a relatively high slide component can be used for driving it, as is the case with the arc gear. For example, a clutch as in the case of the cycloid gear may be provided. For the purposes of the invention, the content of Swiss Patent Application "Arc gear" filed on the same date is considered to have been disclosed. In the case of an embodiment of the second inner tooth system having concave teeth, the 1 : 1 drive of the rotary commutator is now possible by virtue of the fact that the gear extensions of the rotary commutator which are uniformly distributed around the circumference project directly into the

second inner tooth system of the rotary piston, which system possesses concave round tooth flanks, and the number of extensions is equal to the number of teeth of the second inner tooth system, the shape of the extensions being determined in accordance with the guidelines for an arc gear according to Claim 7 and having convex tooth flanks. A sufficient degree of overlap can be achieved for constant 1 : 1 transmission of the angle of rotation from the rotary piston to the rotary commutator.

If the rotary piston has, as a second inner tooth system, teeth having convex tooth flanks, the rotary commutator is then driven by virtue of the fact that the gear extensions of the rotary commutator which are uniformly distributed over the circumference likewise directly project into the second inner tooth system of the rotary piston, which system is provided with convex round tooth flanks; and the number of the extensions is equal to the number of teeth of the second inner tooth system, the extensions once again being determined in accordance with the guidelines for an arc gear according to Claim 6 and having concave tooth flanks.

In the case of rotary piston machines which are similar but are not of the type, it is known that the rotary commutator is driven via a gyratory cardan shaft from the rotary piston. However, this solution cannot be adopted in a machine according to the invention since the driven shaft is passed through the rotary commutator in the central region of the machine. It is true that it could be possible to provide a gyratory hollow shaft in the region between the rotary commutator of the control part, which hollow shaft is provided at its two ends with a third and fourth outer

tooth system which intermeshes at one end with the second inner tooth system of the rotary piston and at the other end with a third inner tooth system of the rotary commutator. However, this possibility is not feasible in practice since the through driven or output shaft in this region would have to be made very much thinner, with the result that, in the case of the same dimensions, inadmissible sagging of the gear shaft might occur.

The design of the displacer tooth system, in this case therefore the first inner or outer tooth system, has a substantial effect on the efficiency of the machine. Here, one of its sources of loss is the normal force with which the tooth tips of the first outer tooth system are pushed against the tooth tips of the associated inner tooth system. This tip-tooth force is smaller the smaller the angle of engagement of the tooth system. Since these tooth tips slide on one another, frictional losses occur there and may also simultaneously lead to signs of wear. A design which has proved its worth often in practice is characterised in that this first inner or outer tooth system is a trochoidal tooth system where, as described in another context (cf. European Patent 43,899, which is considered to have been disclosed for the purposes of this description), the teeth of the first inner tooth system have an approximately trapezoidal shape with convex flanks and tips, and the pitch circle of the first inner tooth system runs outside the circle about the centrepoint of the first inner tooth system through the lower third of the tooth height of the first inner tooth system.

In the case of high speeds of this hydrostatic rotary piston machine, an embodiment which has also

proved suitable is one in which the teeth of the first inner tooth system are formed by rollers rotatably mounted in the housing. These are mounted in the housing with a certain play in the plane bearing, so that a hydrodynamic plane bearing is formed by the operating fluid between the roller and the housing.

In the case of the type of rotary piston machines on which the invention is based, the rotary piston has an annular shape. The hydrostatic force acts on half its outer circumference and attempts to deform this annular body to an oval. This deformation must not be greater than permitted by the tooth play of the first inner tooth system if the rotary piston is still to rotate freely in the inner tooth system of the housing. If this oval deformation becomes too large, the rotary piston jams, resulting in poor efficiency and a high degree of wear of the machine. For this reason, the deformation resistance of the rotary piston must be optimised. This is achieved when the inner tooth system of the rotary piston has the same number of teeth as its outer tooth system and when the rotary piston is produced from a material having a large modulus of elasticity.

Further advantageous possibilities for efficient production of the machine according to the invention are described in the subclaims.

The invention is illustrated in detail below on the basis of embodiments, with reference to the attached, schematic drawings.

Fig. 1 shows a longitudinal section through an embodiment of a hydrostatic rotary piston machine, for greater clarity only the longitudinal pin connection but not the longitudinal screw connection being shown;

Fig. 2 shows a cross-section along the line A-A of Fig. 1, the inner tooth system of the rotary piston having concave tooth flanks;

5 Fig. 3 shows an identical cross-section, the inner tooth system of the rotary piston having a convex tooth flank shape;

10 Fig. 4 shows a cross-section along the line B-B of Fig. 1, the inner tooth system of the rotary piston having concave tooth flanks, as in Fig. 2;

Fig. 5 shows an identical cross-section, the inner tooth system of the rotary piston having a convex tooth flank shape;

15 Fig. 6 shows a cross-section along the line A-A of Fig. 1, the inner tooth system of the displacer tooth system in the housing being formed by cylindrical rollers;

Fig. 7 shows a cross-section along the line C-C of Fig. 1 and

20 Fig. 8 shows a variant having a control part arranged in the centre.

25 The rotary piston machine 101 shown in the Figures has, in addition to the longitudinal screw connection not shown in the longitudinal section, a driven or output shaft 9 which is mounted in a stable manner in two tapered roller bearings 10 on the left and right of the hydraulic part. The machine is sealed leak-tight from the outside by a rotary shaft seal 50, for the sake of clarity the leak oil lines serving for pressure relief of the seal and the leak oil return lines into the low-pressure region not being shown.

30 The shaft 9 is provided with a powerful outer tooth system 8 (8a having convex and 8b having concave tooth flanks 28a and 28b, respectively) which transmits drive

torque or output torque and intermeshes with the inner tooth system 7 (7a having concave and 7b having convex tooth flanks 29a and 29b, respectively) of the rotary piston 5. This revolves with the eccentricity e about the shaft 9 and, since the shaft is mounted coaxially with the inner tooth system 4 of the housing, also inside the housing 3. It is therefore necessary to meet the design requirement that the axial spacing of the toothed inner gear between the shaft 9 and the rotary piston 5 must be equal to the axial spacing of the inner gear between rotary piston 5 and housing 3. The machine furthermore has a drum-like rotary commutator 11 which is mounted pressure-tight in the control part 2 but is provided with play. It has control slots 12 and 13 which open radially outwards, are staggered axially and are distributed uniformly around the circumference. The control slots are connected to the connections 19 and 20 for the delivery medium both via circumferential grooves 15 and 16 and via inner grooves 17 and 18. The mode of action of such a rotary commutator for controlling, for example a rotary piston machine of the generic type is known to the relevant person skilled in the art (cf. OMM Hydromotor from Danfoss) and therefore need not be explained further here. The rotary commutator supplies pressure media to, and removes said media from, the displacement part 1 via the radial control channels 21 and 22 and via the axial channels 23.

As can be seen from Fig. 2 to 6, the channels 23 open into the tooth spaces 26 of the inner tooth system 4 of the housing, which, together with the associated outer tooth system 6 of the rotary piston 5, form the operating spaces of the hydrostatic machine in a known manner. The mode of action of these known

pumps or motors having internal gears is also known to a person skilled in the art and need not be explained further. With correct control of the infeed and outflow of the operating medium from the operating spaces 26a and 26b through the rotary commutator 11, for example, all operating spaces 26a to the left of the centre distance line 40 are connected to the infeed 19 and all operating spaces 26b to the right thereof are connected to the outflow 20. If, as in the case of a fluid power motor, the infeed 19 is at a high pump pressure and the outflow 20 is approximately at atmospheric pressure, the rotary piston 5 is rotated in the clockwise direction with high torque about the engagement point 27 of the tooth system 4 of the housing in the example of Fig. 2 to 6. The magnitude and the uniformity of this torque depends decisively on the number of teeth and on the arc diameter of the outer tooth system of the rotary piston. This gives rise to the linear relationship between the absorption volume of the machine per revolution of the rotary piston about its own axis and its torque. A large number of teeth and great eccentricity e therefore result in high power of the machine for a given size.

The rotary piston 5 now transmits its torque in the form of a high tooth force at the engagement point 44 between its inner tooth system 7 and the outer tooth system 8 of the shaft to the output shaft 9.

The efficiency of this force transmission between the rotary piston and the shaft is influenced by the angle of engagement of the engaged tooth systems. The tooth system according to Fig. 3 is superior to that of Fig. 2 by about 4% if its design has been optimised. This optimisation must be performed on the drawing board and also by calculation,

which need not be described in detail here and is known to a person skilled in the art.

5 What is important for successful operation of such a rotary piston machine is a rigid shaft 9, and it is therefore necessary to ensure that the outer tooth system 8 mounted on it preferably as a single piece has as large a diameter as possible. At the same time, however, the rotary piston 5, too, should be made as rigid as possible. Particularly in Fig. 6, it is
10 evident that it is advantageous if the inner tooth system of the rotary piston 5 has the same number of teeth as its outer tooth system 6.

 As is evident in particular in Fig. 1, little space remains for the 1 : 1 drive of the rotary
15 commutator 11 by the rotary piston 5. In the rotary piston machine according to the invention, a completely new approach has been adopted for achieving this object, as can be seen particularly clearly in Fig. 4. If an arc shape is chosen for the inner tooth flanks
20 29a of the rotary piston 5, it is also possible to find a corresponding round tooth 30 (having a convex tooth flank 30a) for 1 : 1 transmission between rotary piston 5 and rotary commutator 11. The active points of engagement are denoted by the numbers 31 and 32. The
25 method for the calculation and construction of this 1 : 1 tooth system is described in Patent (Swiss Patent Application "Arc gear" filed on the same date) and in the features of Claim 5. The extensions 14 in the form of teeth and having the round tooth flank 30a
30 can be produced as a single piece with the rotary commutator 11, for example by the sinter process. Since the rotary commutator does not consume any power, the tooth load here is virtually zero.

 Fig. 5 shows such a round tooth system between

the rotary piston 5 and the rotary commutator 11, in which the tooth shape 29b is convex. The rules for construction and calculation of the counter-tooth flank 30b are identical. Substantially more stable tooth-shaped extensions 14b are obtained on the rotary commutator 11.

A machine having very high and well established wear resistance is shown in Fig. 6, in which the inner tooth system 4 of the housing is produced from rotatable, hardened and ground rollers 34. Here, it is possible for the rollers 34 to be hydrodynamically mounted on an oil film in the gap 35 between the roller and the housing, so that the efficiency of this displacer tooth system is improved. The cost of production is of course correspondingly higher since the lubricating film is permitted to be only a few micrometres thick, so that the internal shape of the housing must be correspondingly exact.

Fig. 7 shows the arrangement of the radial slots 12 and 13 of the rotary commutator 11, as well as the arrangement of the channels 21 and 22 and of the cylindrical channels 23 in cross-section. Here, the screws 36 of the longitudinal screw connection of the machine and the screw 37 for the separate screw connection of the control housing 38 to the connection housing 39 are also evident.

The variant represented in section in Fig. 8 shows the control part 2a with the connections 19a, 20a and the output end of the shaft 9 in more detail than in the case of the variant according to Fig. 1. This permits radial feed, and the radial bearing forces of the bearing 10 are even better distributed. This furthermore results in lower throttle losses at high speed because there are no pronounced curvatures.

Longer sealing distances "L" on the commutator and hence a better volumetric efficiency with the same construction length as Fig. 1; easier mountability and an arrangement of the connections which is more in conformity with the market. The further components correspond to those of the other Figures and are therefore not designated.

The embodiments shown in the drawing are intended merely as examples of a rotary piston machine according to the invention. Thus, it is also possible for flow through the rotary commutator to be axial instead of radial, the former being preferred in some cases. Furthermore, the infeed and outflow connections may be arranged - as is often preferred - not axially but radially.



CLAIMS

1. A hydrostatic rotary piston machine having a displacement part (1), acting as a driven part or output part, and an adjacent control part (2) which serves for supplying the displacement part (1) with operating fluid and removing the latter from the said displacement part, the displacement part (1) having a first rigid housing (3) with a first inner tooth system (4) which interacts with a rotatable, eccentrically arranged rotary piston (5) having a first outer tooth system (6), which rotary piston has a second inner tooth system (7) which intermeshes with a second outer tooth system (8) on a centrally mounted shaft (9) which also passes at least through the control part (2) and is mounted at both ends, the difference between the number of teeth of the first inner tooth system and outer tooth system (4, 6) being 1, characterised in that the second inner tooth system and outer tooth system (7, 8) have a difference of at least 2 in the number of teeth - the outer tooth system (6, 8) in each case being that with the similar number of teeth - and a rotary commutator (11) of the control part (2) being coupled to the rotary piston (5) via an arc gear (28, 29) having a transmission ratio of 1 : 1.
2. A hydrostatic rotary piston machine according to Claim 1, characterised in that the second inner tooth system (7a) has concave tooth flanks (29a) which are in particular round, and the shape of the tooth flanks (28) of the second outer tooth system (8a) on the driven shaft or output shaft (9) is determined for rolling on the second inner tooth system (7a) of the rotary piston (5a). (Fig. 2)
3. A hydrostatic rotary piston machine according to Claim 1, characterised in that the second inner



tooth system (7b) has convex, in particular round, tooth flanks (29b), and the shape of the tooth flanks (28b) of the second outer tooth system (8b) of the shaft (9) is determined for rolling on the second inner tooth system (7b). (Fig. 3)

4. A hydrostatic rotary piston machine according to any of Claims 1 to 3, characterised in that the rotary commutator (11) has gear extensions (14a; b) which point toward the displacement part (1) and intermesh directly with the second inner tooth system (7) of the rotary piston (5). (Fig. 1, 4)

5. A hydrostatic rotary piston machine according to any of Claims 2 to 4, characterised in that the gear extensions (14a; b) projecting from the rotary commutator are in the form of teeth which are distributed uniformly along an arc, have preferably round tooth flanks (30) and engage the second inner tooth system (7a; b), the radius of which is smaller, by a factor corresponding to the eccentricity (e) of the rotary piston machine, than the radius of the preferably round tooth flanks (30a; b) of the gear extensions (14a; b) or vice versa. (Fig. 4, 5)

6. A hydrostatic rotary piston machine according to Claim 2 or 5, characterised in that the number of gear extensions (14a) is equal to the number of teeth of the second inner tooth system (7a), the gear extensions (14a) having concave tooth flanks (30a). (Fig. 4)

7. A hydrostatic rotary piston machine according to Claim 3 or 5, characterised in that the number of gear extensions (14b) is equal to the number of teeth of the second inner tooth system (7b), the gear extensions (14b) having convex tooth flanks (30b). (Fig. 5)



8. A hydrostatic rotary piston machine according to any of Claims 1 to 7, characterised in that the first inner or outer tooth system (4, 6) is a trochoidal tooth system, each tooth (25) of the first inner tooth system (4) having an approximately trapezoidal shape with convex flanks and tips, and the pitch circle (41) of the first inner tooth system (4) runs outside the circle (42) about the arc centrepont (43) of the first inner tooth system (4) through the lower third of its tooth height. (Fig. 2)

9. A hydrostatic rotary piston machine according to any of the preceding Claims, characterised by one or more of the following features:

- a) the rotary piston (5), but preferably also the first housing (3) holding this piston, and the rotary commutator (11) and, if required, the control disc (36) and the second housing (19) are made of a sintered metal and/or ceramic powder;
- b) the inner tooth system (7a; b) of the rotary piston (5) has the same number of teeth as its outer tooth system (6a; b) and
- c) the teeth of the first inner tooth system (4) are formed by rollers (34) rotatably mounted in the housing (3).

10. A hydrostatic piston machine according to any of the preceding Claims, characterised in that, in the region of the rotary commutator, the control part (2) is divided coaxially into a control housing (38) and a connection housing (39) and has a pressure-tight screw union, the connection housing (39) preferably holding a bearing (51) for the shaft (9).

11. A hydrostatic rotary piston machine according to any of the preceding Claims, characterised in that the shaft input or output (13) transmitting the power



is closer to the displacement part (1) than to the control part (2). (Fig. 8)