United States Patent [19]

Nonnenmacher

[11] **3,910,162**

[45] **Oct. 7, 1975**

[54] **FORCE COMPENSATING ARRANGEMENT FOR A RADIAL PISTON MACHINE**

[75] Inventor: **Gerhard Nonnenmacher**, Korntal, Germany

[73] Assignee: **Robert Bosch G.m.b.H.**, Stuttgart, Germany

[22] Filed: **Apr. 23, 1973**

[21] Appl. No.: **353,625**

[30] **Foreign Application Priority Data**

Apr. 22, 1972 Germany 2219881

[52] **U.S. Cl.** **91/498**

[51] **Int. Cl.²** **F01B 13/06**

[58] **Field of Search** 91/484, 497, 475

[56] **References Cited**

UNITED STATES PATENTS

2,585,232	2/1952	Douglas	91/475
2,823,619	2/1958	May	91/498
3,037,488	6/1962	Barrett	91/498
3,188,918	6/1965	Bjorklund	91/497
3,192,867	7/1965	Bjorklund et al.	91/494
3,408,948	11/1968	Boyd	91/498

FOREIGN PATENTS OR APPLICATIONS

775,853	5/1957	United Kingdom	91/498

Primary Examiner—William L. Freeh

Attorney, Agent, or Firm—Michael S. Striker

[57] **ABSTRACT**

The rotor of a radial piston pump has cylinders, and pistons cooperating with an eccentric actuating ring so that the resultant of the inward directed forces passes through the axis of the actuator ring at an angle less than 90° to a plane through the axes of the rotor and of the actuator ring. The high pressure and low pressure control ports on the control pintle supporting the rotor, are angularly displaced relative to this plane so that the resultant of outward directed pressure forces, which passes through the rotor axis, is parallel to the resultant of the inwardly directed forces, and compensates the same.

1 Claim, 5 Drawing Figures

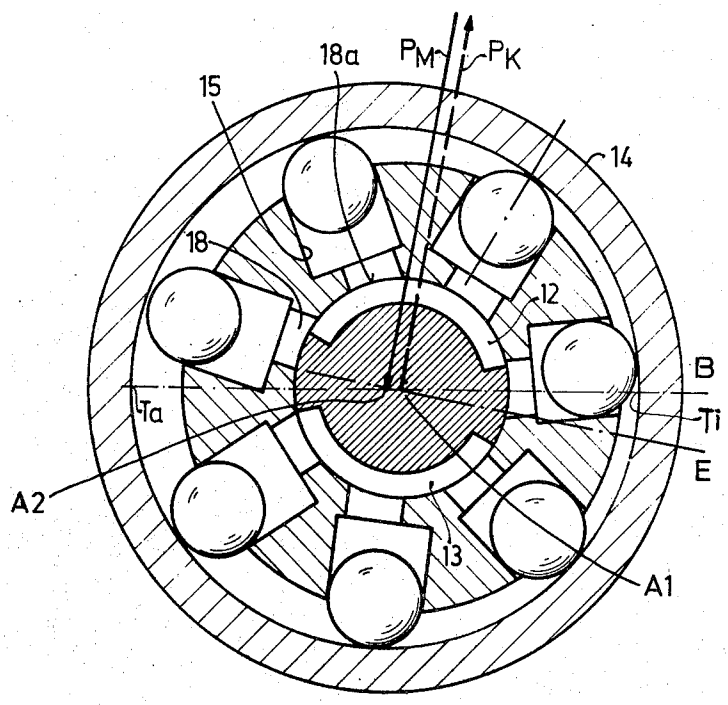


Fig. 1

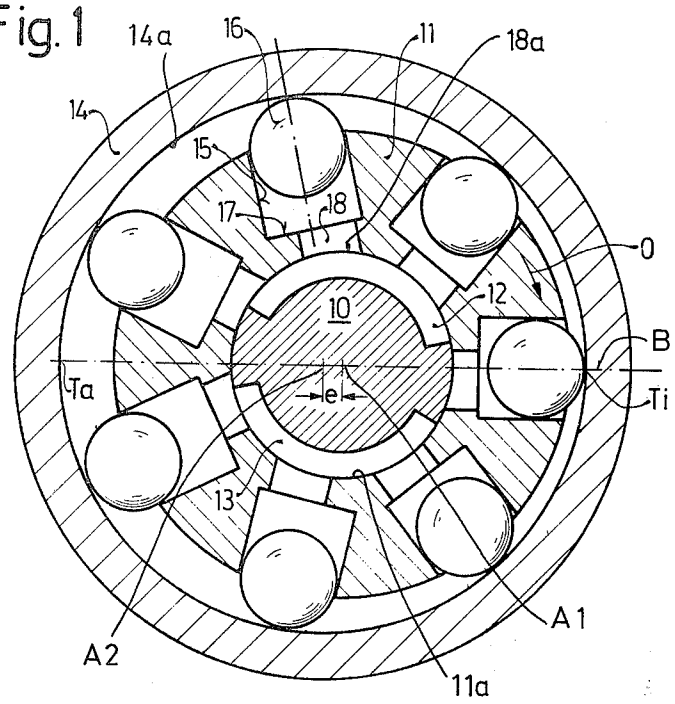

14a
16
11
18a
14
15
17
18
0
12
10
B
Ta
e
Ti
13
A1
A2
11a

Fig. 2

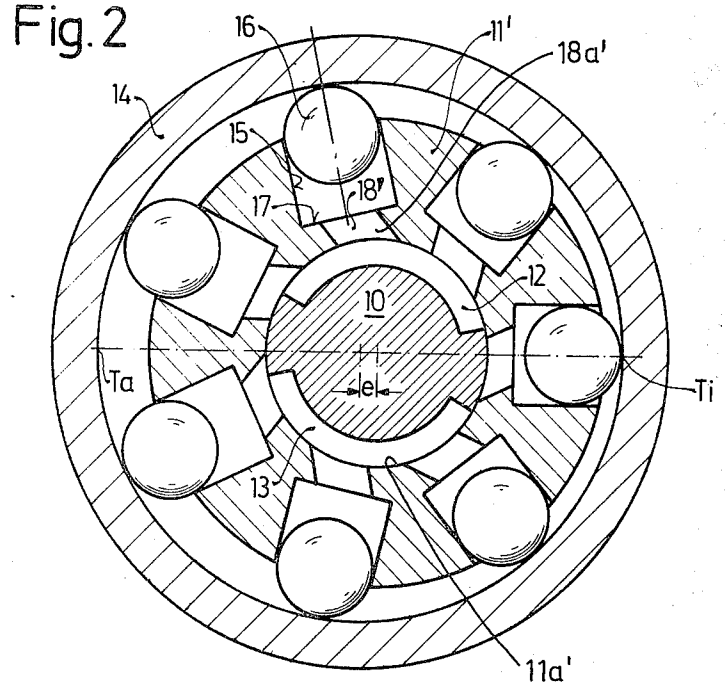

16
11'
18a'
14
15
17
18'
12
10
Ta
e
Ti
13
11a'

Fig.3

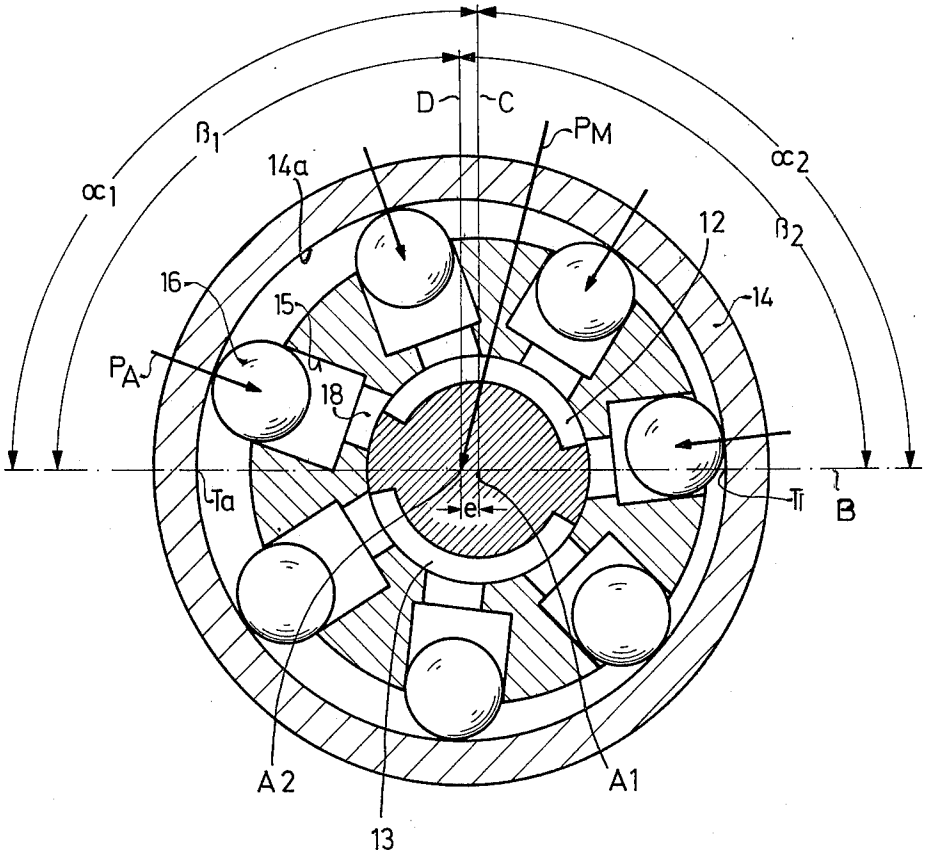


Fig.4

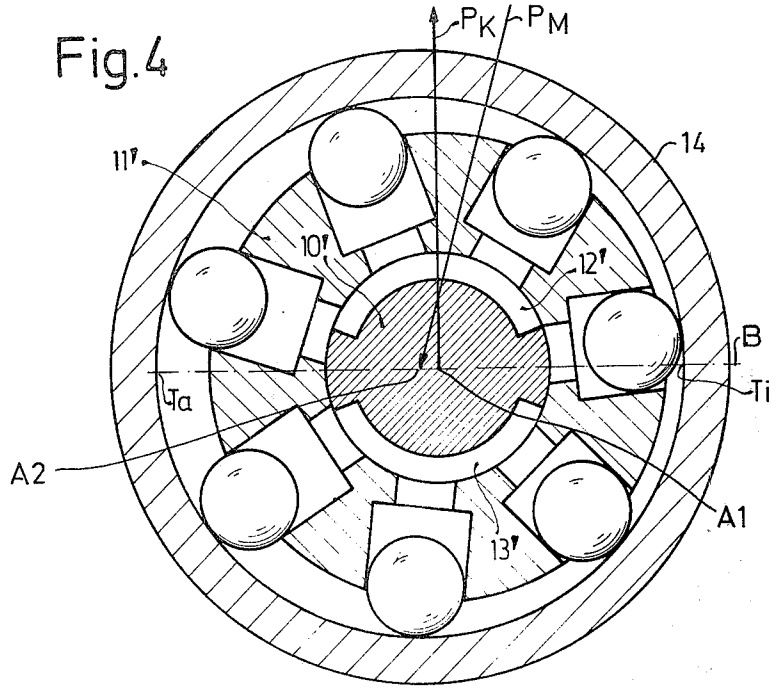
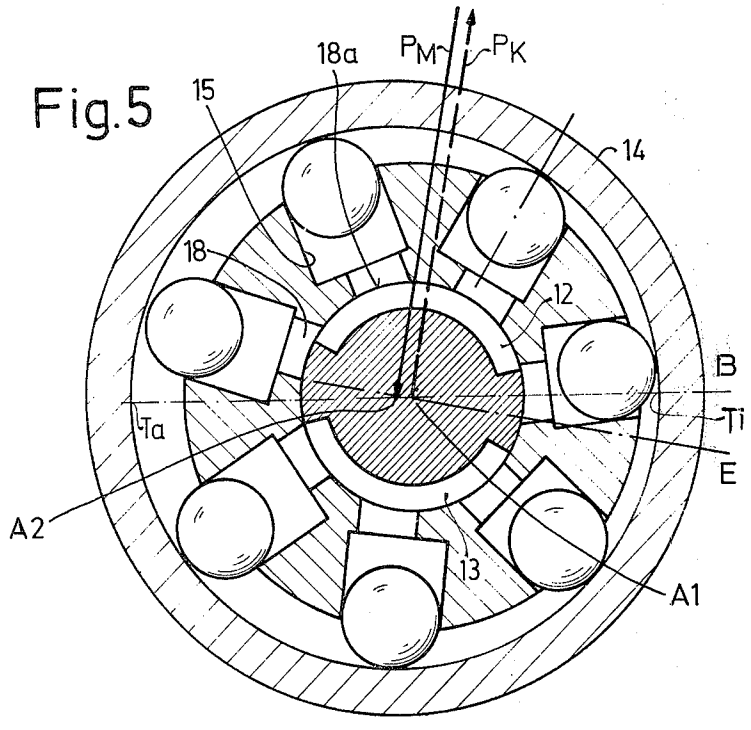

PK
PM
14
11'
10'
12'
B
Ti
Ta
A2
A1
13'

Fig.5


18a
PM
PK
15
14
18
12
B
Ti
Ta
E
A2
13
A1

FORCE COMPENSATING ARRANGEMENT FOR A RADIAL PISTON MACHINE

BACKGROUND OF THE INVENTION

The U.S. Pat. No. **3,064,583** discloses a radial piston machine, pump or hydraulic motor with a rotor mounted on a control pintle having part-circular high pressure and low pressure control ports cooperating during rotation with cylinder ports provided at the inner ends of passages which are connected with the cylinders, respectively. In machines of this type, forces pressing the rotor on the high pressure side against the control pintle are compensated by corresponding pressure areas, but it has been found that substantial forces occur in the direction of the eccentricity of the actuator ring, which are not compensated. This has the result that due to the one sided forces, increased wear of the parts occurs, so that the span of life of the machine is reduced.

SUMMARY OF THE INVENTION

It is an object of the invention to provide the radial piston machine in which all occurring forces are compensated so that the operation and span of life of the machine are improved.

With this object in view, the present invention provides a radial piston machine in which the cylinder ports on the inner surface of the rotor are circumferentially displaced to lead the cylinders during rotation while the machine operates as a pump, for example, while the high pressure and low pressure control ports on the pintle are angularly displaced relative to a plane passing through the axes of the rotor and of the actuator ring so that the high pressure control port is moved toward the inner dead center position of the rotor in which the rotor is closest spaced from the actuator ring.

In such an arrangement, the inner relieving pressure area on the control pintle is turned so that its direction conforms to the resultant of the forces acting on the bearings of the rotor.

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

BRIEF DESCRIPTION OF THE DRAWING

FIG. **1** is a schematic cross-sectional view illustrating a first embodiment of the invention;

FIG. **2** is a schematic cross-sectional view illustrating a second embodiment of the invention;

FIG. **3** is partially a cross-sectional view corresponding to FIG. **1**, and partially a diagram illustrating the inwardly directed forces produced by the eccentric actuator ring;

FIG. **4** is a cross-sectional view illustrating a radial piston machine according to the prior art; and

FIG. **5** is a cross-sectional view of the embodiment of FIG. **1**, and including a diagrammatic illustration of the compensation of the resultant of the inwardly directed mechanical forces by the resultant of outwardly directed pressure forces in accordance with the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

As shown in FIGS. **1**, **3** and **5**, a radial piston machine, which may operate as a pump, has a control pintle **10** surrounded by the inner cylindrical surface **11***a* of a rotor **11**, and supporting rotor **11** for rotation about an axis A_1. A stationary actuator ring **14** has an inner cylindrical surface **14***a* having an axis A_2 parallel with rotor axis A_1 and spaced therefrom the distance *e* representing the eccentricity of the actuator ring **14** in relation to the rotor **11**.

Rotor **11** has six cylinders **15** with radial axes, not shown, in which spherical pistons **16** are mounted for radial movement, and abut the inner surface **14***a* of the actuator ring **14**. During rotation in the direction of the arrow O, each piston **16** passes successively through the inner dead center point Ti and the outer dead center point Ta of plane B. Rotor **11** is closer to the actuator ring **14** at the inner dead center point Ti than at the outer dead center point Ta.

The cylinders **15** do not directly communicate with the control ports **12** and **13**, but are connected with the same by cylinder passages **18** which have inner ports **18***a* sweeping the control ports **12** and **13**.

In the embodiment of FIGS. **1**, **3** and **5**, the passages **18** are parallel to the radial axes of the respective cylinders which intersect the rotor axis A_1.

Assuming that the machine operates as a pump in the direction of the arrow O, the axes of the cylinder passages **18** lead the axes of the respective cylinders **17**.

In the embodiment illustrated in FIG. **2**, the rotor passages **18'** which open in ports **18***a'* on the inner surface **11***a'* of rotor **11'**, are slanted to the radial axes of the respective cylinders **15** so that the outer end of each rotor passage **18** is concentric with the bottom face **17** while the cylinder port **18'** leads the radial axis of the respective cylinder **15** in the same position as the cylinder ports **18**a in the embodiment of FIG. **1**.

When the machine is operated as a pump, rotor **11** or **11'** is driven in the direction of the arrow O. After passing through the inner dead center point Ti, the pistons **16** suck pressure medium through the low pressure control port **13** and the cylinder passages **18** into the radial cylinders **15**, and displace the pressure medium, after passing through the outer dead center point Ta, into the high pressure control port **12**. Between the rotor **11**, **11'** and the control pintle **10**, pressure areas develop due to pressure medium leaking out of the control ports **18***a*. The outwardly directed forces of the pressure areas, compensate the inwardly directed forces acting on the rotor **11** due to the fact that in accordance with the invention, the axial plane of symmetry E of the control ports **12** and **13** is angularly displaced from the plane B, particularly when the cylinder ports **18***a* and **18***a'* are displaced relative to the radial axes of the cylinders **15**, as will now be explained with reference to FIGS. **3** to **5**.

FIG. **3** illustrates the embodiment of FIG. **1**, but it will be understood that the diagram shown in FIG. **3** is also applicable to the embodiment of FIG. **2**.

Referring now to FIG. **3**, in the angular regions α_1 and α_2, which are bounded by the plane B and by a plane C perpendicular to plane B and intersecting the same in the rotor axis A_1, at any time the same number of pistons **16** is located due to the equal angular spacing of the axes of cylinders **15**. In the regions β_1 and β_2,

which are bounded by plane B and by plane D passing through the axis A_2 perpendicular to plane B, on the average, more pistons **16** are located in the region β_2 than in the region β_1. Since the inwardly acting mechanical forces P_A act always perpendicularly to the inner cylindrical surface **14***a* of the actuator ring **14**, the inwardly directed forces in the region β_2 exceed on the average the inwardly directed forces B_A in the region β_1. Accordingly the resultant P_M of the inwardly directed mechanical forces P_A in the pressure region is not perpendicular to the plane B, but defines therewith an angle less than **90°** in the region of the inner dead center point Ti.

FIG. **4** shows a radial piston pump according to the prior art with a rotor **11'** and high pressure and low pressure pressure ports **12'** and **13'** in the control pintle **10'** which are arranged symmetrically to the plane B which passes through the axes A_1 and A_2. In the machine of FIG. **4**, the outwardly directed forces of control pindle **10'** acting on rotor **11''** have a resultant P_K perpendicular to the plane B and passing through axis A_1. Consequently, the outwardly directly resultant P_K of the arrangement of the prior art, cannot fully compensate the resultant P_M of the inwardly directed mechanical forces, described with reference to FIG. **3.**

In the arrangement of the present invention as shown in FIG. **5**, the control ports **12** and **13** are displaced in circumferential direction so that the resultant vector P_K', which passes through the axis A_1, is substantially parallel to the resultant P_M of the inwardly directed mechanical forces. Consequently, the two opposite resultant forces P_M and P_K' fully compensate each other.

In order to prevent influencing of the timing of the flow into and out of the cylinder ports **18***a*, the ports **18**a are displaced relative to the cylinders **15.** The control ports **12** and **13** are geometrically arranged on opposite sides of a second plane E which passes through the rotor axis A_1 at an angle to the plane B. The control ports **18***a* may be displaced relative to the axis of the respective cylinders **15**, the same angular distance. The angle between the planes E and B may be the same as the angle between the plane D and the resultant force P_M.

As compared with the prior art construction shown in FIG. **4**, the high pressure control port **12** is angularly displaced toward the inner dead center point Ti, which is determined by the position of the actuator ring **14.**

During operation as a pump, the rotor **11** rotates in the direction of the arrow O shown in FIG. **1.** When the machine is to be used as a hydraulic motor, fluid under high pressure is supplied through the control pintle **10** to the high pressure port **12** which causes rotation of the rotor **11** in the opposite counterclockwise direction.

It will be understood that each of the elements described above, or two or more together, may also find a useful application in other types of piston pumps and hydraulic motors differing from the types described above.

While the invention has been illustrated and described as embodied in a force compensating arrangement for a radial piston machine in which control ports and cylinder ports are angularly displaced as compared with the conventional arrangement, it is not intended to be limited to the details shown, since various modifications and structural changes may be made without departing in any way from the spirit of the present invention.

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

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

I claim:

1. Force compensating arrangement for a radial piston machine, comprising a control pintle having an outer cylindrical surface formed with diametrically disposed circumferentially extending high pressure and low pressure control ports; a rotor having an inner cylindrical surface mounted on said outer cylindrical surface for rotation about a first axis, said rotor including equally angularly spaced cylinders having radial cylinder axes intersecting said first axis and cylinder ports on said inner cylindrical surface communicating with said control ports, and pistons in said cylinders; and an actuator ring surrounding said rotor and having an inner cylindrical actuator surface in sliding contact with said pistons and having a second axis, said first and said second axes being parallel and located in a first axial plane so that said rotor has in said first plane inner and outer dead center points closest to, and farthest from, respectively, said actuator surface whereby a resultant of inwardly directed forces passes through said second axis defining a predetermined angle less than **90°** with said first plane in the region of said inner dead center point, said high pressure and low pressure control ports being symmetrical to a second plane passing through said first axis and defining such an angle with said first plane so that the resultant of the outward directed pressure forces passing through said first axis is substantially parallel and equal to said resultant of said inward directed pressure forces for compensating the same, said cylinder ports being circumferentially displaced relative to said radial cylinder axes, respectively, an angle which is substantially equal to said angle between said first and second planes.

* * * * *