United States Patent [19]

Smith

[11] **3,793,923**

[45] **Feb. 26, 1974**

[54] **RADIAL PISTON HYDRAULIC MACHINES**

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[22] Filed: **Dec. 3, 1971**

[21] Appl. No.: **204,643**

[30] **Foreign Application Priority Data**

Dec. 5, 1970 Great Britain.....57878/70

[52] **U.S. Cl.** **91/488**

[51] **Int. Cl.** **F01b 13/06**

[58] **Field of Search** 91/486, 487, 488, 491, 492, 91/497, 498; 92/58, 157, 159; 308/73

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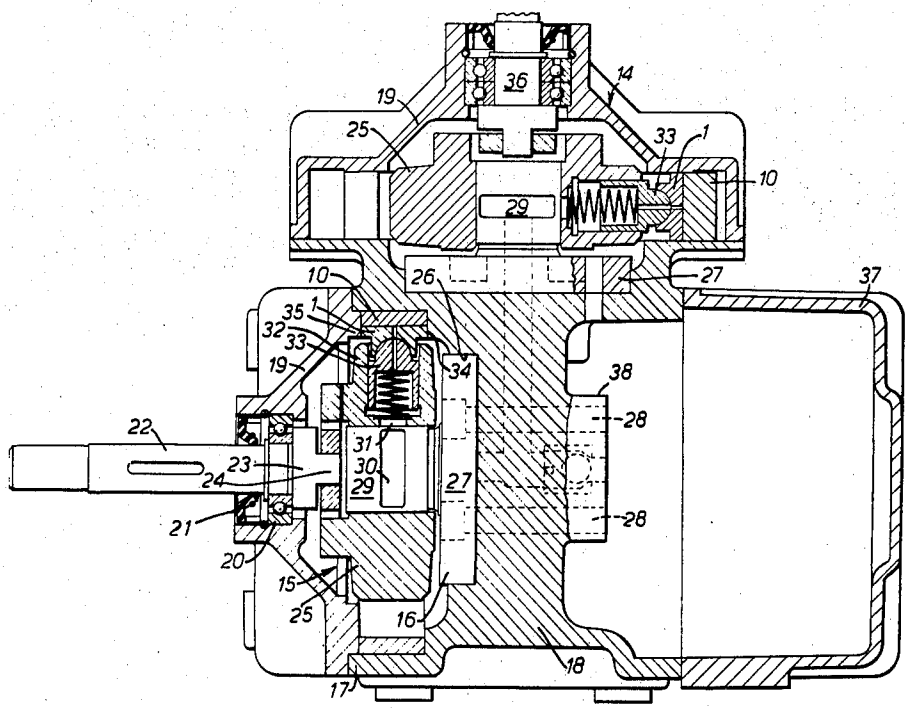
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Assistant Examiner—Gregory LaPointe
Attorney, Agent, or Firm—George Vande Sande

[57] **ABSTRACT**

A slipper and cam track for a radial piston hydraulic machine have engaging generally part-cylindrical and cylindrical surfaces respectively. The engaging slipper surface is relieved over its central area to give clearance between itself and the cam track, either by being of greater radius of curvature than the cam track or having a flat. The slipper bearing surface is also formed with oil pressure and relief grooving.

4 Claims, 5 Drawing Figures



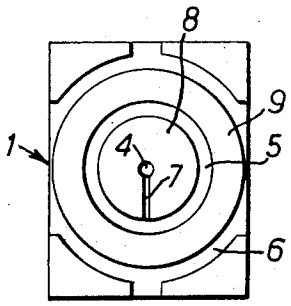


FIG. 1.

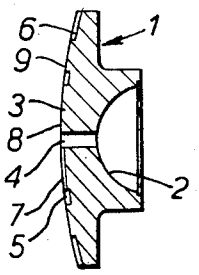


FIG. 2.

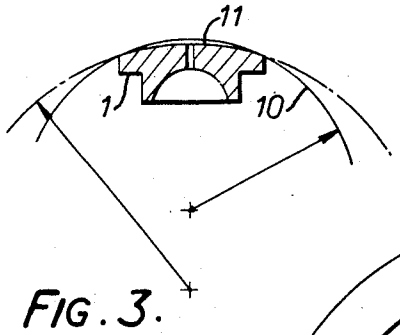


FIG. 3.

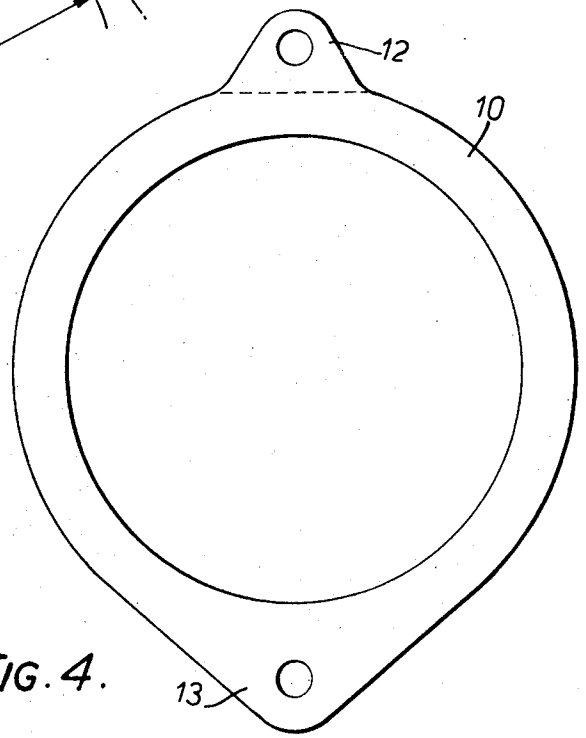


FIG. 4.

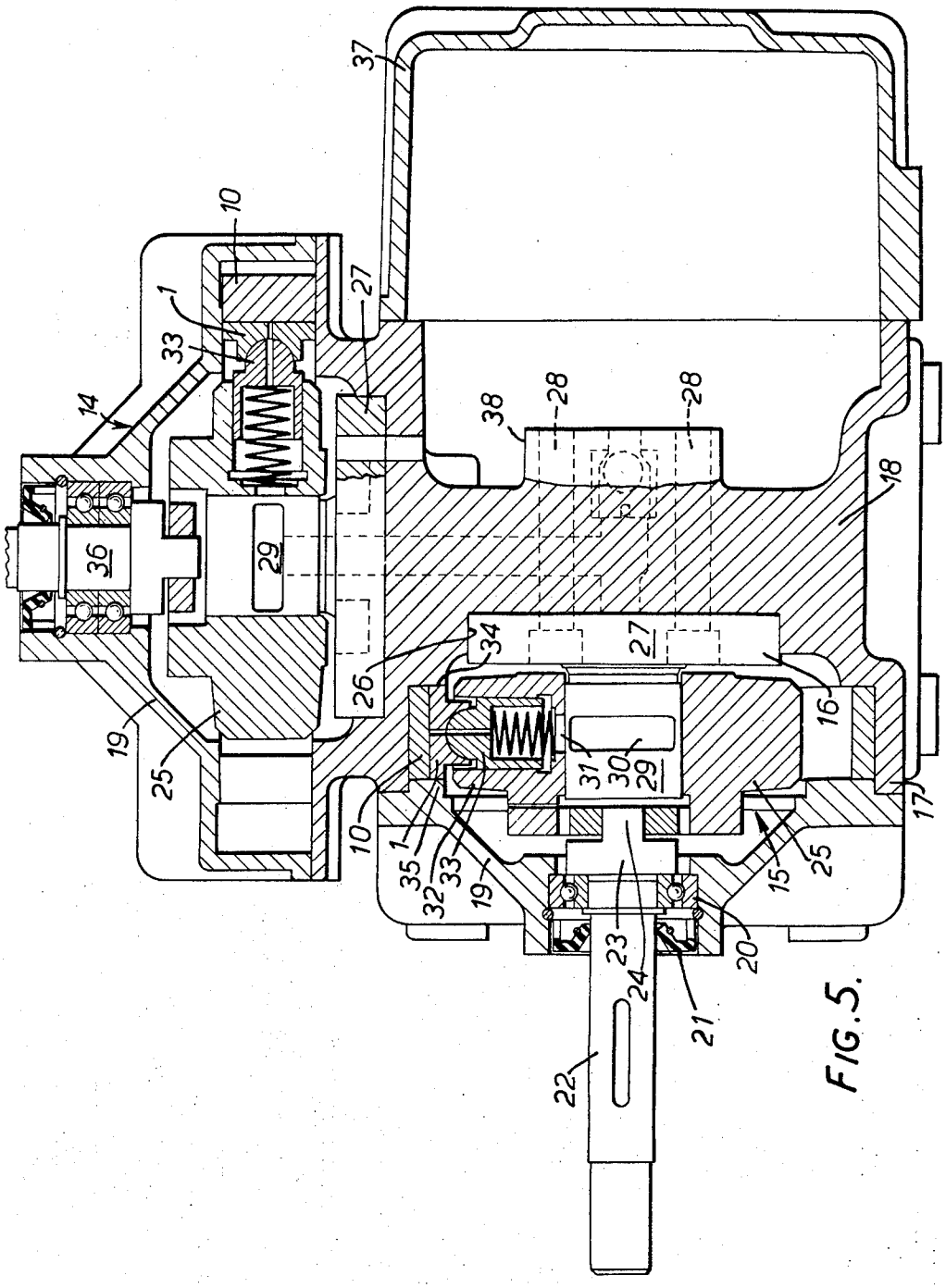

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FIG. 5.

RADIAL PISTON HYDRAULIC MACHINES

This invention relates to radial piston hydraulic machines of the type comprising a rotary member or assembly providing a number of generally radially extending cylinders each containing a radially movable piston whose outer end is connected to a slipper or bearing pad which slides on and engages a surrounding cam track or ring. The cam track is usually circular and eccentric with respect to the cylinder member so that rotation of the member is accompanied by reciprocating radial movements of the pistons, and vice versa.

It is, of course, important that the frictional resistance exerted between each slipper and the surrounding cam track should be kept at a minimum and the problem is augmented by the fairly substantial loads exerted when the fluid in the radial cylinders is under pressure, which may, for example, be up to 4,000 or 5,000 p.s.i. or more. The rotational speed may also be quite considerable, and it is an object of the present invention to provide an improved slipper having a bearing surface designed to co-operate with the surface of the surrounding cam track to reduce the frictional resistance.

Broadly the invention consists in a slipper or bearing pad for a radial piston hydraulic machine, such as a pump or motor, having a bearing surface of generally part-cylindrical form to engage the internal cylindrical bearing surface of a cam track or ring surrounding the pistons, the bearing surface of the slipper being relieved over its central area with respect to the cooperating cylindrical surface of the cam track to provide a slight clearance between the two surfaces.

The clearance at the central area of the slipper may vary in different applications and in different sizes of machine but generally speaking will probably be of the order of 0.0002 inches to 0.0004 inches.

The bearing surface of the slipper may be formed with a central "flat," to provide the clearance mentioned above, but in a preferred construction the bearing surface of the slipper has a radius of curvature which is slightly larger than the radius of the cam track. The radius of curvature of the slipper bearing surface may be, for example, approximately 0.005 inches greater than that of the cam track.

In any case the invention is of particular application where the slipper is formed with a central oil supply port communicating via the respective piston with the associated cylinder volume. Oil under pressure from the cylinder is supplied through this port to the surface of the slipper and fills the clearance volume formed by the special design of the bearing surface in accordance with the present invention.

In a particular preferred construction according to the invention the bearing surface of the slipper has a central land surrounded by a circular pressure groove, surrounded in turn by an annular land surrounded by a circular pressure relief groove.

The invention also consists in a radial piston hydraulic machine incorporating a number of slippers as defined above, and preferably each of the slippers in such a machine has a rectangular profile with generally flat opposed parallel flanks and is located between parallel annular guide walls on opposite sides thereof.

The invention may be performed in various ways and one specific embodiment will now be described by way

of example, with reference to the accompanying drawings, in which:

FIG. 1 is an end view onto the bearing surface of a slipper according to the invention,

FIG. 2 is a sectional elevation through the slipper of FIG. 1,

FIG. 3 is a diagrammatic illustration showing in a much exaggerated form the relationship between the slipper and the surrounding cam track,

FIG. 4 is an end view of one of the adjustable cam tracks, and

FIG. 5 is a sectional side elevation through a combined hydraulic motor and pump transmission unit incorporating bearing slippers according to the invention.

The slipper **1** illustrated in FIGS. 1 and 2 is formed of brass, bronze or similar bearing metal, preferably by a sintering operation, and has a part-spherical socket **2** in its rear face to engage the end of a ball joint (not shown) formed on the piston. The head **3** of the slipper has a rectangular profile as seen in FIG. 1. A central oil supply port **4** leads from the part-spherical cup **2** to the centre of the bearing surface and receives oil under pressure direct from the respective cylinder of the pump or motor as the case may be.

As seen in FIG. 1 the bearing surface of the slipper has two concentric annular grooves **5** and **6** surrounding the central delivery port **4**, the inner groove **5** communicating with this port via a single radial supply groove **7**, while the outer groove **6** is a relief channel and breaks into the four straight (in plan view) edges of the slipper head **3**. Oil supplied under pressure from the central port **4** to the inner groove **5** causes the central circular land **8** to act as a pressure area while the outer annular land **9** between the two grooves **5** and **6** acts as an oil throttling clearance so that the effective pressure is progressively reduced towards the outer relief groove **6**.

As illustrated diagrammatically in FIG. **3**, the bearing surface of the slipper as seen in cross-section, perpendicular to the rotary axis, is of somewhat increased radius of curvature relative to the surrounding cam track **10** and this has the effect of producing a clearance **11** over the central area of the slipper, whereas the leading and trailing edges are in relatively close engagement or contact with the surface of the cam track. The radial dimension of this clearance at the central area may be of the order of 0.0002 inches to 0.0004 inches. In the present instance where the radius of the cam track is approximately 1.6 inches this amounts to a difference of approximately 0.005 inches between the two radii of curvature, that of the slipper surface being greater than that of the cam track.

FIG. **4** is an end view of such a cam track **10**. It has a lug **12** at the top with a hole for entry of a control member and a lug **13** at the bottom, also with a hole by which it is pivotally mounted for swinging transversely to the axis to a selected eccentricity.

Such slippers and cam tracks may be employed in a hydrostatic transmission as shown in FIG. **5**, comprising a rotary radial piston pump **14** combined with a rotary radial piston motor **15**, with the pump axis perpendicularly intercepting the motor axis.

The pump and motor are essentially of similar construction with minor points of difference as described below and it is convenient first to describe the motor **15** in some detail.

The motor is housed in a casing defined partly by an end wall 16 and surrounding flange 17 forming part of a common central housing or mounting block 18 for the complete unit, and partly by a separate casing member 19 attached thereto. The casing member 19 has a central aperture provided with a bearing 20 and seal 21 through which extends a drive shaft 22. The inner end of the drive shaft 22 is provided with a flange 23 formed with a transverse slot or rib 24 forming part of an Oldham coupling designed to transmit torque between the end of the drive shaft and a rotary cylinder member 25 of the motor mounted within the casing.

The main central housing block 18 has a circular recess 26 in its end wall opposite the cover 19, and in this recess is mounted a stationary pintle having a circular flange or base 27 which is securely located by bolts 28, in the recess, and a projecting stub shaft 29 which acts as a bearing support for the rotary cylinder member 25. The pintle is also formed with two internal hydraulic fluid flow passages (not shown) extending parallel to its length, each passage terminating in a port at one end to co-operate with one of two hydraulic flow passages in the end wall 16 of the housing, and a further port 30 at the other end designed to co-operate with radial ports 31 in the rotary cylinder member, to act as timing control ports as the rotor rotates. The flow ports or passages in the first mentioned end of the pintle may be provided with surrounding O-rings to form fluid seals with the co-operating face of the recess 26 but it will be appreciated that leakage would in any case be confined within the casing since the pintle is bodily contained within the housing.

The rotary cylinder member 25 in this example has five spaced projecting bosses 32 each formed with an open cylinder extending radially inwards from the outer end, and a central cylindrical bore or cavity designed to fit over the projecting end of the pintle shaft 29. The radial passages or ports 31 extend inwards from the inner ends of the cylinders into the central bore and co-operate with the timing ports 30 on the pintle to control the supply and discharge of hydraulic fluid. Each cylinder contains a piston 33 formed with a hollow skirt extending radially inwardly and a compression spring is seated within the skirt and bears against the inner end of each cylinder to urge each piston radially outwards. The outer end of each piston is formed with a part-spherical ball designed to fit in the corresponding part-spherical socket 2 in one of the bearing slippers 1 as described above, so as to act as a universal joint.

The external bearing surface of each slipper is also as described above and bears against the inner surface of the annular cam track 10 which is housed within the flange 17. Each slipper 1 is located axially on the cam track 10 between a pair of abutment shoulders 34 and 35, one shoulder 34 provided by the block 18 at the junction of flange 17 and recess 26 and the other shoulder 35 by the base of the casing member 19. The arrangement is such that the cam track can, if required, be arranged to pivot about a bottom pivot pin in order to vary its eccentricity and thus vary the capacity of the motor, as described with reference to FIG. 4. The rotary cylinder member 25 on the central pintle is free to move axially on the pintle and is connected in an axial sense with the various slippers 1 via the pistons 33 and ball joints. The fairly close clearance between the flat side or flank surfaces of each slipper and the two shoulders 34 and 35, between which the cam track 10 also moves, provides an effective seal at the opposite ends of the clearance volume at the central area of each slipper 1. Without this sealing effect the two ends of the clearance volume referred to would, of course, be vented to relief. In this particular example the total axial clearance on both sides of each slipper between the two annular guide shoulders may be of the order of 0.01 to 0.02 inches.

The central common housing member or block 18, to which the pintles of the pump and motor are attached, is formed with internal "L" shaped supply and return passages (not shown) by which fluid from the pump is delivered in to the motor and returned from the motor to the pump. The two units are of the same size and capacity and the pump 14 is also arranged to be capable of adjustment to vary its capacity and thus vary the speed ratio of the transmission. The drive to the pump is from a shaft 36 and its other parts being similar to those of the motor 15, are similarly referenced.

Opposite the motor 15, on the other side of the block 18, is a chamber formed by a casing 37, which serves as an oil reservoir. A disc 38 is attached to the block 18 and houses two non-return valves for the supply and return passages between the motor and the pump.

Preferably the arrangement is such that the casing 37 and disc 38 may be removed and a duplicate of the motor 15 be fitted to that side to give a "T" shaped transmission, the pump driving two co-axial motors at right angles to the pump axis.

I claim:

1. A radial piston hydraulic machine comprising a rotor, a plurality of pistons carried in respective cylinders therein, a slipper at the radially outer end of each piston, and a cam track with an internal cylindrical bearing surface with which bearing surfaces of the slippers co-operate, wherein the improvement comprises the provision, on each slipper, of

a. a (relieved central area of the) bearing surface with a radius of curvature greater than that of (respect to) the co-operating cylindrical surface of the cam track to provide a clearance between the two surfaces reducing from the centre outwards,

b. a central oil supply port providing unrestricted communication between the associated cylinder and said relieved central area to create a pocket of oil between said bearing surfaces.

c. from the centre of the bearing surface a radial sequence of a central land, an annular pressure groove, an annular land and an annular pressure relief groove, and

d. surface grooving through the central land from the oil supply port to the annular pressure groove.

2. A machine according to claim 1, in which the clearance at the central relieved area is approximately 0.0002 inches to 0.0004 inches.

3. A machine according to claim 1, in which the radius of curvature of each slipper bearing surface is approximately 0.005 inches greater than that of the cam track.

4. A machine according to claim 1, in which the slippers have generally flat parallel flanks, and parallel annular guide walls on opposite sides of said cam track guide said slippers and locate them axially.

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