May 21, 1968 O. H. THOMA 3,384,028

HYDRAULIC PUMPS OR MOTORS

Filed Oct. 3, 1966 3 Sheets-Sheet 1

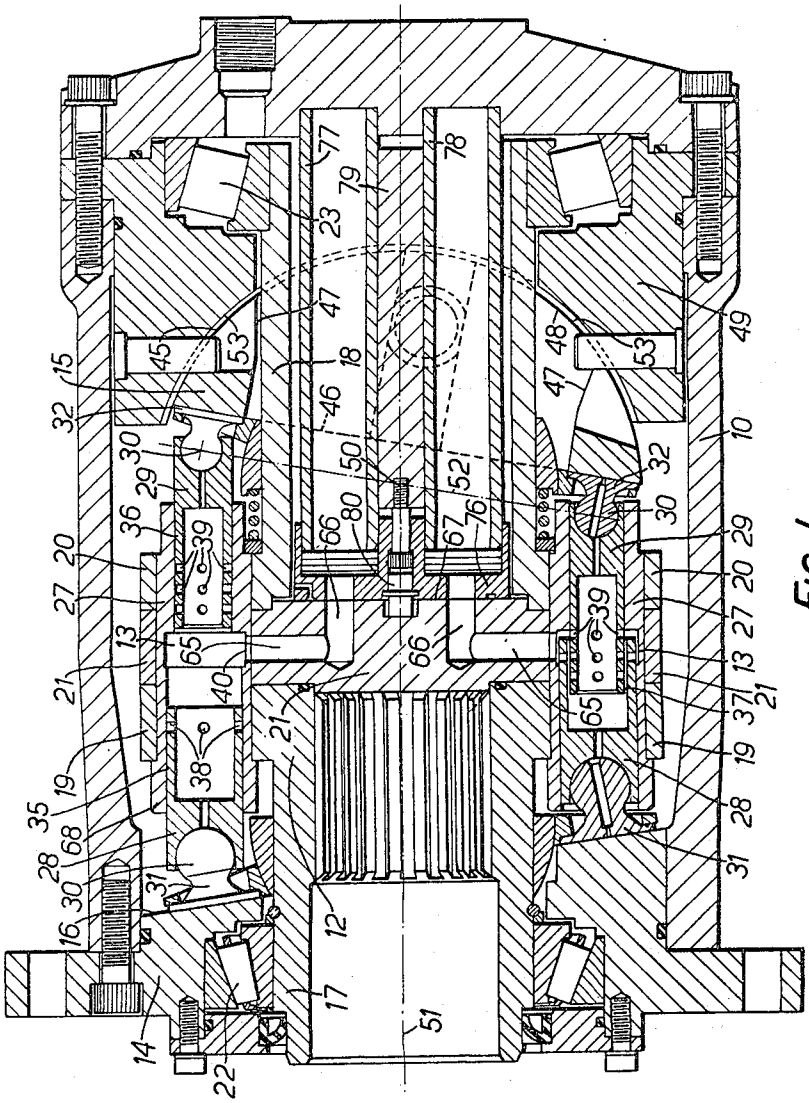


FIG. 1.

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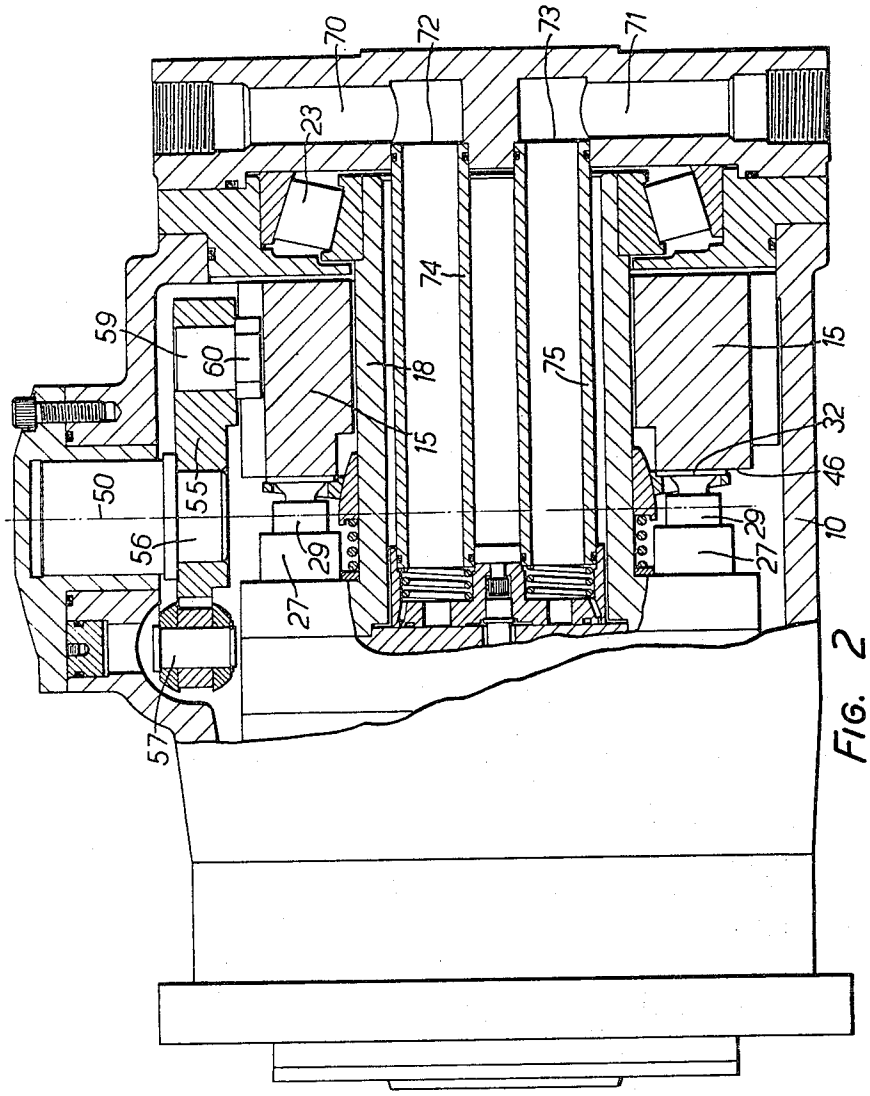


FIG. 2

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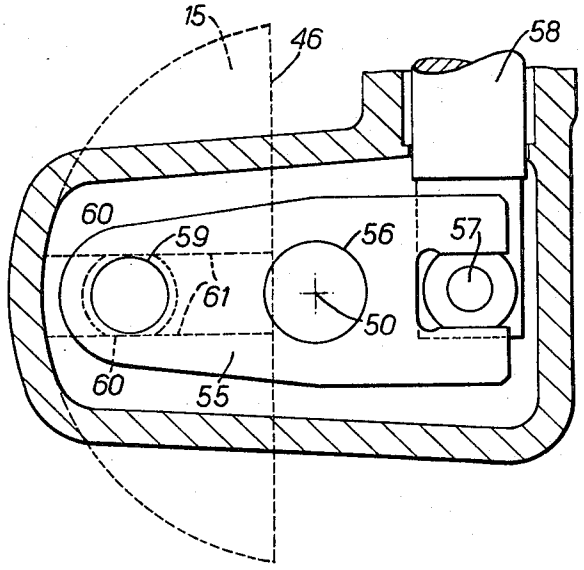


FIG. 3

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HYDRAULIC PUMPS OR MOTORS

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6 Claims. (Cl. 103—162)

ABSTRACT OF THE DISCLOSURE

A rotary hydraulic motor having a number of double-ended rotating axial cylinders each containing pairs of separate opposed pistons cooperating with inclined cam plates at opposite ends of the machine, and valve means for admitting and discharging hydraulic fluid from the cylinders, the two pistons of each pair being of different diameters and sliding in cylinder bore sections of different sizes, the inner ends of the two pistons to each opposed pair being provided with hollow skirt portions which telescope one within the other when the two pistons are in their closest relative positions.

This invention relates to axial piston hydraulic pumps or motors of the opposed piston type, which will be referred to for convenience herein as opposed piston machines. Such machines comprise in general a rotary member formed with a number of parallel cylinder bores spaced around its axis, each cylinder containing a pair of opposed pistons which react respectively against inclined cam surfaces or swash plates at opposite ends of the machine. Thus as the rotary member rotates, when the machine is used as a pump, the pistons reciprocate in opposition causing displacement of the fluid within the cylinders, or when used as a motor, supply of fluid under pressure to selected cylinders causes rotation of the machine.

In all such machines it is undesirable for the pistons to project excessively from the ends of the cylinder bores, in view of the side loads applied from the reaction against the inclined cam surfaces. In particular it is found that the ratio of the length of the projecting part of each piston, measured from the end of the cylinder bore to the centre of the usual ball joint connecting the piston to a slipper, to the length of the part of the piston contained within the cylinder bore has a maximum acceptable practical value, which may be approximately 1:1. The pistons must of course reciprocate in the cylinder bores and must project in their extended positions sufficiently for the usual slippers to make contact with the respective cam surfaces. If the inclination of the cam surfaces is appreciable quite substantial projection of the pistons is necessary. Accordingly, to maintain the desired ratio between the projecting part of the contained part of the piston, the overall length of each piston must be made substantial. Particularly in an opposed piston machine these essential increases in the length become cumulative with the result that the overall axial dimension of the machine is undesirably great. The increase in length requires extra material and involves further machining operations and added weight.

Accordingly it is an object of the invention to provide an improved opposed piston machine of the type specified, so designed that the overall axial dimensions of the machine can be maintained within reasonable limits.

The invention consists broadly in an opposed axial piston pump or motor comprising a rotary member provided with spaced cylinders substantially parallel to

the axis of rotation, each containing a pair of opposed pistons, a non-rotary cam member located at each end of the rotary member, each cam member having a surface inclined or capable of being inclined, to a plane perpendicular to the axis of rotation, and means acting between the pistons and the cam surfaces to cause rotation of the rotary member as the pistons reciprocate, or vice versa, a non-rotating distributing member having fluid admission and discharge passages, and a rotating valve member connected to rotate with the rotary member and cooperating with the non-rotating distributing member to control the admission and discharge of fluid to and from the individual cylinders, and in which the two pistons of each opposed pair are formed to telescope one within the other, or otherwise overlap, in their innermost positions.

Preferably the ports in the rotating valve member communicate respectively with intermediate points in the length of the cylinders, and the two pistons of each pair are formed to provide a port or clearance, when in their innermost positions, to allow fluid to flow between this passage and the space between the two pistons. Thus conveniently both pistons have hollow skirt portions at their inner ends and the skirt portions have apertures or port through their walls.

According to a preferred feature of the invention one piston of each pair is of larger effective diameter than the other piston.

Moreover preferably the construction of the pistons and cylinders is such that when the pistons are fully extended the ratio of the length of the projecting part of each piston relative to the part of the piston within the cylinder does not substantially exceed the desired ratio and the overall length of each cylinder bore is appreciably less than the combined lengths of the two opposed pistons when not telescoped. The cylinders in the rotary member may be formed by separate hollow cylindrical sleeves which have stepped internal bores.

In one particular construction according to the invention the effective differential pressure force substantially or fully counterbalances the opposing pressure force exerted on the ported axial end face of the rotary valve member.

When the two pistons of each pair of different effective diameters a further important advantage flows from the invention, particularly when the non-rotating distributing member abuts axially against the rotating valve member. In any such construction the hydraulic forces tend to separate the valve member from the distributing member and means must be provided for overcoming these separating forces. Now with the present invention this can be achieved simply by arranging the pistons of larger diameter at the ends of the cylinder bores remote from the non-rotating distributing member. The resultant end thrust can be so designed as to reduce or eliminate any undesirable end load on the thrust bearings supporting the rotary members.

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:

FIGURE 1 is a sectional front elevation through a hydraulic motor according to the invention,

FIGURE 2 is a plan view, partly in section, of the motor of FIGURE 1, and

FIGURE 3 is a fragmentary rear view of the motor, viewed in the direction of arrow A in FIGURE 2, with part of the casing removed to show the operating linkage for the adjustable cam member.

In this example the invention is applied to a high torque axial piston hydraulic motor of the opposed piston

type, comprising a cylindrical casing 10 closed at one end by an end plate 11, within which casing is mounted a rotary member 12 providing eleven cylinder bores 13 equally spaced about the axis. At opposite ends of the casing there are provided non-rotary annular cam members 14, 15, one of which 14 is fixed in the casing and provides an annular cam surface 16 inclined at a fixed angle of approximately 8°–15° to a plane perpendicular to the axis, while the other cam member 15 is adjustable as will be described in detail below.

The rotary member 12 is formed in three main parts, namely two hollow end parts 17, 18, each having a radial flange 19, 20 at its inner end, and an intermediate substantially flat circular plate-like part 21 sandwiched between the two flanges 19, 20. The part 21 also constitutes the rotary valve member. The two end parts 17, 18 are supported in roller thrust bearings 22, 23 from the casing of the machine and the end part 17 at the open end of the casing has internal splines 24 by which it can be connected to an output shaft of the motor (not shown).

The two flanges 19, 20 on the end parts, and the intermediate plate-like part 21, are formed with a number of aligned apertures spaced around the axis of rotation to receive a corresponding number of cylinder sleeves 27, each sleeve projecting axially a substantial distance beyond the respective flange. Each cylinder sleeve can be formed of a suitable material different from that of the rotary member, and can be suitably heat treated, machined and finished independently of the three component parts of the rotary member. Within each cylinder sleeve 27 are mounted a pair of opposed pistons 28, 29, which may be urged apart by a spring (not shown), and each piston has a spherical socket at its outer end to receive the spherical ball 30 on a sliding slipper 31 or 32 engaging the respective annular cam surface. The pistons 28, 29 of each pair are of different effective areas, the piston 28 at the open end of the machine being of larger diameter.

Each piston 28, 29 has a hollow skirt 35, 36 and the diameters of the pistons are such that the skirt 36 of the smaller piston 29 can telescope freely within the skirt 35 of the larger piston 28 when both are in their innermost positions. This provides an annular gap or clearance 37 through which fluid can enter or escape from the enclosed space within the skirts. Moreover the skirts are also formed with a number of openings 38, 39 through their walls for the same purpose, and the internal surface of each cylinder liner sleeve 27 is formed with a shallow annular groove 40 to allow fluid to pass to and from the annular clearance. The two skirt portions 35, 36 also act in the normal way as locating recesses for the ends of a spring acting between the two pistons.

In this particular example the length of each piston from its inner end to the centre of the respective ball joint is 3.1", the length of each cylinder bore is 5.4", and in their innermost positions the pistons overlap by 0.95", the distance between ball centres being 4.65". In their extended positions the distance between ball centres is 7.05" and the pistons may each project approximately 0.8" beyond the cylinder bores. It will be appreciated that this affords a substantial length of each piston within the cylinder bore, even when fully extended, and also provides a very compact unit of reduced axial dimensions.

The adjustable cam member 15 at the closed end of the machine is of generally part-cylindrical shape as viewed in front elevation, being somewhat less than a semi-cylinder, and of generally annular shape when viewed in the direction of the main axis of rotation. The part-cylindrical rear surface 45 is positioned adjacent the closed end of the machine and the cam face 46 which the slippers 32 engage, is flat. The cam member 15 is formed with a large central aperture 47 to accommodate the adjacent hollow end part 18 of the rotary member. The part-cylindrical rear surface 45 of this adjustable cam member seats against a corresponding part-cylindrical surface 48 of an annular stationary backing member 49 secured in the adjacent end of the casing, and also surrounding the hollow end part 18 of the rotary member. The part-cylindrical surface 48 on this backing member 49 is generated about a transverse axis 50 passing through the point of intersection of the rotary axis 51 of the machine and the plane 52 containing the centres of the ball joints 30 when the slippers 32 are in contact with the cam surface 46. Thus the cam member 15 can be pivotally adjusted about this transverse axis 50. The cam member has no supporting trunnions and the reaction thrust of the pistons 29 through the slippers 32 is contained purely by the backing member. It will be seen that the area of contact between the cam member and the backing member is of annular form and is in line with the path swept by the slippers, so that no bending forces are imposed on the cam member 49. To reduce friction and facilitate adjustment of the cam member 15 strips 53 of an anti-friction material, such as a bearing material impregnated with polytetrafluorethylene, are let into the part-cylindrical surface 48 of the backing member. The shape of the central aperture 47 through the cam member 15 is somewhat elongated transversely to allow the necessary pivotal movement in relation to the hollow end part 18 of the rotary member of the machine.

The operating mechanism for controlling the pivotal adjustment of the cam member 15 is shown in FIGURES 2 and 3 and includes a lever 55 pivoted on a transverse bearing pin 56 aligned with the transverse tilting axis 50 of the cam member and carried by the casing 10. One end of this lever 55 is connected by a pivotal fork point 57 to a control member 58 passing through a wall of the casing to an external control point. The other end of this lever 55 carries a pin 59 having a flattened end 60 engaging in a slot 61 formed in the flank of the cam member 15 and extending parallel to the rotary axis of the machine. Since the lever 55 is pivoted about the same transverse axis 50 as the cam member 15 pivotal movement of the lever will create the same pivotal movement of the cam member through the flatted pin 59, 60.

From the mid-point of each cylinder sleeve 27 a radial fluid passage 65 extends inwards through the sleeve and the rotary valve member, to intersect an axially extending passage 66 which opens into a port at the planar axial end face 67 of the rotary valve member 21 remote from the open end of the machine. These axial and radial passages 65, 66 are of comparatively large diameter and the substantial port areas necessarily produce a large area at the said face 67 of the valve member on which the hydraulic pressure fluid acts in a direction towards the open end of the machine, i.e. to the left in FIGURE 1.

To accommodate the pistons 28, 29 of different cross-sectional areas each cylinder sleeve 27 has an internal bore which is correspondingly of relatively large diameter at the open end of the machine and of relatively small diameter at the opposite end. The change in diameter occurs approximately at the mid-point in the length of the sleeve and this provides in effect an annular pressure surface on which the fluid pressure exerts a force acting in an axial direction away from the open end of the machine, i.e. to the right in FIGURE 1, and towards the adjustable cam member 15. The major part of the length of each cylinder sleeve is of uniform external diameter but the portion 68 of the sleeve which projects beyond the locating apertures in the components of the rotary member, at the open end of the machine, is of slightly enlarged external diameter, thus providing an annular abutment. The differential pressure forces acting on each cylinder sleeve thus tend to force the sleeve towards the closed end of the machine and so urge this abutment 68 on the sleeve against the adjacent face on the rotary member, this thrust being transmitted through the rotary member to the rotary valve member. The effective differential pressure force exerted by those cylinders which are under pressure at any instant may be designed to counter-

balance almost exactly the force in the opposite direction generated by the pressure fluid at the sealing face **67** of the rotary valve member. The differential pressure force also acts as a locating force which at all times holds the cylinder sleeves in position in the rotary member. It is therefore unnecessary to provide any locking means or detent to hold each cylinder sleeve in position, and withdrawal of the sleeves when necessary for repair or replacement is facilitated.

This construction also provides an increased thickness in the wall of the cylinder sleeve at both ends which project from the locating apertures in the rotary member. This is of considerable advantage in that these projecting parts of the sleeve are required to absorb the lateral forces generated by the sliding slippers and are therefore subject to considerable bending moments.

The end plate **11** at the closed end of the machine is formed with two diametrically opposed through-passages **70, 71**, constituting inlet and outlet passages. On the internal side of this end plate the passages issue into cylindrical sockets or recesses **72, 73** in which are mounted the ends of two elongated fluid pipes **74, 75**. These pipes extend within the respective hollow end part **18** of the rotary member, and thus also within the central aperture **47** in the adjustable cam member **15** and its backing member **49**, and at their opposite ends the pipes **74, 75** seat in two corresponding sockets in a non-rotary timing plate **76**. The two fluid pipes **74, 75** each communicate with one of two opposed kidney-shaped fluid ports **81, 82** in the timing plate. Two further diametrically opposite pipes **77, 78**, similarly connected to the end plate **11** of the casing and to the timing plate **76**, are rigidily interconnected by a welded web **79** to increase the torsional stiffness of the pipes which therefore hold the timing plate **76** against rotation. The timing plate has a substantially flat end surface arranged to bear directly on the corresponding flat surface **67** of the intermediate part **21** of the rotary member, which constitutes the rotary valve member. A stub or spigot **80** mounted centrally in the timing plate seats in a central recess in the rotary valve member to hold the two in alignment on the rotary axis.

The effective area over which the fluid pressure acts on the non-rotary distributor plate **76** in a direction away from the rotary valve member **21** is considerably greater than the opposing area on the opposite side of the distributor plate over which the fluid pressure acts, and the resultant force tends to urge the distributor plate away from the rotary valve member towards the adjacent end of the machine. This axial force is transmitted through the four elongated pipes **74, 75, 77, 78** which seat at their opposite ends in recesses in the end plate of the machine so that the end thrust is counteracted by the casing itself. Each of these fluid pipes is itself hydraulically balanced.

It will be appreciated that the invention can also be applied readily to pumps or motors having timing valves of the pintle type, with radial ports, and to constructions in which the two pistons of each opposed pair are of the same diameter.

I claim:

1. An opposed axial piston pump or motor comprising a rotary member provided with spaced cylinders substantially parallel to the axis of rotation, each containing a pair of separately movable opposed pistons, a non-rotary cam member located at each end of the rotary member, each cam member having a surface inclined, or capable of being inclined, to a plane perpendicular to the axis of rotation, and means acting between the respective pistons of each pair and the said cam surfaces to cause rotation of the rotary member as the pistons reciprocate, or vice versa, a non-rotating distributing member having fluid admission and discharge passages, and a rotating valve member connected to rotate with the rotary member and cooperating with the non-rotating distributing member to control the admission and discharge of fluid to and from the individual cylinders, and in which the two pistons of each opposed pair are of different diameters and slide within sections of the cylinders having correspondingly different internal diameters, the piston of each pair which is of larger diameter being formed with a skirt portion defining a central recess having an internal diameter somewhat greater than the external diameter of the other piston of the said pair, whereby the adjacent ends of the two pistons of each pair telescope one within the other in their innermost positions.

2. An axial piston pump or motor as claimed in claim **1**, in which the ports in the rotating valve member communicate respectively with intermediate points in the length of the cylinders, and the two pistons of each pair are formed to provide a port or clearance, when in their innermost positions, to allow fluid to flow between this passage and the space confined between the two pistons.

3. An axial piston pump or motor as claimed in claim **1**, in which both pistons have hollow skirt portions at their inner ends and the skirt portions have apertures or ports through their walls.

4. An axial piston pump or motor as claimed in claim **5**, in which the cooperating surfaces of the distributing member and the valve member are axially directed and the effective differential pressure fluid force acting on the rotary member resulting from the different diameters of the two pistons of each pair, substantially or fully counterbalances the opposing pressure force exerted on the ported axial end face of the rotary valve member.

5. An axial piston pump or motor as claimed in claim **1**, in which the overall length of each cylinder bore is appreciably less than the combined lengths of the two opposed pistons when not overlapping.

6. An axial piston pump or motor as claimed in claim **1**, in which the cylinders in the rotary member are formed by separate hollow cylindrical sleeves which have stepped internal bores, and in which the sleeves are insertable into locating apertures in the rotary member from one end, and have enlarged external flanges acting as locating abutments at the said end.

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