TELESCOPIC PISTON-CYLINDER ASSEMBLY FOR HYDRAULIC MACHINES AND MACHINE COMPONENTS

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ABSTRACT
A piston-cylinder assembly is provided, comprising two interengaging sleeve members telescopically extendable and contractable. The assembly has at each end an internal frusto-conical sealing surface for pivotally engaging a seat having a convex spherical surface, the contact line between the conical sleeve end and the spherical seat thus being a circle. The assembly is intended to be operated by a pressure fluid, primarily hydraulic and of a pressure of 5,000-6,000 psig. The tremendous operative force developed is automatically exactly centered between the two spherical seats, in any relative position thereof, without subjecting the sleeve members to tilting moments. By keeping the contact line diameter very close to the diameter of the effective pressure area of the assembly the sealing forces are low, and therefore the friction developed during the relative pivot movements between the sleeve members and the seats is low, and yet the sealing is perfect.

7 Claims, 3 Drawing Figures
TELESCOPIC PISTON-CYLINDER ASSEMBLY FOR HYDRAULIC MACHINES AND MACHINE COMPONENTS

The present invention relates generally to pressure-fluid machines or machine components, particularly hydraulic piston machines such as axial and radial piston machines, but also to machines operating with gaseous media such as pneumatic machines, and such machines components as hydraulic or pneumatic swivel couplings and the like.

With hydraulic piston machines, particularly axial piston machines, the working pressure acting on the piston is normally transmitted to the drive shaft of the machine, or a member connected with the shaft, by means of a piston rod, piston shoe or some corresponding machine component. With a conventional piston movement of the type envisaged, the force transmitting member executes a definite angular movement in relation to the piston. The oblique position of the member during the force transmitting moment brings lateral forces and tilting loads to bear on the piston, resulting in increased friction, increased wear and the risk of binding between both the piston and cylinder and between the piston and the force transmitting member at the common contacting surfaces of these members. Particularly in the case of hydraulic piston machines, these conditions create serious disadvantages, because of the high working pressures involved and because the machines must be able to deliver high forces and be capable of being driven at high loads, even during the starting-up period and when working at low revolutions, since during these stages of machine operation lubrication of the machine parts is incomplete. The lateral forces and tilting loads acting on the piston also mean that those surfaces of the piston which contact the cylinder, or the piston guide surfaces therein, must be long in relation to the diameter of the piston and cylinder and the length of piston stroke, thereby increasing the total dimensions of the machine with subsequent increase in weight and expense of the machine. Other disadvantages, well known in the art, are associated with pivoted mechanical force transmission means located between the working piston and the driven or driving shaft of the machine.

The object of the present invention is to provide for hydraulic machines and machine components an assembly, which can be designated a telescopic piston-cylinder assembly, whose main function is to permit the working fluid or pressure fluid of the machine to come into direct contact with the driven or driving shaft thereof (or a member fixedly connected with the shaft) thereby establishing a direct force transmitting connection between the fluid and the shaft without the intermediary of mechanical elements such as pivoted piston rods and the like. Thus, the force developed by the pressure fluid is transmitted practically completely to the shaft of the machine hydraulically and the piston-cylinder assembly of the invention can in actual fact, be considered to constitute a movable seal in the hydraulic system. It is another object of the invention to construct the piston-cylinder assembly in such a way that the working pressure occurring therein constantly produces a force whose resultant coincides with the symmetry axis of the assembly and is thereby incapable of subjecting the parts of the assembly to lateral forces or tilting moments of force. These objects are achieved, and a telescopic piston-cylinder assembly is provided which permits the construction of hydraulic piston machines with small external dimensions, low total weight in relation to the power delivered thereby and of high efficiency under all working conditions, by the assembly of the invention obtaining the characteristic features defined in claim 1.

The invention will now be described in more detail with reference to a number of embodiments thereof illustrated in the accompanying drawings, in which

FIG. 1 illustrates a central longitudinal section through a telescopic piston-cylinder assembly constructed in accordance with the invention and forming part of a hydraulic piston machine, and also shows associated seating surfaces,

FIG. 2 illustrates a part of the embodiment of FIG. 1 in slightly larger scale, and

FIG. 3 is a longitudinal section through a piston-cylinder assembly of the present invention in an embodiment adapted for use in a hydraulic radial piston machine, the parts of which surrounding the assembly have been shown diagrammatically in radial section.

The basic, constructive principles of the invention will be described first with reference to FIG. 1 and 2. Thus, the piston-cylinder assembly illustrated in FIG. 1 comprises an outer, sleeve-like, rotation-like, rotation-symmetrical cylinder 10 and a piston 12 slidable accommodated therein and being of the same constructional design, i.e. in the form of a rotation-symmetrical sleeve, the piston being sealed in a conventional manner in the cylinder by means of a sealing ring 13. The outer, outwardly turned end portions 14 and 16 of the cylinder and piston, respectively, are provided with seating surfaces 28 and 30 which slidably engage opposing seating surfaces 22 and 24 located on stud-like seating bodies 18 and 20 respectively. The seating surfaces 22 and 24 have a convex spherical shape. In the illustrated embodiment, there extends through one of the seating bodies, e.g. 20, a central passage 26 for supplying and removing pressure fluid to the interior of the piston-cylinder assembly to and from a commutating means or distributor valve, through which the assembly communicates with the hydraulic system.

An important feature of the invention resides in that the seating surfaces located in the ends 14, 16 of the piston-cylinder assembly are in the form of straight, truncated, internal conical surfaces or cones 28 and 30. The axes of the conical surfaces coincide with the common center axis of the cylinder 10 and piston 12, and their wider ends extend outwardly of the center of the assembly. Thus engagement between the two seating surfaces in each end of the piston-cylinder assembly takes place along circle lines 32, as can best be seen from FIG. 2, which in slightly larger scale illustrates the right half of the embodiment of FIG. 1, it being assumed that the geometrical conditions are in principle the same at both ends of the piston-cylinder assembly. Half of the cone angle of the conical seating surface 28 of the cylinder 10 is equal to y i.e. the cone angle is 2y. By fulfilling certain conditions, to be hereinafter discussed, the seating surfaces 30 and 24 of the opposing piston end (see FIG. 1) can deviate from corresponding values at the end of the cylinder with respect to the cone angle and the radius of the arcuate seating body.

By constructing the telescopic piston-cylinder assembly of the invention in the aforesaid manner, a series of problems which have previously caused difficul-
ties in hydraulic machines and machine components of the type envisaged are solved. In summary, it can be said that three decisive advantages are gained by means of the invention, these advantages being important both in principle and in practice. Since the aforementioned advantages are extremely important to a better understanding of the invention, they will now be described in detail.

When considering first the cylinder 10, it will be evident that the total fluid pressure acting on the inside thereof will propagate over the seating surface 22, i.e. over the end of the seating body 18, out to the contact circle 32, where it will drop almost immediately to zero on the outside of the circle, i.e. the circle demarcates a sharp limit between the pressure prevailing within and externally of the assembly. In this connection, it is assumed that the cone angle 2\(\gamma\) is so selected in relation to the radius of the seating surface 22 that the radius of the contact circle 32 is less than the internal radius R of the cylinder by the magnitude \(\varepsilon\), FIG. 2. If a pressure diagram is written on an inwardly projecting abutment 34 located at the end of the cylinder and lying axially in front of the conical seating surface 28, it will be seen that the cylinder will engage the seating 22 with a force which corresponds to the prevailing fluid pressure \(p\) multiplied by the area of the circular ring with the width \(\varepsilon\). This latter measurement can thus be selected so that said circular area comprise a very small portion of the inner cross-sectional area of the cylinder, e.g. between 0.5 and 2 percent, say 1 percent thereof. Thus, this means that the mechanical abutment forces between the cylinder 10 and the seating body 18 corresponds to only approximately 1 percent of the total force axially exerted by the fluid pressure, while the remainder of this force, approximately 99 percent, is transmitted directly to the seating body 38 in the form of hydrostatic pressure exerted by the pressure fluid.

The conditions are identical at the opposite end of the piston cylinder assembly, i.e. at piston end 16, and the piston 12 will be balanced hydraulically so that its mechanical abutting force against the seating surface 24 of the seating body 20 merely constitutes approximately 1 percent of the total force developed axially by the pressure fluid. If considered desirable, for reasons of construction or otherwise, the cone angle \(\gamma\) and the radius of the part-spherical seating body can, as aforementioned, have other values than those at the cylinder end 14 of the assembly, since obviously the condition for the hydraulic balancing of the assembly as a whole are merely that the contact circles 32 are equal or practically equal at both ends.

The second of the three decisive advantages afforded to the inventor is that because of the symmetrical design the described pressure and force conditions will prevail independently of the position of the piston-cylinder assembly 10, 12 in relation to the outer spherical seating bodies 18, 20, i.e. independently of the angles \(\alpha\) and \(\beta\), between the center lines 36 of the assembly and the center lines 36 and 40 of the seating bodies 18 and 20 resp., FIG. 1. Irrespective of the angular position of the assembly in relation to the spherical seating bodies at the ends thereof, the resultant force of the forces created by the fluid pressure will always act along the center line of the unit. In conjunction with the low mechanical abutting force created by the fluid pressure between the ends of the assembly and the outer seating surface, with subsequent small frictional forces, this centering of the forces prevents the fluid pressure from either directly or indirectly giving rise to appreciable lateral forces or tilting moments between the cylinder and the piston. In turn, this implies that the internal friction in the assembly, apart from that caused by the sealing ring 42, is low and thereby the risks of abnormal wear or binding or seizing in the assembly are small. Finally, the common guide surface between the piston and the cylinder, and thus also the assembly, can be made short.

The third advantage obtained with the piston-cylinder assembly of the present invention is connected with the fact that the seating surface 28 and 30 at the ends of the piston-cylinder assembly have an internally conical configuration. As a matter of course, these surfaces can in principle be of another geometrical shape, for example concave spherical surfaces having the same radius as respective outer seating surfaces 22 and 24, this latter shape being perhaps the more conventionally obvious one, wherewith the assembly would still obtain substantially the same functional properties as those aforedescribed. However, when studying more closely the consequences of a construction in which the seating surfaces are concave and arcuate in shape, it will be seen that such a design is quite detrimental from a practical and economic point of view. First and foremost, seating surfaces 28 and 30 having a concave spherical configuration must conform extremely accurately with the corresponding convex spherical seating surfaces, in order for the sealing clearances between the surfaces in question to obtain an exact and reproducible geometrical form (parallelity) and size (clearance gap), a condition which requires expensive and time consuming production methods, finishing, for example, with a lapping operation. In contradistinction hereto, an internal conical surface is relatively simple to produce with sufficient accuracy, e.g. by turning and grinding, at same time as relatively moderate tolerances on the diametrical measurements of the cooperating convex, spherical seating surface can be permitted. Thus tolerance can be permitted since deviation in the diameter of the sphere from a nominal measurement only affect the contact circle 32, i.e. the degree of hydraulic balancing of the cylinder and piston, but not the geometrical shape and size of the sealing gap, since the tapering surfaces can adjust themselves to the spherical surfaces by mutual axial movements.

However, spherical seating surfaces introduced throughout have still more serious consequences in connection with the unavoidable elastic deformations which occur in apparatus of the type in question. In order, among other things, to keep dynamic forces, such as centrifugal forces, as low as possible, it is desirable that the wall thickness of the cylinder as well as of the piston of the assembly is as thin as possible. On the other hand, very high fluid pressures occur and these two conditions cause the diametrical dimensions of the piston-cylinder assembly to vary remarkably under load, but this fact, in analogy with the foregoing does not affect the sealing conditions at the ends of the suggested assembly. Instead the variations are compensated by axial movements of the cylinder and piston in relation to their respective seating surface 22 and 24 and only affect the axial balancing conditions to a slight extent. If, again, the seating surfaces 28 and 30 in the ends of the assembly were spherical in shape, these
5 variations in diameter would strongly affect both the sealing conditions and the axial hydraulic balancing conditions, and the result would be the occurrence of high frictional forces, increased leakage and other difficultly resolved problems; furthermore, the previously described functional advantages of the assembly would be completely or partially lost.

The aforesaid functional and manufacturing advantages afforded with conical seating surfaces 28 and 30 in relation to seating surfaces of spherical or other geometrical shape are accentuated with reducing value of the angle \( \gamma \), i.e. the nearer the diameter of the sealing surface 32 approaches the diameter of the spherical seating surface 22. This means that the assembly of the invention is particularly suited for use in hydraulic machines having axially arranged piston systems which coincides with the conditions hereinafter described in connection with a description of such a machine.

When the assembly occupies its rest position or when the pressure of the working fluid is low, the ends of the piston-cylinder assembly can be held in engagement with respective outer seating surfaces by means of a coil spring 44 mounted externally of the assembly, the spring being arranged to engage external abutments 46 and 48 on the outer ends of the cylinder 10 and the piston 12 respectively. The ends of the assembly can however, be mechanically held against cooperating outer seatings in other, known ways.

The principle construction of the piston-cylinder assembly of the invention illustrated in FIG. 1 also teaches a practical non-motorial application of the assembly. If the outer seating body 18 cooperating with the cylinder 10 is, similarly to the opposing seating body 20, also provided with a through passing passage 27, as indicated with dotted lines, the assembly of the present invention can be used as a hydraulic machine element, e.g. in the form of a swivel or pivotal joint in a high pressure line for hydraulic or pneumatic fluids.

In the aforesaid, the piston-cylinder assembly has been described as cooperating with outer seatings which are both spherically shaped. It should be understood, however, that the invention is not restricted to the described embodiments having the more "universal" movement ability. In many instances it is fully sufficient or desirable that the assembly executes a pendulum movement in one plane, in which case it is sufficient to arrange the described seating engagement between an internally conical and a convex curved surface at one end of the piston-cylinder assembly, while engagement at the opposite end of the assembly can be of a more conventional character. A typical example of this application of the invention is shown diagrammatically in FIG. 3, which illustrates how a piston-cylinder assembly according to the invention can be applied in a radial piston machine. This machine includes a housing 72, in which a shaft 74 is rotatably mounted. The shaft is provided with an eccentric 76 having a cylindrical surface and the center points of the eccentric are indicated by the references 0 and 0'. The outer cylindrical surface 78 of the machine is provided on the inside thereof with a cylindrical surface 80, against which a seating body 82 bears, the supporting surface of the seating body conforming to the cylindrical surface of the housing, as shown in FIG. 3. The radially inwardly turned surface 84 of the seating body 82 is convex and spherical in shape, and engaging said surface is the internally conical end surface of a piston 90 forming part of a piston-cylinder assembly according to the invention, the cone angle of said conical end surface being 2\( \gamma \). The interior of the piston-cylinder assembly communicates with the surroundings through a passage 86 arranged centrally in the seating body 82.

The opposite, radially inwardly directed end surface of the piston-cylinder assembly, i.e., the end surface of the cylinder 92 forming part of the assembly, is shaped to conform with and engage the cylindrical surface of the eccentric 76. The hydraulic conditions are essentially the same with this embodiment as those described in the aforesaid, the fluid pressure prevailing in the assembly acting directly on the surface of the eccentric, and by suitable design and dimensioning of the edge surface of the cylinder abutting the eccentric a high degree of hydraulic balance between the surfaces can be obtained. The hydraulic balance is obtained at the opposing piston end in the manner aforesaid, the cone angle 2\( \gamma \), being adapted according to the radius of the seating surface 84.

Upon rotation of the shaft 74, the piston-cylinder assembly will execute a pendulum movement around the center of the seating surface 84. With respect to known piston arrangements in radial piston machine, the unit 90, 92 presents the same principal and manufacturing advantages as the previously described embodiment of the assembly.

Owing to the extensive balancing of the forces created by the fluid pressure of the movable parts of the mechanism, characteristic of the piston-cylinder assembly according to the invention, with subsequent slow friction losses, the assembly can be used to advantage for machines intended for gaseous working medium, e.g. compressed air motors or compressors.

The invention is not restricted to the described and illustrated embodiments thereof but can be modified within the scope of the following claims.

What is claimed is:

1. A telescopic piston-cylinder assembly for use with a pressure fluid, comprising:
   a telescopic unit having first and second bearing surfaces formed on the opposite ends thereof;
   first and second reaction-force absorbing seat means respectively engaging said first and second bearing surfaces for supporting said unit;
   said first and second seat means having first and second seating surfaces, respectively, disposed in slidable bearing engagement with said first and second bearing surfaces, respectively, for permitting angular movement of said unit relative to at least one of said seating surfaces;
   said telescopic unit having a maximum cross-sectional pressure area defined by a predetermined diameter, and including a sleeve-like cylinder and a sleeve-like piston slidably received in said cylinder, the opposed ends of said cylinder and sleeve being axially overlapped, the other end of said cylinder and sleeve respectively having said first and second bearing surfaces thereon;
   passage means communicating with the interior of said unit for permitting pressure fluid to be supplied to the interior of said cylinder; and
   said one seating surface being provided with a convex spherical slide surface, and the corresponding one of said first and second bearing surfaces as dis-
posed in engagement with said one seating surface having an internal conical configuration whereby it engages said slide surface substantially along a circular line of contact, said circular line of contact between said slide surface and the corresponding bearing surface having a diameter substantially equal to or only slightly smaller than said predetermined diameter.

2. An assembly according to claim 1, wherein at least one of said seat means is substantially solid and forms a reaction surface for the pressure fluid contained within said cylinder.

3. An assembly according to claim 2, wherein said passage means extends through said other seat means and communicates with the interior of said cylinder.

4. An assembly according to claim 1, wherein the annular surface between the periphery of the maximum cross-sectional pressure area of the unit and the said circular line of contact is approximately between 0.5 and 2 percent of the total maximum cross-sectional pressure area.

5. An assembly according to claim 1, further including spring means positioned externally of said unit in surrounding relationship to said piston and cylinder for resiliently urging said piston and cylinder in opposite axial direction for maintaining the opposite ends of said unit in secure engagement with said first and second seat means.

6. In a telescopic piston-cylinder assembly for a hydraulic machine including a telescopic unit having a sleeve-shaped piston slidably received in a sleeve-shaped cylinder to which a pressure medium is introduced, the unit being movably disposed between and supported on a pair of spaced reaction-force absorbing seating surfaces each engaging a respective end of the unit, comprising the improvement wherein at least one of the seating surfaces is provided with a convex spherical slide surface disposed in engagement with the adjacent end of the piston-cylinder unit, said adjacent end of said unit having an internal bearing surface of conical configuration whereby said bearing surface engages said slide surface substantially along a circular line of contact whose diameter is only slightly smaller than the maximum diameter of the active pressure area of the piston-cylinder unit, said pressure area being the interior cross-sectional area of said unit perpendicular to the longitudinal axis of said unit.

7. In an assembly according to claim 6, wherein the circle of contact has a diameter smaller than the diameter of that active pressure area so as to define a narrow annular region therebetween, said narrow angular region having an area perpendicular to the longitudinal axis of said unit of approximately between 0.5 and 2 percent of the maximum active pressure area of the piston-cylinder unit.
UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3 742 819 Dated July 3, 1973

Inventor(s) Lennart Werner Preese

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

On the Title page, after "Assignee:", the spelling of the Assignee should be corrected to read ---New-Invent S.A., Geneva, Switzerland---.

Column 6, line 60; change "end" to ---ends---.

Signed and sealed this 19th day of February 1974.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR. C. MARSHALL DANN
Attesting Officer Commissioner of Patents