United States Patent [19]

Freese

[11] **3,885,459**

[45] **May 27, 1975**

[54] **TELESCOPIC PISTON-CYLINDER ASSEMBLY FOR HYDRAULIC MACHINES AND MACHINERY COMPONENTS**

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[73] Assignee: **New-Invent S.A.,** Geneva, Switzerland

[22] Filed: **Mar. 26, 1973**

[21] Appl. No.: **345,073**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 114,935, Feb. 12, 1971, Pat. No. 3,742,819.

[30] **Foreign Application Priority Data**

Feb. 20, 1970 Sweden 2183/70

[52] **U.S. Cl.** **92/119;** 92/130

[51] **Int. Cl.** **F01b 15/04;** F01b 31/00

[58] **Field of Search** 92/119, 130, 72, 66; 91/490, 499

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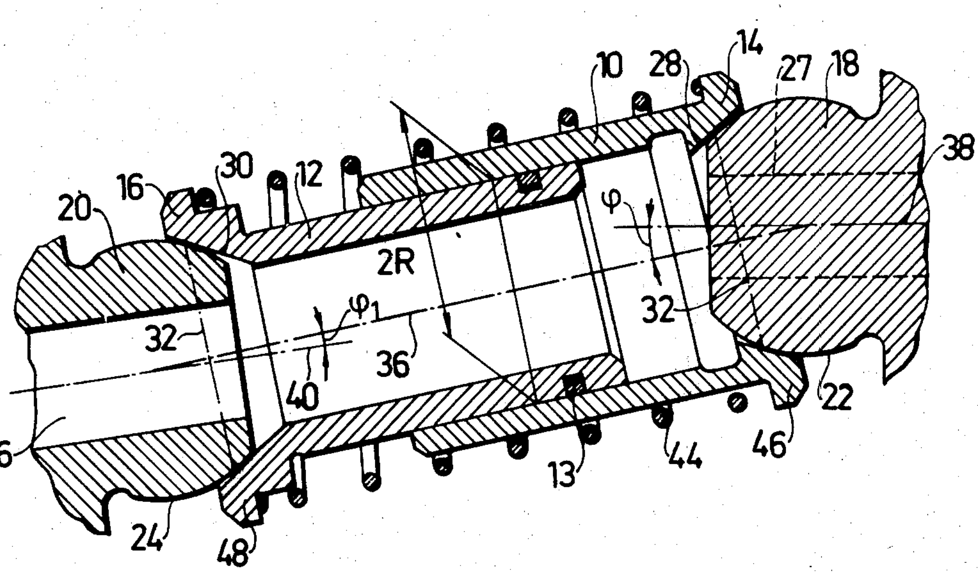
Primary Examiner—Paul E. Maslousky

Attorney, Agent, or Firm—Woodhams, Blanchard and Flynn

[57] **ABSTRACT**

A piston-cylinder assembly is provided, comprising two interengaging sleeve members and two seats. The sleeve members are telescopically extendable and contractable and form a piston-cylinder unit having at either end a supporting surface for pivotally engaging the seating surface of a respective seat. The supporting and seating surfaces form together two pairs of mating bearing surfaces. In at least one of said pairs the one bearing surface is convexedly spherical whereas the other has an internal conical configuration, the contact line between the two bearing surfaces thus being a circle. By keeping the diameter of said circle very close to the diameter of the effective pressure area of the assembly the seating forces are low and therefore the friction developed during the relative pivoting movements between the piston-cylinder unit and the seats is low, even at operating pressures of 5000-6000 psig.

5 Claims, 5 Drawing Figures



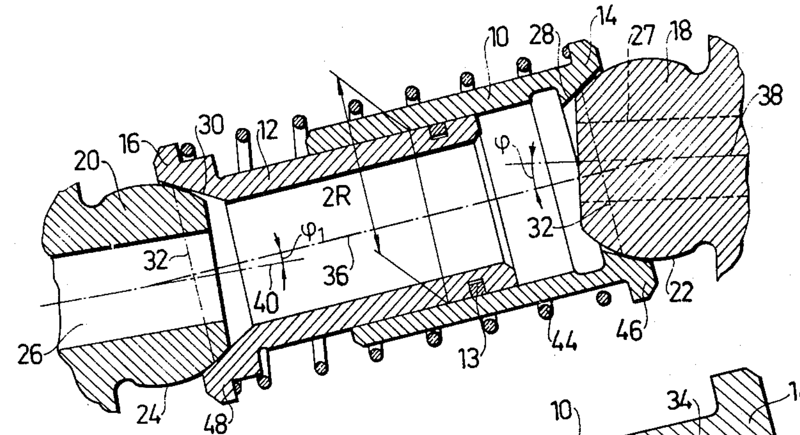


Fig.1

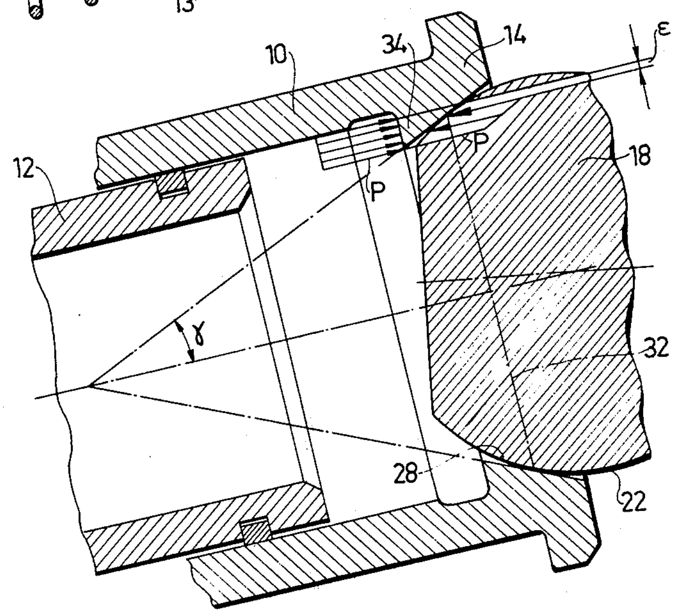


Fig. 2

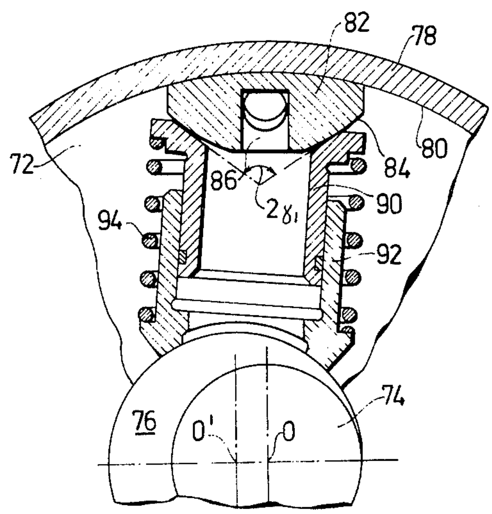


Fig.3

Fig. 4

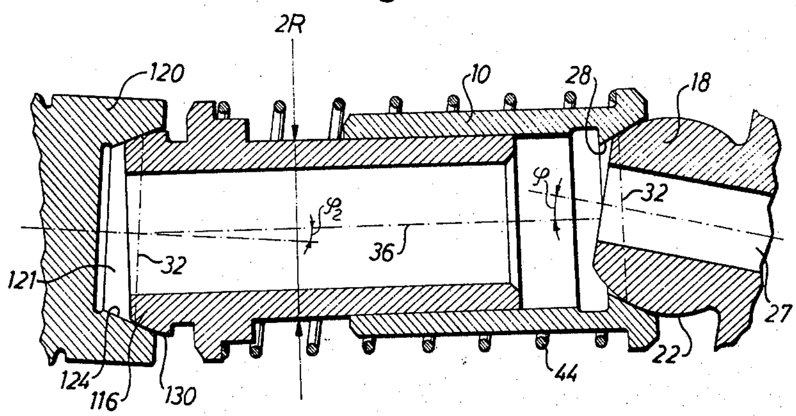

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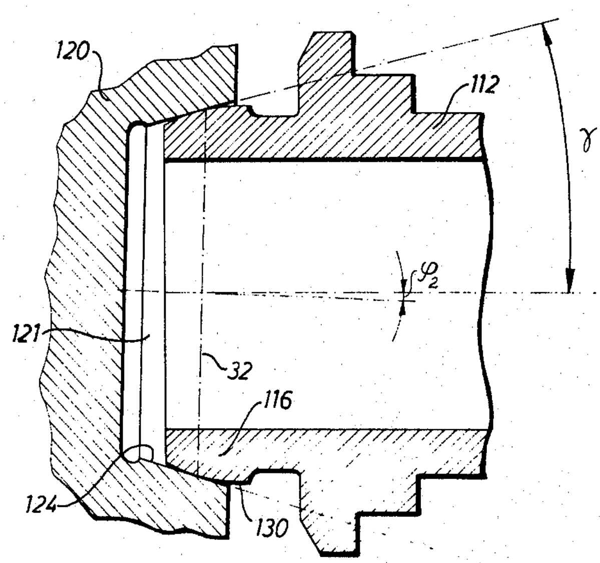
Fig. 5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TELESCOPIC PISTON-CYLINDER ASSEMBLY FOR HYDRAULIC MACHINES AND MACHINERY COMPONENTS

CROSS-REFERENCE TO RELATED APPLICATION

This application is a continuation-in-part of my copending application Ser. No. 114,935, filed Feb. 12, 1971, now U.S. Pat. No. 3,742,819.

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 machine components as hydraulic or pneumatic swivel couplings and the like. The invention includes a further improvement of the arrangement described in my earlier U.S. Pat. application Ser. No. 114,935, now U.S. Pat. No. 3,742,819.

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 imcomplete. The lateral forces and tilting loads acting on the piston also means 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 occuring 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. Still another object is to provide a telescopic piston-cylinder assembly 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.

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

FIG. 1 illustrates a central longitudinal section through a telescopic piston-cylinder assembly constructed in accordance with the invention and comprising a piston-cylinder unit inserted between associated end seats;

FIG. 2 illustrates a part of the embodiment of FIG. 1 on a slightly larger scale;

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;

FIG. 4 is a central longitudinal section corresponding to FIG. 1 but taken through a modified embodiment of the piston-cylinder assembly of the invention; and

FIG. 5 illustrates a part of the embodiment of FIG. 4 on a slightly larger scale.

The basic, constructive principles of the invention will be described first with reference to FIGS. **1** and **2.** Thus the piston-cylinder assembly illustrated in FIG. **1** comprises an outer, sleeve-like, rotation-symmetrical cylinder **10** and a piston **12** slidably accomodated 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 cylinder and piston form together a piston-cylinder unit, whose outer, outwardly facing end portions **14** and **16** are provided with seating surfaces **28** and **30,** respectively, which slidably engage opposing seating surfaces **22** and **24** located on stud-like seating bodies **18** and **20** respectively, component of the assembly; thus the piston-cylinder assembly as a whole comprises the piston-cylinder unit **10, 12** plus the outer seats or seating bodies **18, 20.** The seating surfaces **22** and **24** of said bodies 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 in general.

An important feature of the invention resides in that the seating surfaces located in the ends **14, 16** of the piston-cylinder unit 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 centre axis of the cylinder **10** and the piston **12,** and of course the surfaces diverge outwardly of the centre of the unit. 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 γ i.e. the cone angle is 2γ. 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 aforedescribed manner, a series of problems which have previously caused difficulties 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 2γ 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 ϵ, 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 ϵ. 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 0.2 and 2 percent, say 1 percent thereof. Thus, this means that the mechanical abutment force 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 γ 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 by the invention is that because of the symmetrical design the described pressure and force conditions will prevail independently of the position of the piston-cylinder unit 10, 12 in relation to the outer spherical seating bodies 18, 20, i.e. independently of the angles ϕ and ϕ_1, between the centre lines 36 of the unit and the centre lines 36 and 40 of the seating bodies 18 and 20 resp., FIG. 1. Irrespective of the angular position of the unit 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 centre line of the piston-cylinder unit. In conjunction with the low mechanical abutting force created by the fluid pressure between the ends of the unit and the outer seating surfaces, 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 13, 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 as a whole, 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 unit 10, 12 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 acurately with the corresponding convex shperical seating surfaces, in order for the sealing clearances between the surfaces in question to obtain an exact and reproducable geometrical form (parallelitity) 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 the same time as relatively moderate tolerances on the diametrical measurements of the cooperating convex, spherical seating surface can be permitted. This tolerance can be permitted since deviation in the diameter of the sphere from a nominal measurement only affect the diameter of 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 defomations 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 wal 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 aforegoing 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 variations in diameter would strongly affect both the sealing conditions and the axial hydraulic balancing conditions, and the result would be the occurance of high frictional forces, increased leakage and other difficulty resolved problems; furthermore, the previously described functional advantages of the assembly would be completely or partially lost.

When the assembly occupies its rest position or when the pressure of the working fluid is low, the ends of the piston-cylinder unit 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 other ends of the cylinder **10** and the piston **12** respectively. The ends of the unit can however, be machanically 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 pivotable joint in a high pressure line for hydraulic or pneumatic fluids.

In the aforegoing, the piston-cylinder assembly has been described as including 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 and 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 centre 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** component of a piston-cylinder unit forming part of a piston-cylinder assembly according to the invention, the cone angle of said conical end surface being $2\gamma_1$. 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 aforegoing, 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 opposite piston end in the manner aforedescribed, the cone angle $2\gamma_1$, 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 centre 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.

The structure as described above relative to FIGS. **1–3** corresponds to the structure disclosed in my U.S. Pat. No. 3,742,819.

A further case of great interest, also of the kind when the piston-cylinder assembly according to the invention has an external convex shperical seat at one end only, is illustrated in FIGS. **4** and **5.** This embodiment is particularly useful in such applications where the relative angular movement of the assembly is moderate at one end, that is, when the piston-cylinder unit need not make too great angular movements in relationship to the external seat at this end.

A comparison between FIGS. **1** and **4** shows clearly that the mating between the piston-cylinder unit and the seat at the left end, as viewed in FIG. **4,** of the assembly is completely analogous to that occurring at the right end, but mating is converse, so to speak, i.e. there is still mating between an internal conical surface and a convex spherical surface with the accompanying line contact, but in this case the conical sealing surface is provided in the seat as a depression therein, whereas the convex spherical surface is provided on the end of the piston-cylinder unit in the form of a comparatively narrow spherically segmental band.

Thus the embodiment according to FIGS. **4** and **5** has certain components in common with the embodiment of FIGS. **1** and **2,** and these components have retained their reference characters. Accordingly, the piston-cylinder assembly comprises a sleeve-like cylinder **10** telescopically coacting with a piston **112,** likewise sleeve-like, said parts forming a piston-cylinder unit which is brought to expand resiliently, for example by means of a coil spring **44,** into engagement with respective external seats **18** and **120.** As in the first embodiment the seat **18** has the configuration of a part of a sphere with a convex spherical seating surface **22,** while the opposite seat **120** is provided in the form of a recess or depression **121** having a straight conical wall **124** constituting the seating surface. As before, the cylinder **10** engages the seat **18** by an internal conical surface **28** provided in the cylinder end, whereas the piston **112** is provided, at its end engaging seat **120,** with a convex spherical surface **130** which extends as a comparatively narrow band about the piston end and engages the internal conical seating surface **124,** as may be readily seen in FIGS. **4** and **5.** A passage **27** may be provided in the seat **18** for supplying and removing pressure fluid to and from the interior of the assembly.

Disregarding the converse situation brought about at the left end of the assembly the geometrical conditions will be the same as in the embodiment of FIGS. **1** and **2.** Thus the engagement between the conical and spherical sealing surfaces takes place along a circular line of diameter **32** which is smaller than by an insignificant amount the diameter 2R of the operative pressure area of the assembly, that is, the cross sectional area of the common sliding surface of the piston and the cylinder. The spherical sealing surfaces may exhibit the same radius of curvature which, however, is in no way necessary, and the conical sealing surfaces may thus have the same cone angle γ (see FIG. 5). The relative pivoting or tilting angle between the piston-cylinder unit **10, 112** and the seat **18,** i.e. the angle between the centre line of the seat and the axis **36** of the unit, is designated also here by ϕ, whilst the corresponding angle at the opposite end is designated ϕ_2. If the geometrical conditions are considered it will be seen that the possible angular movement at the left end of the assembly, i.e. the possible range of ϕ_2, is rather limited and is determined by the width of the spherical sealing band **130** at the end of piston **112.** However, in many cases it is possible to adapt the assembly in such a way that in operation the angular movement will occur substantially at that end of the assembly where the seat has a spherical configuration, whereas the movement at the opposite end is limited to slight tilting, and in such cases the embodiment according to FIGS. **4** and **5** offers great advantages in the form of a simplified structure of minimum axial length. Also, and as a matter of course, there may be cases when it is to advantage that both ends of the assembly is constructed with internal conical seats as disclosed above and specifically illustrated in FIG. **5.**

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 low 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.

The embodiments of the invention in which I claim an exlcusive property or privilege are defined as follows:

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 supporting 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 piston being axially overlapped, the other ends of said cylinder and piston 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 thereto and removed therefrom;

one of said seating surfaces being provided with an internal conical configuration, and the corresponding one of said bearing surfaces as disposed in engagement with said one seating surface having an outer convex spherical configuration whereby it engages said one seating surface substantially along a circular line of contact, said line of contact having a diameter of magnitude similar to said predetermined diameter; and

the other of said seating surfaces being provided with an outer convex spherical configuration, and the other of said bearing surfaces as disposed in engagement with said other seating surface having an internal conical configuration whereby it engages said other seating surface substantially along a circular line of contact having a diameter of magnitude similar to 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 the assembly.

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

4. An assembly according to claim **1,** wherein the diameter defined by said circular line of contact is only slightly smaller than said predetermined diameter, and wherin the annular surface between the periphery of the maximum cross-sectional pressure area of the unit and said circular liner of contact is approximately between 0.2 and 2 percent of said maximum cross-sectional pressure area.

5. An assembly according to claim **1,** further including spring means positioned externally of said unit 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.

* * * * *

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,885,459 Dated May 27, 1975

Inventor(s) Lennart Werner Freese

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

The term of this patent subsequent to July 3, 1990,

has been disclaimed.

Signed and Sealed this

Twentieth **Day of** July 1976

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks