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(54) Piston and connecting rod assembly

(57) To combine the advantages of the ball joint connection between a connecting rod 8 and a piston 3 with the advantages of a cylindrical joint and to facilitate the replacement of all sliding parts, a pivot member 5 is releasably secured to the connecting rod 8. Member 5 is in sliding contact by way of the circular generated surface 14 on its periphery with a matching concave surface 15 in a pressure member 4. By way of a convex circular surface 18 the member 4 bears on an inside surface 16 of the piston crown 17.

Projections 30 of pivot member 5 may engage with clearance in recesses in the piston skirt 20. The contacting surfaces of the members 5 and 4 can be barrel-shaped. In adaptation to the circular surface 18 of the pressure member 4 the piston crown inside surface 16 can be in the form of a circular generatrix.

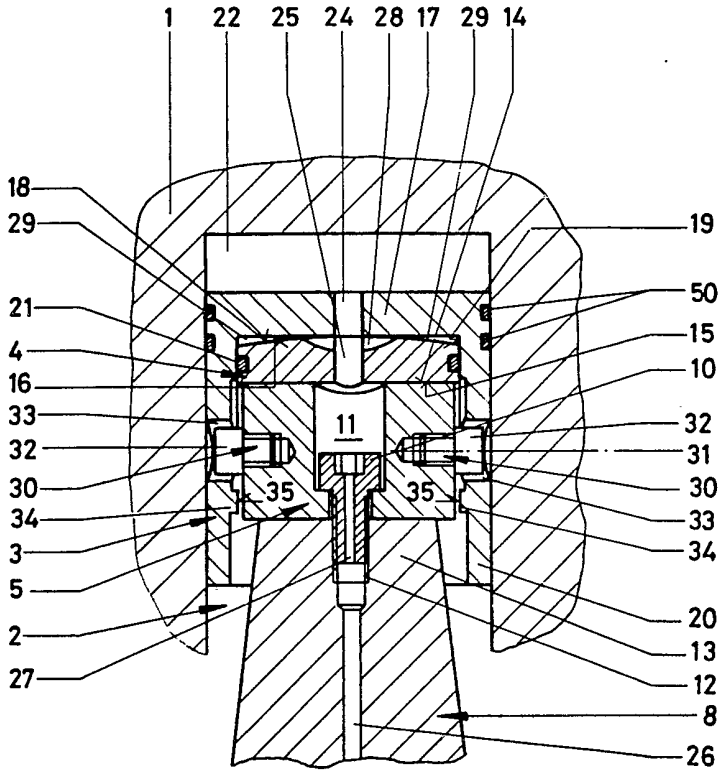

Fig. 2

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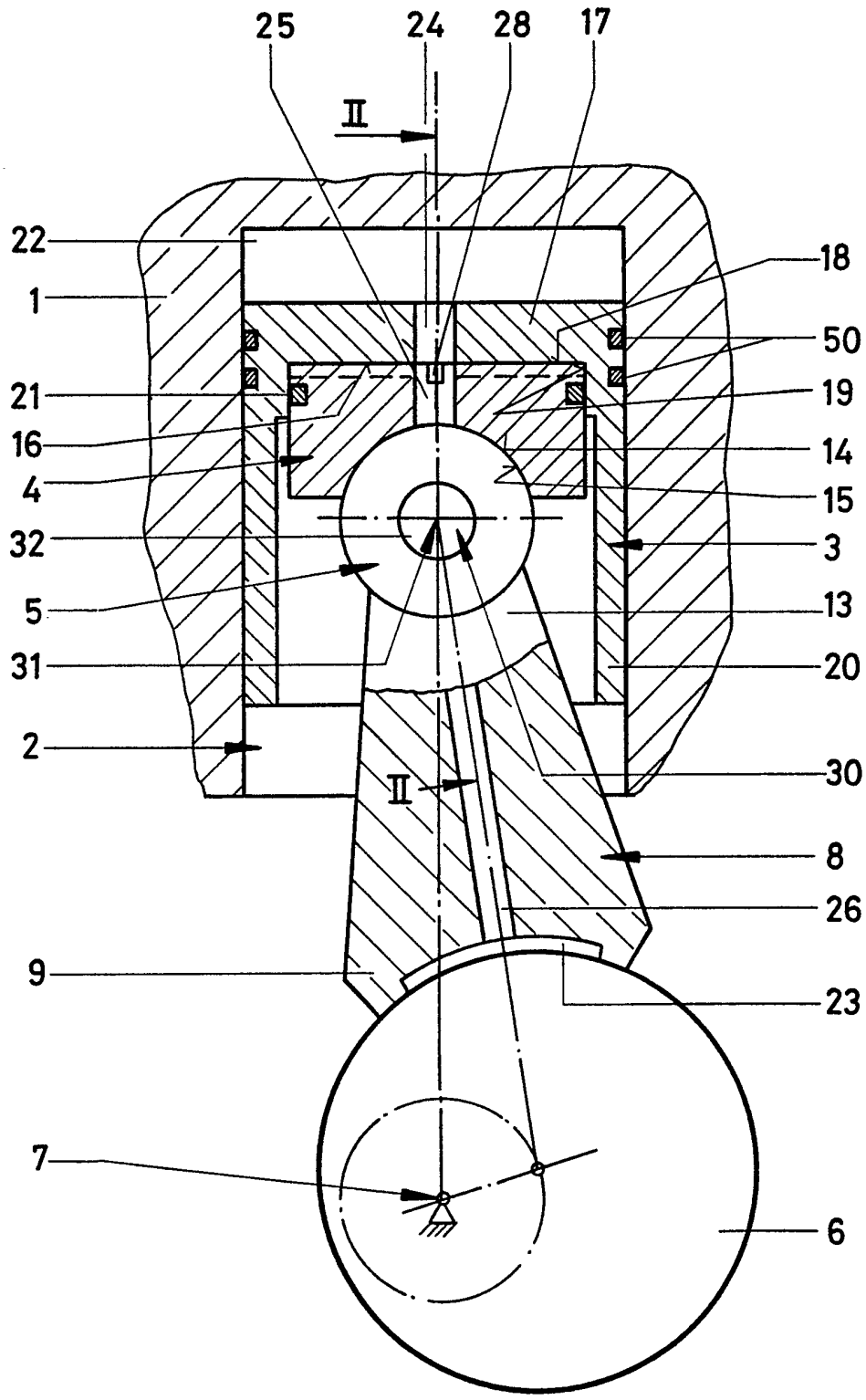
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Fig. 1

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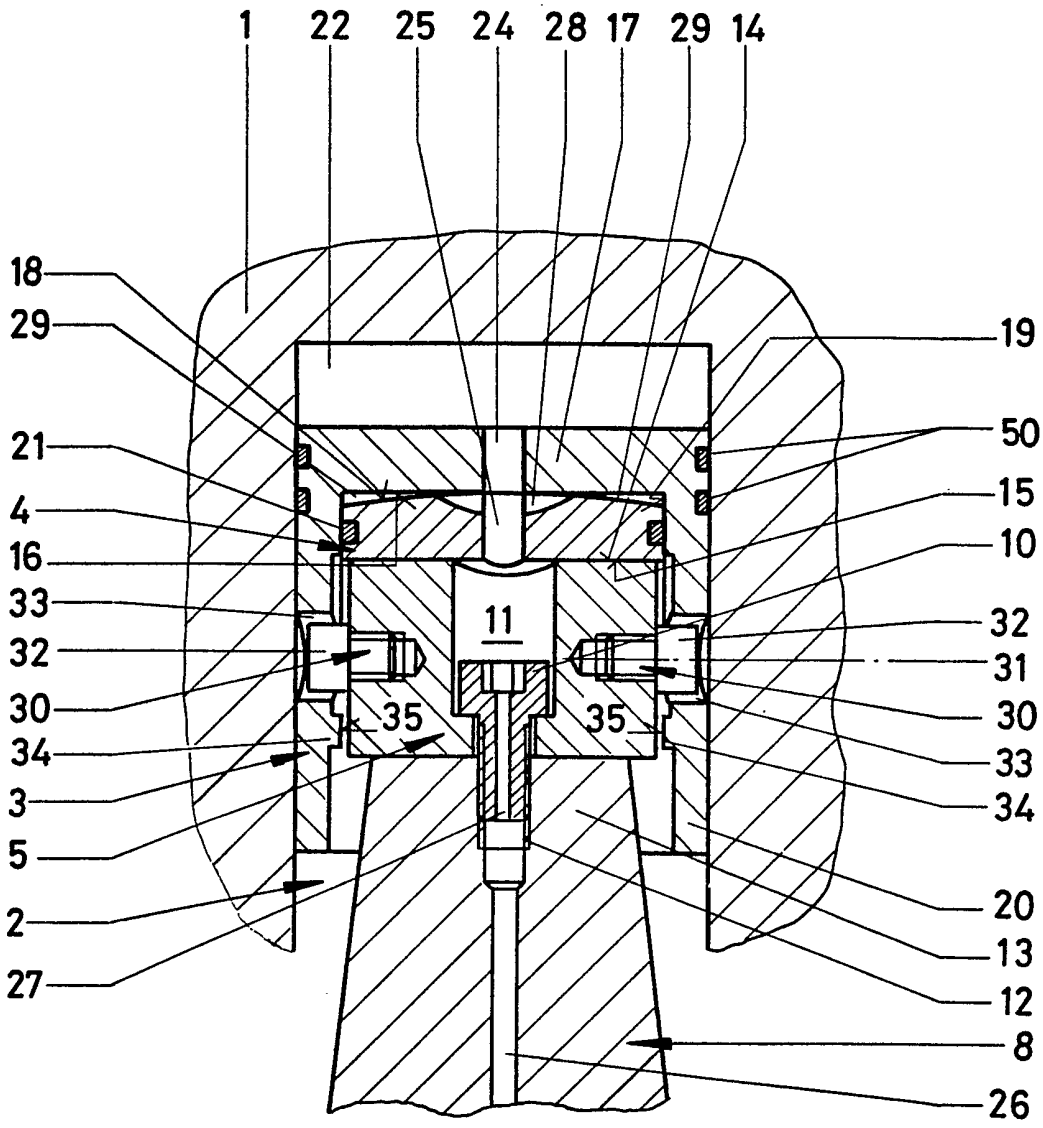

Fig. 2

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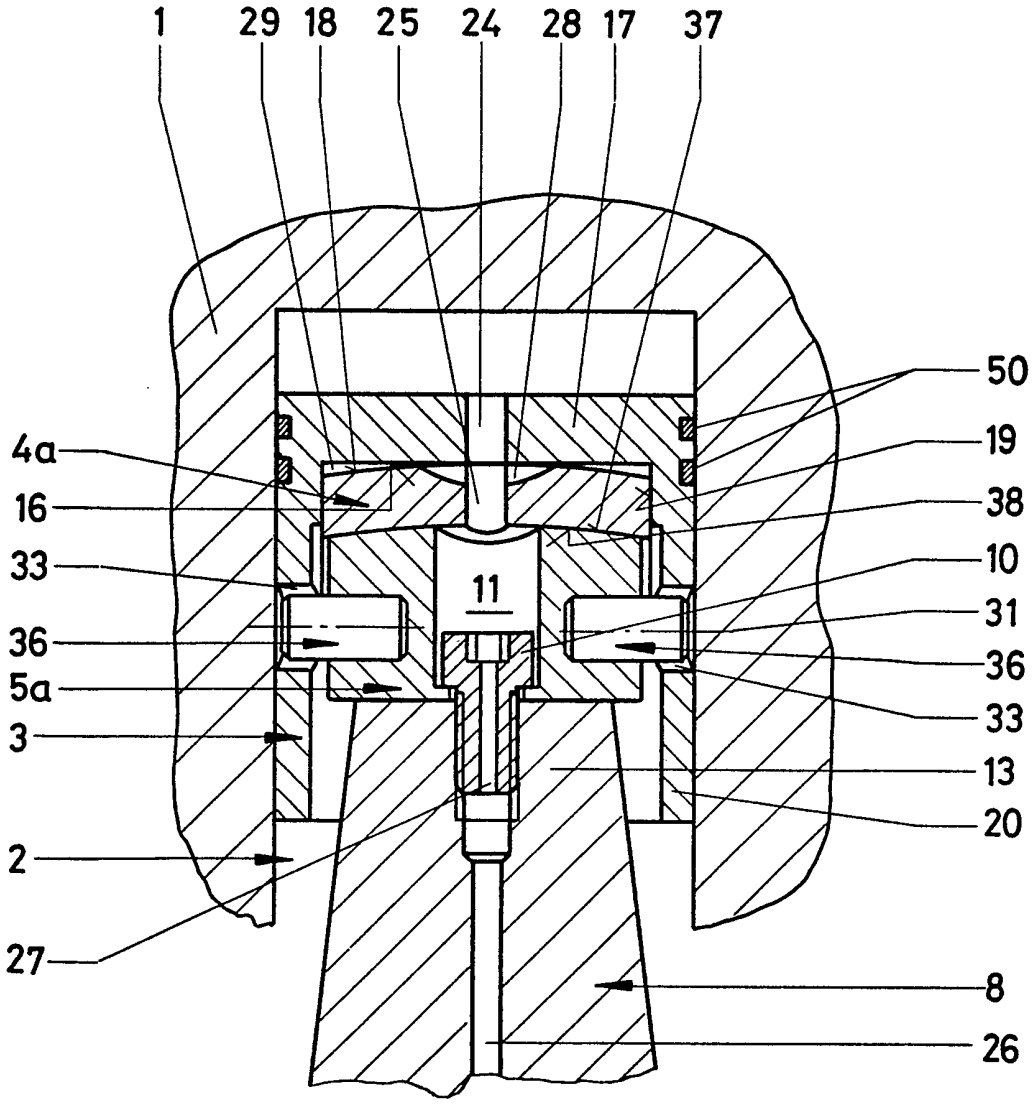

Fig.3

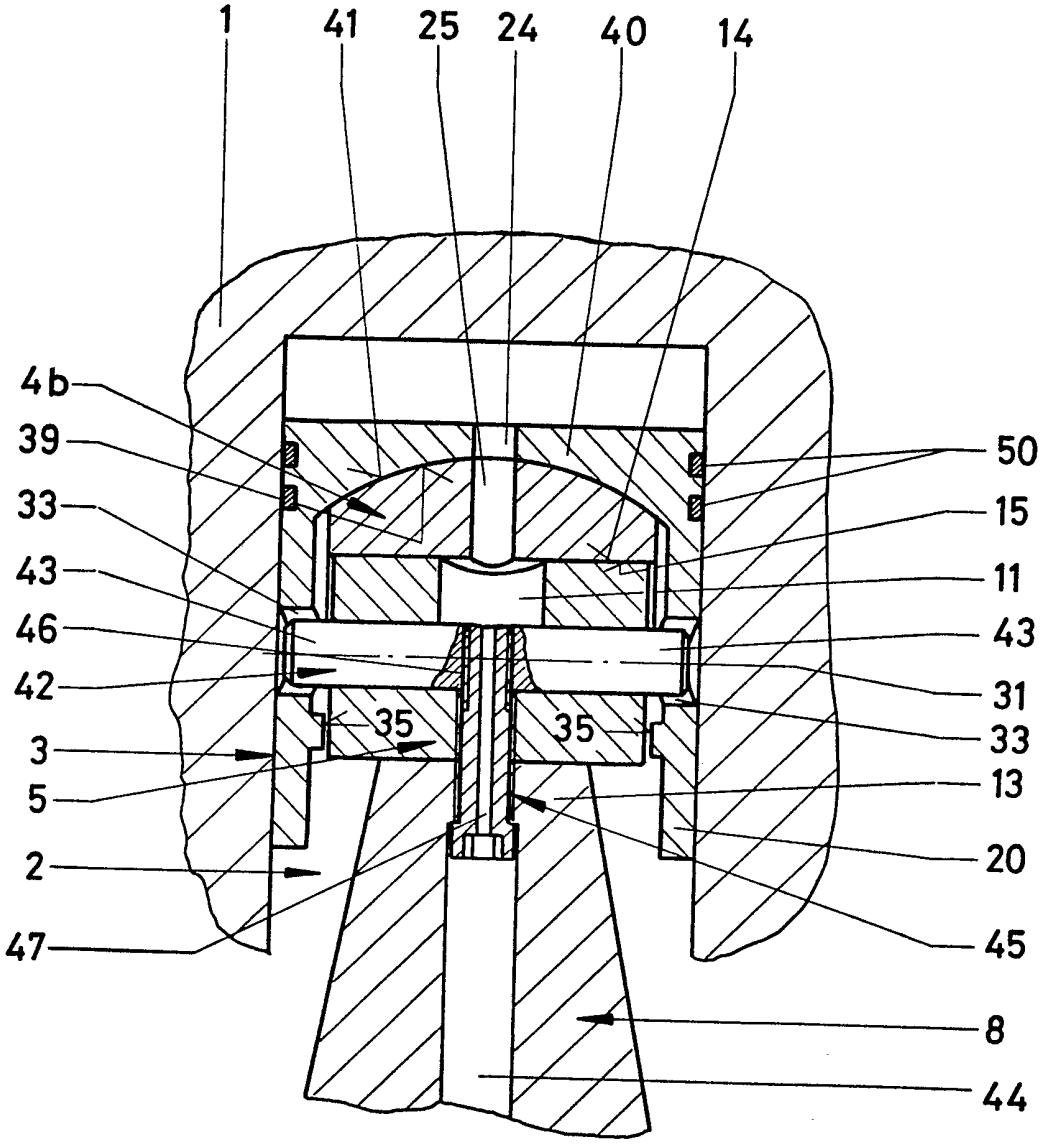
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Fig. 4

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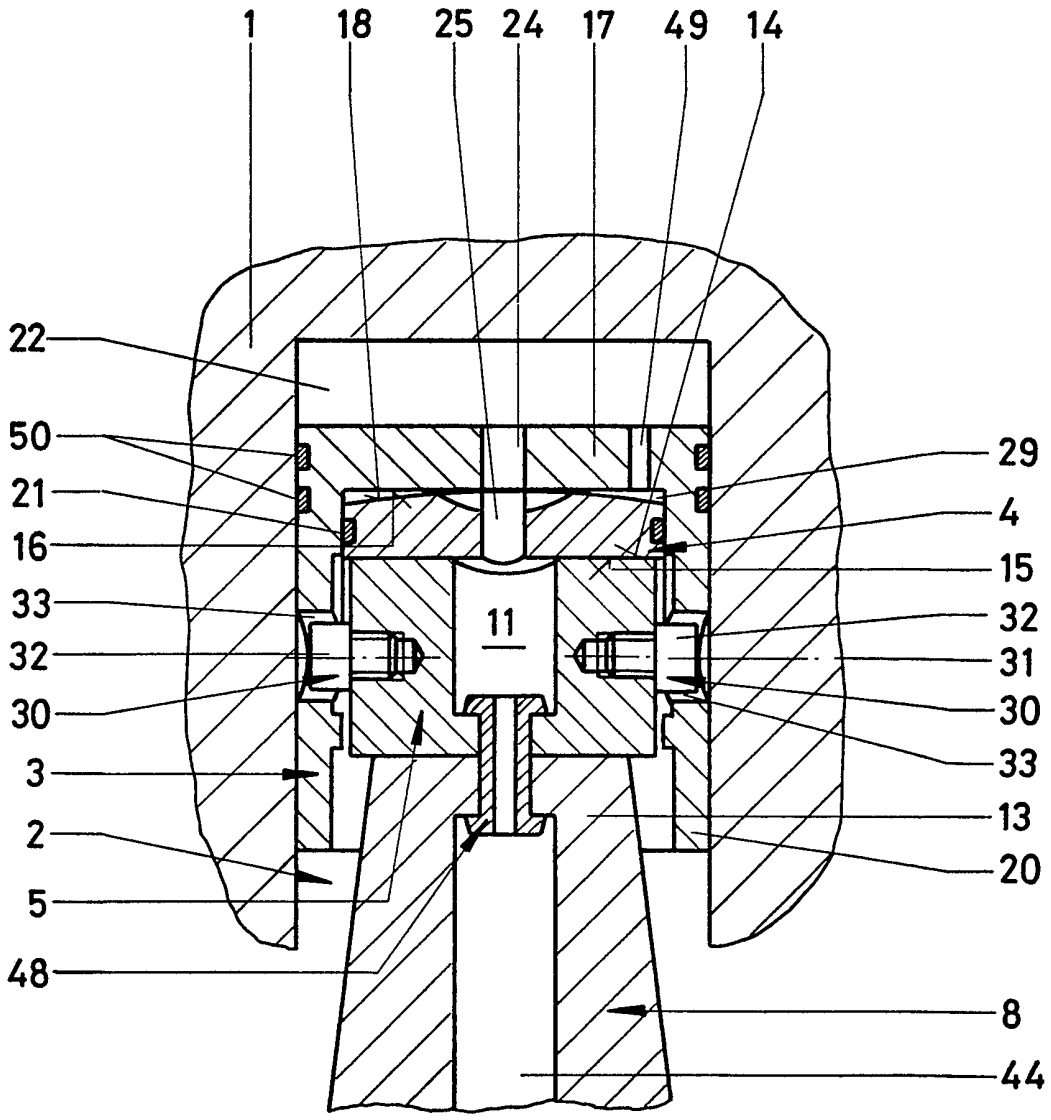

Fig. 5

SPECIFICATION

Reciprocating machine

THIS INVENTION relates to a reciprocating machine, such as a hydraulic engine, having at least one cylinder receiving, for axial sliding, a piston pivoted to a connecting rod.

In a known kind of reciprocating engine using a piston and connecting rod (Projektblatt RM 1-002 of the company styled G. Dusterloh GmbH, of 4322 Sprockhovel) the piston force is transmitted to the connecting rod by way of a cylindrical pin. The same is rigidly secured to the piston and slides in a bore, known as the little end, in the connecting-rod stem, such bore extending transversely to the direction of piston movement. The connecting rod bears by way of a shoe or pad or the like on an eccentric of the crankshaft.

An advantage of this construction is that friction is relatively reduced and the relative velocities of the components in sliding contact with one another are substantially the same everywhere. Wear in the contact zones is therefore approximately equal.

A disadvantage is the impossibility of being able to compensate for virtually unavoidable errors in the parallelism of the little end central axis to the crankshaft. Satisfactory guiding is therefore impossible. Edge running is responsible for increased leakage and irregular wear, with the possible result of unreliable operation due, for example, to the piston seizing up. A very high degree of precision in manufacture is therefore necessary if these disadvantages are to be avoided.

In another known construction (disclosed in the journal "asr.digest fur angewandte Antriebstechnik" No. 10, October 1975, page 28, Figure 2) the force of the piston is transmitted to the connecting rod by way of a ball joint, the stem of the connecting rod having disposed on it a ball which slides in a matching recess in the piston. The ball can be fashioned directly on such stem.

The advantage of this construction is that errors in parallelism cannot cause a disturbance since the joint is free to adjust itself in all directions around the centre of the ball. Manufacturing requirements are therefore not stringent.

However, friction may be greater than in the cylindrical connection hereinbefore described. Friction is considerable more particularly when the ball of a hydrostatic reciprocating engine is faceted or chamfered near the connecting rod axis to allow pressure medium to reach the load relief zone. In this case the force of the piston may be transmitted entirely by way of a spherical annular contact zone. High centrally acting forces therefore arise which therefore increase friction. The efficiency of the machine is reduced considerably. Another disadvantage is that the spherical surfaces wear unevenly because of the different relative velocities in the contact zone.

A common feature of the two kinds of reciprocating engine using pistons and connecting rods is that the joints experience high surface pressures, more particularly in the case of heavily loaded hydraulic machines. The resulting increased rate of wear and the reduced working life can be avoided only by the use of materials adapted to withstand extremely high loads. Another disadvantage is that when wear is heavy it is usually necessary to replace the piston, the piston rod and the gudgeon pin since it is not really economical to repair the worn joints.

Another disadvantage of the two constructions described is the radial forces which become increasingly perceptible with widening of the bore or the piston skirt when wear has become considerable.

It is the object of the invention to provide a reciprocating engine which combines the advantages of the ball joint connection with the advantages of the cylindrical joint connection and in which all the sliding parts of the joints are readily replaceable.

According to the invention, there is provided a reciprocating engine, more particularly a hydraulic engine, having at least one cylinder receiving, for axial sliding, a piston pivoted to a connecting rod, characterised in that a pivot member is secured to the connecting-rod and has on the side remote from the connecting-rod stem a convex circular generated surface in sliding contact with a matching concave surface in a pressure member, said pressure member having, on the side remote from the pivot member a convex circular surface which bears on an inside surface of the piston crown and the projection of the central axis of which forms an approximate right angle with the projection of the central axis of the ci rcular generated surface or of the concave surface along their common orthogonal.

A construction of this kind has the advantage of simple and low-cost repair. Only the worn parts of the joint need to be renewed and it is unnecessary to renew the piston and/or connecting rod as well. Another advantage is that, irrespective of other requirements on the materials used for the piston and the connecting rod, materials capable of transmitting very high surface pressures and specifically adapted to requirements can now be selected. A noteworthy feature is that even when wear is very advanced, there is no resilient widening of the piston skirt nor distortions of the connecting rod.

The connecting rod is secured, preferably releasably, to the pivot member at its stem end. The connection can be by way of screws which can be screwed from the pivot member into the connecting rod stem and from the latter into the pivot member. Riveted connections can also be used. The pivot member has

its convex surface engaged in a positively matching concave surface of a pressure member which bears on the piston crown and which therefore transmits the piston forces to the pivot member. When the crankshaft turns, sliding movements occur between the pivot member and the pressure member. In the event of the central axis of the convex surface of the pivot member not being parallel to the crankshaft, the connecting rod can therefore make adjusting movements transversely to the actual pivot axis, the circular surface of the pressure member rolling with very reduced friction on the inside surface of the piston crown.

The contact surfaces of the pivot member and pressure member may be cylindrical. Cylindrical contact surfaces are fairly simple to manufacture.

For accurate axial guiding of the pivot member and therefore of the connecting rod so that special abutment surfaces are unnecessary, in a preferred embodiment the contact surfaces of the pivot member are barrel-shaped.

The inside surface of the piston crown may be completely plane. If in this case the circular surface of the pressure member is cylindrical, the pressure member bears linearly on the piston crown.

Alternatively, the inside surface of the piston crown may be adapted to the cylindrical circular surface of the pressure member, whereby sliding movements can readily occur to compensate, for example, for manufacturing tolerances.

Registering passages may be provided in the piston crown, the pivot member, the pressure member and the connecting rod stem. These features can be used with advantage to enable the load relief zone in the connecting rod shoe or pad to be pressurised. The working medium can therefore flow from the cylinder chamber to such relief zone.

If the supply with the working medium makes it necessary, the securing means for the pivot member can be pierced through. The securing means then form part of the pressure supply passages.

To ensure very reduced Hertzian pressure, the entire circular surface of the pressure member advantageously experiences the pressure of the working medium in the cylinder chamber, with a considerable reduction of force. For this purpose, for example, the space or chamber bounded by the circular surface of the pressure member, by the inside surface of the piston crown, and by the piston skirt, may communicate with at least one of said registering passages. The working medium can transfer from the passages into the chamber by way of grooves in the circular surface of the pressure member.

The space or chamber bounded by the circular surface of the pressure member, by the inside surface of the piston crown and by the piston skirt may communicate directly with the cylinder chamber by way of at least one passage in the piston crown for the supply of the pressure medium. The communicating passage can extend parallel to the passages connecting the cylinder chamber to the load relief zone in the pad or shoe of the connecting rod.

Excessive leakage is obviated if the radial periphery of the pressure member is guided in sealing-tight manner in a cylindrical recess in the piston skirt.

Sealing of the gap between the periphery of the pressure member and the recess helps to enhance this advantageous feature. The gap between the pressure member periphery and the recess may be sealed by, for example, an O-ring.

In order to hold the piston and connecting rod together with the pivot member and pressure member in the assembled state and also to enable tensile forces to be transmitted between the piston and the connecting rod, at least one centering-pin secured in the pivot member is preferably disposed near the central axis of the pivot member or near the concave surface of the pressure member and engages with considerable clearance in recesses in the piston skirt, the latter recesses being aligned coaxially of one another. The clearance between the recesses in the piston skirt and the centering-pin parts engaging in this region is such as always to ensure the requisite adjustability. Conveniently, the longitudinal parts engaging in the recesses are cylindrical.

Two cheese head screws may be provided as centering-pins. Such screws are screwed in co-axially from the end faces of the pivot member and engage in the recesses in the piston skirt.

Instead of the cheese head screws cylindrical pins can be pressed in from the pivot member end faces.

Alternatively, just a single cylindrical pin may extend through the pivot member over the whole length thereof. Such pin is then secured in the pivot member, conveniently by a force fit.

However, the centering-pin can be secured by the securing means securing the pivot member to the connecting rod.

Embodiments of the invention are described below by way of example with reference to the accompanying drawings in which:

FIGURE 1 is a view in vertical longitudinal section of a hydrostatic engine embodying the invention,

FIGURE 2 is a view in vertical longitudinal section, perpendicular to Figure 1, along the line 11 -11 and

FIGURES 3, 4 and 5 are views, corresponding to Figure 2, of respective variant hydrostatic engines embodying the invention.

Referring to Figures 1 -5, a hydrostatic radial piston engine has a casing 1 in which a

number of cylindrical bores 2 are distributed in a uniform peripheral arrangement in a common plane. A hollow piston 2 slides in each cylinder bore 2 and is connected, in a manner to be described in greater detail hereinafter, by way of a pressure member 4, 4a, 4b and of a pivot member 5, 5a, to a connecting rod 8 which bears on an eccentric 6 of a crankshaft 7. The connecting rod 8 slides on the eccentric 6 with the interposition of a shoe or pad or the like 9 (Figure 1).

In the embodiment of Figures 1 and 2, a cylindrical pivot member 5 is secured to the connecting rod 8. Securing is by means of an axially pierced screw 10 which is pushed into a stepped bore 11 in the member 5 and screwed into a screwthreaded part 12 of connecting rod stem 13.

Circular generated surface 14 of member 5 is in sliding contact, on the side remote from the stem 13, with a concave surface 15 in the pressure member 4, the surface 15 being adapted to the surface 14. On the side remote from the pivot member 5 the pressure member 4 has a convex circular surface 18 which bears on an inside surface 16 of piston crown 17. Surface 16 is plane. The projection of the central axes of the surfaces 14, 18 along the common orthogonal of the central axes forms approximately a right angle, i.e. the axis of curvature of surface 14 extends in a direction perpendicular to that in which extends the axis of curvature of surface 18.

The cylindrical pressure member 4 engages by way of its periphery in a matching recess 19 in piston skirt 20. An O-ring 21 seals the gap between the member 4 and the recess 19.

The piston crown 17, pressure member 4, pivot member 5 and stem 13 have passages 24, 25, 11, 26 via which working medium can be transferred from cylinder chamber 22 to a load relief zone 23 in the shoe 9 (Figure 1). bore 27 in screw 10 serves as another transfer passage. By way of grooves 28 in the member 4 working medium can flow into a space or chamber 29 bounded by surface 18 of member 4, by surface 16 of crown 17 and by piston skirt 20. This feature reduces the Hertzian pressure.

To hold the connecting rod 8 and piston 3 together when assembled and to transmit tensile forces, cheese head screws 30 are screwed into the pivot member 5 from its end faces. The screws 30 extend along central axis 31 of surface 14 of member 5. Their cylindrical heads 32 engage with considerable clearance in coaxially registering recesses 33 in the skirt 20.

As can be gathered more particularly from Figure 2, projections 34 having abutment or striking surfaces 35 project inwardly from the skirt 20 to guide the member 5 and connecting rod 8 axially.

In the embodiment of Figure 3 cylindrical pins 36 which are pressed into the pivot member 5a from its end faces are used instead of the cheese head screws 30.

Also, circular generated surface 37 of pivot member 5a is barrel-shaped in the embodiment of Figure 3. Concave surface 38 of pressure member 4a is devised accordingly, thus rendering unnecessary the axial abutment surfaces 35 of Figure 2.

The pressure member 4a engages sealingly by way of its periphery in recess 19 in piston skirt 20. No other sealing means are present but sealing means can be provided.

In the embodiment shown in Figure 4 the surface 14 of member 5 and, therefore, the concave surface 15 of the pressure member 4b are cylindrical as before. Abutment or striking surfaces 35 on the skirt 20 for the pivot member 5 are therefore advisable. However, in contrast to the embodiments hereinbefore described, inside surface 39 of piston crown 40 is adapted to circular surface 41 of pressure member 4b. Consequently, real sliding movements are possible between the member 4b and piston crown 40.

Pivot member 5 is secured to piston 3 by a cylindrical pin 42 which extends through the member 5 over the whole length thereof. End parts 43 of pin 42 engage in recesses 33 in skirt 20 with considerable clearance. The concave inside surface 39 of piston crown 40 makes special guidance of the pressure member 4b in the piston 3 unnecessary.

Figure 4 shows another variant in connection with the securing of the member 5 and pin 42. In Figure 4 the screw 45 is introduced through an enlarged-diameter bore 44 in connecting rod stem 13 and screwed into a tapped bore 46 in pin 42. This feature locks pin 42 in member 5 and locks member 5 to connecting rod 8. Screw 45 is formed with an axial bore 47.

The embodiment of Figure 5 is very similar to the embodiment of Figures 1 and 2.

The securing of the pivot member 5 to the connecting rod 8 is dif ferent, a hollow rivet 48 being used instead of the screws 20.

Also, the space or chamber 29 communicates directly with the cylinder chamber 22 by way of at least one passage 49 in piston crown 17. This embodiment lacks the grooving in the pressure member 4 as found in the embodiments of Figures 1—3.

Passage 44 in connecting rod stem 13 corresponds approximately to that of Figure 4.

The common feature of all the embodiments is that the gap between the casing 1 and the piston 3 is sealed by ring seals 50 disposed around the periphery of the piston 3.

CLAIMS

1. A reciprocating engine, more particularly a hydraulic engine, having at least one cylinder receiving, for axial sliding, a piston pivoted to a connecting rod, characterised in that a pivot

member (5, 5a) is secured to the connecting-rod (8) and has on the side remote from the connecting-rod stem (13) a convex circular generated surface (14, 37) in sliding contact with a matching concave surface (15, 38) in a pressure member (4, 4a, 4b), said pressure member having, on the side remote from the pivot member (5, 5a), a convex circular surface (18, 41) which bears on an inside surface (16, 39) of the piston crown (17, 40) and the projection of the central axis of which forms an approximate right angle with the projection of the central axis of the circular generated surface (14, 37) or of the concave surface (15, 38) along their common orthogonal.

2. An engine according to claim 1, characterised in that the circular generated surface (14) of the pivot member (5) and/or the concave surface (15) of the pressure member (4, 4b) and/or the circular surface (41) thereof are/is cylindrical.

3. An engine according to claim 1, characterised in that the circular generated surface (37) of the pivot member (5a) and/or the concave surface (38) of the pressure member (4a) are/is barrel-shaped.

4. An engine according to claim 1, character ised in that the inside surface (16) of the piston crown (17) is plane.

5. An engine according to claims 1 and 2, characterised in that the inside surface (39) of the piston crown (40) is adapted to the cylindrical circular surface (41) of the pressure member (4b).

6. An engine according to any of claims 1—5, characterised in that registering passages (24, 25, 11, 26, 44) are provided in the piston crown (17) 40), the pivot member (5, 5a), the pressure member (4, 4a, 4b) and the connecting rod stem (13).

7. An engine according to claims 1 and 6 characterised in that the securing means (10, 45, 48) for the pivot member (5, 5a) are pierced through.

8. An engine according to claim 6 or 7, characterised in that the space or chamber (29) bounded by the circular surface (18) of the pressure member (4, 4a), by the inside surface (16) of the piston crown (17) and by the piston skirt (20) communicates with at least one of the passages (24, 25, 22, 26).

9. An engine according to claim 6 or 7, characterised in that the space or chamber (29) bounded by the circular surface (18) of the pressure member (4), by the inside surface (16) of the piston crown (17) and by the piston skirt (20) communicates with the cylinder chamber (22) by way of at least one passage (49) in the piston crown 917).

10. An engine according to any of claims 1—9, characterised in that the peripherally cylindrical pressure member (4, 4a) is engaged in a matching recess (19) in the piston skirt (20).

11. An engine according to claim 10, characterised in that the gap between the periphery of the pressure member (4, 4a) and the recess (19) is sealed.

12. An engine according to any of claims 1—11, characterised in that at least one centering pin (30, 36, 42) secured in the pivot member (5, 5a) is disposed near the central axis (31) of the pivot member (5, 5a) or near the concave surface (15, 38) of the pressure member (4, 4a, 4b) and engages with considerable clearance in recesses (33) in the piston skirt (20), the latter recesses being aligned coaxially of one another.

13. An engine according to claim 12, characterised in that two coaxially aligned cheese head screws (30) are screwed into the pivot member (5) as centering pins.

14. An engine according to claim 12, characterised in that two cylindrical pins (36) secured in a press fit are provided in the pivot member (5a) as centering pins.

15. An engine according to claim 12, characterised in that the centering pin (42) extends as a cylindrical pin through the pivot member (5) over the whole length thereof and its end parts (43) engage in the recesses (33) in the piston skirt (20).

16. An engine according to claim 12 or 15, characterised in that the centering pin (42) and the securing means (45) for the pivot member (5) are interconnected.

17. An engine according to claim 1 and substantially as hereinbefore described with ref erence to, and as shown in, Figures 1 and 2 of the accompanying drawings.

18. An engine according to claim 1 and substantial ly as hereinbefore described with reference to, and as shown in, Figure 3 of the accompanying drawings.

19. An engine according to claim 1 and substantially as hereinbefore described with reference to, and as shown in, Figure 4 of the accompanying drawings.

20. An engine according to claim 1 and substantially as hereinbefore described with reference to, and as shown in, Figure 5 of the accompanying drawings.

21. Any novel feature or combination of features described herein.

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