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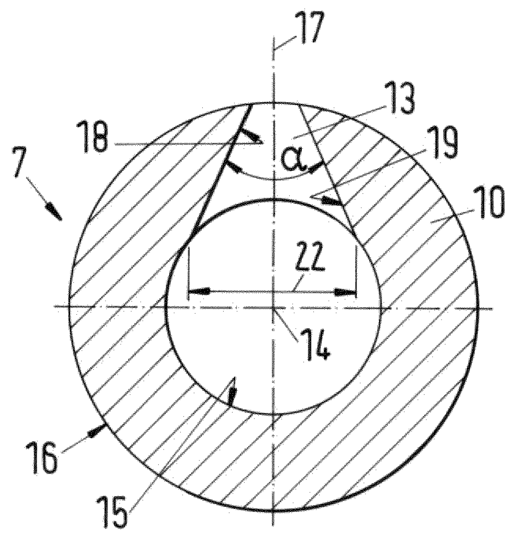
(54) **A high pressure fluid rail**

(57) A high pressure fluid feed device (1) for a common rail system for a large internal combustion engine comprises a pressure accumulator unit (7) for supplying high pressure fluid to a plurality of fluid injectors (8), and the pressure accumulator unit (7) has an outer circumferential surface (16) and is disposed with a central bore (11) substantially extending along the longitudinal axis (14) of the pressure accumulator unit (7) and having an inner circumferential surface (15) and at least one radial bore (13) extending from the central bore (11) to the outer circumferential surface (16) of the pressure accumulator unit (7). The radial bore (13) has a width (22) measured in a plane normal to the longitudinal axis (14) of the pressure accumulator unit (7), which is greater at the inner circumferential surface (15) than at the outer circumferential surface (16) and decreasing continuously from the inner circumferential surface (15) to the outer circumferential surface (16) preferably for at least half of the distance between the inner circumferential surface (15) and the outer circumferential surface (16), and the radial bore (13) has a first lateral wall (18) and a second lateral wall (19), whereby the width (22) is measured as the distance between the first and second lateral walls (18, 19) of the radial bore (13) in a plane containing the central axis (17) of the radial bore and arranged normally to the longitudinal axis (14) of the pressure accumulator unit (7).

Fig.6
A-A



EP 2 511 517 A1

Description

BACKGROUND OF THE INVENTION

[0001] The invention relates to a high pressure fluid rail for storage of a high pressure fluid to be employed in particular for a piston engine.

BACKGROUND ART

[0002] In common rail fuel injection systems of diesel engines the fuel is fed by means of low-pressure and high-pressure pumps into a high-pressure fuel storage, such as a high-pressure accumulator or a so-called common rail. From the high-pressure fuel storage the fuel is fed further along separate conduits to the fuel injector of each cylinder. From the fuel injector the fuel is led to the respective combustion chamber of the cylinder at a desired moment according to the operation of the engine. There may be several high-pressure fuel storages, whereby fuel is fed from each fuel storage to two or more injection nozzles.

[0003] The fuel pressure in a high-pressure fuel storage is high, even 2000 bar, whereby the fuel accumulator is exposed to strong stresses resulting in cracks that may develop in its construction material, especially in the area of the openings in the fuel storage wall and sharp-edged cross-sectional changes thereof.

[0004] The highest pressures typically occur in the common rail system for fuel injection. Due to the fact that the high fluid pressure of up to 2000 bar, the dynamic loads, which are caused by the variation of this high fluid pressure result in high stresses the walls of the fuel space have to sustain for a prolonged operation period. In particular for marine engines, it is essential, that a prolonged operation period can be guaranteed as the rail should not be replaced or repaired during the lifetime of the engine.

[0005] The strength of the walls of the fuel space is in particular limited by radial bores extending from the central bore. The local stresses introduced by these radial bores can be increased by a factor of three or four.

[0006] The document EP 1413744 discloses a fuel storage for a piston engine, where the fuel space consists of two elongated cylindrical bores, which are arranged so as to partially overlap in cross-section thereby forming a main bore. The main bore in the rail of the Wärtsilä RT-flex engines is machined with two eccentric bores resulting in a cross-section of the main bore having a peanut shape. From the main bore, an auxiliary bore connects the fuel space to the fuel injector supply conduit. The junction drilling for this auxiliary bore is located at the intersection of the two eccentric bores. Thereby the stress at the crossing between the junction drilling and the two main rail drillings is reduced, since the crossing between the bores is not directly exposed to the main force lines.

[0007] The document W02008/145818 discloses a fuel storage having an orifice opening into the fuel space which is configured such that the stresses exerted on the fuel storage in the area of orifice are reduced. Thereby the risk of crack formation in the construction material of the fuel space body is reduced. Consequently the durability of the fuel storage is increased.

[0008] However, the solution proposed in W02008/145818 suffers from some disadvantages preventing its use in fuel rails of substantial length. The recess has to be machined from the interior, that means from the central bore. Due to the fact that the tool for machining the recess can protrude only a small distance into the central bore, such a solution is not applicable to long, thin fluid spaces having a total length of at least on meter while having a bore diameter of the central bore of at most 100 mm.

[0009] An object of the invention is to provide a fluid space having one or more auxiliary bores which can be employed in such long, thin fluid spaces.

BRIEF SUMMARY OF THE INVENTION

[0010] The objects of the invention can be obtained by the features of claim 1. Advantageous embodiments of the invention are subject of the dependent claims.

[0011] A high pressure fluid feed device for a common rail system for a large internal combustion engine comprises a pressure accumulator unit for supplying high pressure fluid to a plurality of fluid injectors. The pressure accumulator unit has an outer circumferential surface and is disposed with a central bore substantially extending along the longitudinal axis of the pressure accumulator unit and an inner circumferential surface. At least one radial bore extends from the central bore to the outer circumferential surface of the pressure accumulator unit. The radial bore has a width measured in a plane normal to the longitudinal axis of the pressure accumulator unit which is greater at the inner circumferential surface than at the outer circumferential surface and the width decreases continuously from the inner circumferential surface to the outer circumferential surface preferably for at least half of the distance between the inner circumferential surface and the outer circumferential surface. The radial bore has a first lateral wall and a second lateral wall. The width is measured as the distance between the first and second lateral walls of the radial bore in a plane containing the central axis of the radial bore and arranged normally the longitudinal axis of the pressure accumulator unit.

[0012] According to an embodiment, the continuous decrease is substantially linear. Thereby it is intended that in a section along a plane normal to the longitudinal axis of the pressure accumulator unit, which contains the axis of the radial bore, the first lateral wall and the second lateral wall are represented as a straight line. the opening angle between the first and second lateral wall is in a range of 10 to 75 °, preferably 10 to 60° particularly preferred 10 to 45°. The opening angle is constant from the inner circumferential surface to the outer circumferential surface.

[0013] The width of the radial bore at the inner circumferential surface is at least twice the width at the outer circumferential surface. The width of the radial bore at the inner circumferential surface is measured in a projection onto a plane containing the longitudinal axis of the pressure accumulator unit when viewed in the direction of the axis of the radial bore. The width is the distance from the first lateral wall to the second lateral wall.

[0014] If the width at the inner circumferential surface is considerably greater than the width at the outer circumferential surface, thus at least twice the width at the outer circumferential surface, the stresses acting upon the inner circumferential surface of the pressure accumulator unit in the surroundings of the radial bore can be distributed over a considerably larger surface as compared to a cylindrical bore as disclosed e.g. in EP1426607 A1. Furthermore due to the fact that the first lateral wall extending between the inner circumferential surface and the outer circumferential surface has a constant angle of inclination, the stresses are reduced also in a particular smooth way due to the fact that no local sharp transition of the wall surface is present, such as in W02008/145818.

[0015] In an embodiment, the central bore has a circular cross-section. Such a central bore is particularly easy to manufacture, which means that a length of the pressure accumulator unit of a couple of meters is possible and can be manufactured at a low cost. Due to the fact that the pressure of the high pressure fluid can be in the range of 600 to 2000 bar, a central bore of circular cross-section is the most advantageous shape for pressure distribution. However it is also possible to use an oval shape or a shape such as disclosed in EP1 413 744 A1.

[0016] The radial bore has a thickness of approximately the same size as the width at the outer circumferential surface. The thickness of the bore is measured in a direction normal to the width. The thickness is measured in the direction of the longitudinal axis of the pressure accumulator unit. By keeping the width and thickness of the bore at the outer circumferential surface small, the stresses acting on the tube walls in the vicinity of the outer circumferential surface are reduced. Moreover a minimum of material is to be disposed of which on the one hand reduces waste and surprisingly has also a beneficial effect on the stress distribution. Therefore in an embodiment, the radial bore has a thickness at the inner circumferential surface which deviates from the thickness at the outer circumferential surface not more than 15 %, preferably not more than 10 % particularly preferred not more than 5 %.

[0017] The high pressure fluid feed device in accordance with one of the embodiments can be used advantageously for fuel or a servo oil or water as high pressure fluid.

[0018] An internal combustion engine for which the pressure fluid feed is most advantageously employed includes a cylinder, in which a piston is arranged to be movable along a longitudinal cylinder axis to and fro between a top dead centre position and a bottom dead centre position. In the cylinder, a combustion chamber is confined by a cylinder cover, by a cylinder wall of the cylinder and by a piston surface of the piston. A fuel injector is provided for injecting fuel into the combustion chamber and a high pressure fluid feed device in accordance with one of the preceding embodiments can be used to supply the high pressure fuel to the fuel injector. The combustion engine can be configured as a crosshead engine, in particular a crosshead large diesel engine, a trunk piston engine, a two-stroke or a four-stroke internal combustion engine, a dual fuel engine being operable either in diesel or otto mode or a gas engine.

[0019] A method for manufacturing a high pressure fluid feed device for any of the embodiments comprises the step of milling or of drilling the radial bore. If the radial bore is drilled, the drilling is performed by drilling a central radial bore and at least a further side radial bore, whereby the drill hole of the central radial bore and the side radial bore is the same on the outer circumferential surface and is inclined with respect to the central radial bore hole at the inner circumferential surface. The milling or drilling step is performed from the outer circumferential surface to the inner circumferential surface of the pressure accumulator unit, which contains the central bore. Surprisingly it is possible to perform the milling or drilling step from the outside of the pressure accumulator unit. This is an unexpected advantage as compared to the prior art as disclosed in W02008/145818 or also EP2299102. The recesses in these disclosures have to be machined from the central bore which is considerably more cumbersome. Furthermore it is not possible to introduce a drilling or milling tool into a pressure accumulator unit formed as a unitary long tube. Therefore a plurality of pressure accumulator units of a small length have to be used as shown in W02008/145818. This has the consequence that not only the assembly of these prior art pressure accumulator units becomes more difficult and time consuming, but also the control thereof.

[0020] Thus, the pressure accumulator unit can be particularly advantageously configured as an elongated tube. The fluid space in the interior of the central bore can be a fuel space, a servo fluid space, but also a gas space or a space used for water injection. The fluid space is surrounded by the inner circumferential surface and at least one of the ends, the bore can be closed by a closure stopper.

[0021] At least one of the outer circumferential surfaces or inner circumferential surfaces of the body of the pressure accumulator unit can have a circular, oval shaped or also a polygonal cross-section, for example a rectangular in particular square cross section, a triagonal or hexagonal cross-section. Furthermore the outer circumferential surface does not need to be concentric to the inner circumferential surface. Thus the central bore can have a longitudinal axis which is not the same as the longitudinal axis of the body of the pressure accumulator unit.

[0022] By means of pumps, a fuel for the injection, a hydraulic fluid for e.g. actuation of a valve or a working medium for controlling e.g. a fuel injector is fed under high pressure from the fluid space to the desired use. The fluid space forms a central bore in the pressure accumulator unit. The pressure accumulator unit forms a common rail for supplying the high pressure fluid to a plurality of users. For this reason a plurality of radial bores can be foreseen. In case of a fuel injection common rail system, fuel is distributed by the common rail via fuel conduits to the fuel injectors located at each cylinder. The fuel injector contains a fuel injection nozzle for supplying the fuel to the combustion chamber of each cylinder.

BRIEF DESCRIPTION OF THE DRAWINGS

[0023] In the following the invention will be described with reference to the accompanying schematic drawings

Fig. 1 shows an arrangement of a high pressure fluid feed device for a common rail system for fuel injection for a large internal combustion engine;

Fig. 2 shows a section of a pressure accumulator unit taken along its longitudinal axis;

Fig. 3 shows section B-B of Fig. 2;

Fig. 4 shows view C of a radial bore manufactured by drilling;

Fig. 5 shows view C of a radial bore manufactured by milling;

Fig. 6 shows a sectional view of a pressure accumulator unit along a plane normal to its longitudinal axis;

Fig. 7 is a view on a radial bore from the outer circumferential surface of the pressure accumulator unit.

DETAILED DESCRIPTION OF THE INVENTION

[0024] Figure 1 shows a fuel feed device 1 of a diesel engine comprising several cylinders, especially a large diesel engine. Large diesel engine refers here to such engines that can be used, e.g., as main or auxiliary engines in ships or in power plants for the production of electricity and/or heat. Fuel is fed from a fuel tank 2 by a pump 4 along a piping 3 to a high pressure fuel storage 5. The high pressure fuel storage 5 is configured as a high pressure accumulator unit 7.

[0025] The flow rate of the pump can be regulated by an electronic control unit 20 based on the amount of fuel required by the fuel injectors 8. The electronic control unit provides the timing for the operation of the fuel injectors 8 based on engine parameters, such as load of the engine or engine speed. Furthermore the control unit receives input from a pressure sensor 21 mounted at the pressure accumulator unit 7. The pressure sensor 21 detects the fluid pressure in the pressure accumulator unit 7. Based on the detected pressure value, the operation of the pump is controlled, e.g. by controlling the rotational speed of a pump motor 24.

[0026] According to an alternative embodiment, a plurality of high-pressure pumps 4 can be provided. Each of the pumps can be provided with control valves and piston members (not shown). The piston members can receive their guidance from cam members of a camshaft of the engine. When necessary, each cam member may include several cams, and thereby, when a high-pressure pump 4 provides a certain volume flow rate per time unit into a pressure accumulator unit 7, the outer dimensions of the pump may respectively be kept smaller and accordingly, the pressure shocks provided by it to the pressure accumulator are smaller.

[0027] The pressure accumulator 7 unit is, in turn, connected by separate fuel injection conduits 9 to the fuel injectors 8 of the cylinders. The pressure accumulator unit 7 is thus connected to two or more fuel injectors 8.

[0028] The fuel pressure in the pressure accumulator unit 7 is at least 600 bar, typically 1000 - 2000 bar. The operation of high-pressure pumps 4 and the injection pressures to be used can be controlled in accordance with the engine load, operating speed or other parameters in a manner known as such.

[0029] Fig. 1 shows, in a schematic illustration a cylinder arrangement 100 according to the invention with the fuel injection device according to the invention for an internal combustion engine which is in the present example a longitudinally scavenged crosshead large diesel engine. The cylinder arrangement 100 according to Fig. 1 includes a cylinder 101 in which a piston 103 is arranged to be movable along a longitudinal cylinder axis 107 to and fro between a top dead centre position and a bottom dead centre position. In the cylinder 101 a combustion chamber 104 is defined by a cylinder cover 102, by a cylinder wall 105 of the cylinder 101 and by a piston surface 106 of the piston 103. Only one charge-cycle valve 109 being here an outlet valve is provided in a gas exchange opening 110 of the cylinder cover 102 which gas exchange opening 110 is connected via a gas feeding conduct 111 I to a not shown turbocharger assembly in a per se known manner. The charge-cycle valve 109 includes a valve disk 112 cooperating in the operation state with a valve seat 113 of the gas exchange opening 110 in such a way, that in a closed position of the charge-cycle valve 109, the combustion chamber 104 is sealed with respect to the gas feeding conduct 111, wherein in an open position of the charge-cycle valve 109 combustion gases can be fed out of the combustion chamber 104 to the turbocharger assembly.

[0030] Fig. 2 shows a section of a pressure accumulator unit 7 taken along its longitudinal axis. The pressure accumulator unit 7 comprises a body 10 with an elongated fuel space for pressurised fuel. The fuel space comprises a central bore 11 of cylindrical shape. As shown in Fig. 6, the cross-section of the central bore 11 is circular. The central bore 11 extends along the longitudinal axis of the pressure accumulator unit 7. Typically, the diameter of the bore is in the range of 20 to 100 mm. In addition, the body 10 comprises a feed channel (not shown in Fig. 2) from piping 3, which opens into the fuel space. Pressurised fuel is led into the fuel space through the feed channel from the high-pressure pump 4 along the piping 3. In addition, the body 10 comprises discharge channels, which open into the fuel space, through which channels fuel is discharged from the fuel space and led along to the injection nozzles of the fuel injectors 8 (see Fig. 1). A separate discharge channel is foreseen each injection nozzle, which is in flow connection with the pressure accumulator unit 7. A radial bore 13 is foreseen for each discharge channel. The radial bore extends from the inner circumferential surface 15 central bore 11 to the outer circumferential surface 16 of the body 10. Each of the radial bores 13 has a longitudinal axis 17.

[0031] Due to the high fuel pressure prevailing in the fuel space, cracks and failures due to fatigue of the material may develop in the material of the body 10, especially in the area of the radial bores 13 for the discharge channels. In order to prevent such damages to the pressure accumulator unit, the radial bore 13 has a width measured in a plane normal to the longitudinal axis 14 of the pressure accumulator unit 7, which is greater at the inner circumferential surface 15 than at the outer circumferential surface 16 and is decreasing continuously from the inner circumferential surface 15 to the outer circumferential surface 16 as shown in Figs 3 to 7. According to Fig. 3 or Fig. 6, the radial bore has a first lateral wall 18 and a second lateral wall 19.

[0032] Fig. 3 shows section B-B of Fig. 2. The radial bore 13 shown in Fig. 3 is manufactured by a drilling method. The drilling is performed by drilling a central radial bore 25 and at a first and a second side radial bore 26, 27. The central drill hole 28 of the central radial bore 25 and the first and second side radial bore 26, 27 is the same on the outer circumferential surface 16 of the body 10. The first and second side radial bores 26, 27 are inclined with respect to the central radial bore 25. Therefore the first and second side drill holes 29, 30 of the first and second side radial bores 26, 27 only partially overlap with the central radial bore 25 at the inner circumferential surface 15.

[0033] Fig. 4 shows view C of a radial bore manufactured by drilling, which is essentially the radial bore shown in Fig. 3. A first and a second side wall 31, 32 extend between the first and second lateral walls 18, 19. Due to the circular shape of the drill, each of the central radial bore, the first and the second side radial bore is of cylindrical shape. Due to the fact that at the level of the inner circumferential surface 15, the central radial bore, the first and the second side radial bore do not overlap completely anymore, the first and second side walls 31, 32 contain protrusions and recesses. The thickness of the radial bore at the inner circumferential surface is defined as the maximal distance between the first and second side walls 31, 32, which corresponds to the diameter of one of the central radial bore, first side radial bore or second side radial bore. The thickness 23 of the bore 13 at the inner circumferential surface 15 is thus substantially the same as the thickness of the bore 13 at the outer circumferential surface 16. In this embodiment, the thickness at the inner circumferential surface 15 is at no location greater than the thickness at the outer circumferential surface 16.

[0034] Fig. 5 shows view C of a radial bore 13 manufactured by milling. In this case the side walls 31, 32 do not have any protrusions or recesses. The thickness 23 of the bore 13 corresponds to the diameter of the bore 13. The thickness 23 of the bore 13 at the inner circumferential surface 15 is thus substantially the same as the thickness of the bore 13 at the outer circumferential surface 16.

[0035] However it would also be possible that the thickness at the inner circumferential surface 15 is at least at one location somewhat greater than the thickness at the outer circumferential surface 16. The advantageous distribution of stresses can be observed also if the thickness 23 of the bore deviates from the thickness at the outer circumferential surface not more than 15 %, preferably not more than 10 % particularly preferred not more than 5 %.

[0036] Fig. 6 shows a sectional view of a pressure accumulator unit 7 along a plane normal to its longitudinal axis 14. The width 22 of the radial bore13 is measured as the distance between the first and second lateral walls 18, 19 of the radial bore in a plane containing the central axis 17 of the radial bore 13 and arranged normally to the longitudinal axis 14 of the pressure accumulator unit 7. The width is greater at the inner circumferential surface 15 than at the outer circumferential surface 16 and decreases continuously from the inner circumferential surface 15 to the outer circumferential surface 16. The decrease of the width occurs progressively, it is thus substantially linear. In the section shown in Fig. 6, the decrease is shown as a straight line.

[0037] Fig. 7 is a view on a radial bore from the outer circumferential surface of the pressure accumulator unit. The width 22 of the radial bore 11 at the inner circumferential surface 15 is at least twice the width of the radial bore 11 at the outer circumferential surface 16.

[0038] In a further embodiment, which is shown in Fig. 8 a flat surface 33 may be machined on the outer circumferential surface 16. Thus a segment of the body 10 of the pressure accumulator unit 7 may be removed. Thereby the outer circumferential surface 16 of the body 10 is flattened at the location of the radial bore 13. Advantageously the maximal thickness of the segment is not more than 30% of the distance between the inner and outer circumferential surface, preferably not more than 25% of the distance, particularly preferred not more than 15% of the distance. The maximal thickness is the distance from the raw outer circumferential surface before machining to the flat surface measured in an angle of 90° from the flat surface. In other words, the maximal thickness is the shortest distance between the flat surface and the parallel plane forming a tangential plane to the raw outer circumferential surface.

[0039] The bore may also be disposed with a recess on the outer circumferential surface 16. Two examples of such recesses are shown in Fig. 9 and Fig. 10. Such a recess may be used to attach an adapter element for a high pressure conduit leaving the radial bore 13. According to Fig. 9, the recess 34 is of conical shape. According to Fig. 10, the recess 35 is of cylindrical shape. Advantageously the depth of the recess is not more than 30% of the distance between the inner and outer circumferential surface, preferably not more than 25% of the distance, particularly preferred not more than 15% of the distance if the recess is machined directly into the raw outer circumferential surface.

[0040] If a recess is foreseen in combination with a flat surface as shown in Fig. 9 and Fig. 10, then the depth of the segment removed for obtaining the flat surface 33 and the depth of the recess 34, 35 in combination should be not more than 30% of the distance between the inner circumferential surface 15 and the outer circumferential surface 16, preferably not more than 25% of the distance, particularly preferred not more than 15% of the distance.

[0041] In an embodiment according to Fig. 11 there is shown a section through a body 10 of a pressure accumulator unit 7. The outer circumferential surface 16 does not need to be concentric to the inner circumferential surface 15. Thus the central bore 11 has a longitudinal axis 36 which does not correspond to the longitudinal axis 14 of the body 10 of the pressure accumulator unit 7. The longitudinal axis 14 can be parallel to the longitudinal axis 36 as shown in Fig. 11, however it is also possible, that the longitudinal axis 14 and the longitudinal axis 36 form a staggered arrangement, are e.g. arranged in an angle to each other

[0042] In Fig. 12 a section through a body 10 of a pressure accumulator 7 of a further embodiment is shown. In this case the outer circumferential surface is square-shaped. The inner circumferential surface 15 is circular. Alternatively, at least one of the outer circumferential surfaces 16 and the inner circumferential surfaces 15 can have a polygonal cross-section, e.g. triangular, rectangular or hexagonal cross-section.

[0043] From the foregoing it should not be construed, that the invention is in any way limited to the embodiments according to the figures. It can be implemented in many other different embodiments within the scope of the inventive idea and the attached claims. The reference numbers in the figures are the same for parts having the same function.

[0044] Tensions in the basic body of a pressure reservoir of a common rail system, which occur due to the high difference in pressure or through dynamic pressure fluctuations in the system can be clearly reduced in a simple manner by means of the present invention. In particular in the region of the through bores, through which a medium which is under high pressure, can be extracted, stress increases in the wall of the pressure reservoir are lowered significantly. By this means, for a given fatigue strength of the material from which the basic body of the pressure reservoir is constructed, the permissible interior pressure and the band width of dynamic pressure fluctuations can be raised significantly and/or the outer dimensions of the pressure reservoir, in particular its diameter, can be reduced because, for example, the wall of the basic body can be made thinner. Thus not only a remarkable gain in space occurs in the operating state and a considerable saving in weight from up to several 100 kilos, but also the pressure reservoir can be manufactured considerably more cheaply.

Claims

1. A high pressure fluid feed device (1) for a common rail system for a large internal combustion engine, comprising a pressure accumulator unit (7) for supplying high pressure fluid to a plurality of fluid injectors (8), and the pressure accumulator unit (7) has an outer circumferential surface (16) and is disposed with a central bore (11) substantially extending along the longitudinal axis (14) of the pressure accumulator unit (7) and having an inner circumferential surface (15) and at least one radial bore (13) extending from the central bore (11) to the outer circumferential surface (16) of the pressure accumulator unit (7), **characterised in that** the radial bore (13) has a width (22) measured in a plane normal to the longitudinal axis (14) of the pressure accumulator unit (7), which is greater at the inner circumferential surface (15) than at the outer circumferential surface (16) and decreasing continuously from the inner circumferential surface (15) to the outer circumferential surface (16) preferably for at least half of the distance between the inner circumferential surface (15) and the outer circumferential surface (16) and the radial bore (13) has a first lateral wall (18) and a second lateral wall (19), whereby the width (22) is measured as the distance between the first and second lateral walls (18, 19) of the radial bore (13) in a plane containing the central axis (17) of the radial bore and arranged normally to the longitudinal axis (14) of the pressure accumulator unit (7).

2. A high pressure fluid feed device (1) in accordance with claim 1, wherein the continuous decrease is substantially linear.

3. A high pressure fluid feed device (1) in accordance with claim 1 or 2, wherein the width (22) of the radial bore (11) at the inner circumferential surface (15) is at least twice the width of the radial bore (11) at the outer circumferential surface (16).

4. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the central bore (11) has a circular cross-section.

5. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the pressure of the high pressure fluid is in the range of 600 to 2000 bar.

6. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the radial bore (13) has at the outer circumferential surface (16) a thickness of approximately the same size as the width at the outer circumferential surface.

7. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the radial bore has at the inner circumferential surface a thickness which deviates from the thickness at the outer circumferential surface not more than 15 %, preferably not more than 10 % particularly preferred not more than 5 %.

8. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the high pressure fluid is a fuel.

9. A high pressure fluid feed device (1) in accordance with any one of the preceding claims, wherein the high pressure fluid is a servo oil or water.

10. An internal combustion engine (100), including a cylinder (101), in which a piston (103) is arranged to be movable along a longitudinal cylinder axis (105) to and fro between a top dead centre position and a bottom dead centre position, wherein in the cylinder (101) a combustion chamber (104) is confined by a cylinder cover(102), by a cylinder wall (105) of the cylinder (101) and by a piston surface (106) of the piston (103) and a fuel injector is provided for injecting fuel into the combustion chamber, including a high pressure fluid feed device (2) in accordance with one of the preceding claims.

11. An internal combustion engine (100) according to claim 10, wherein the combustion engine is a crosshead engine, in particular a crosshead large diesel engine, a trunk piston engine, a two-stroke or a four-stroke internal combustion engine, a dual fuel engine being operable either in diesel or otto mode or a gas engine.

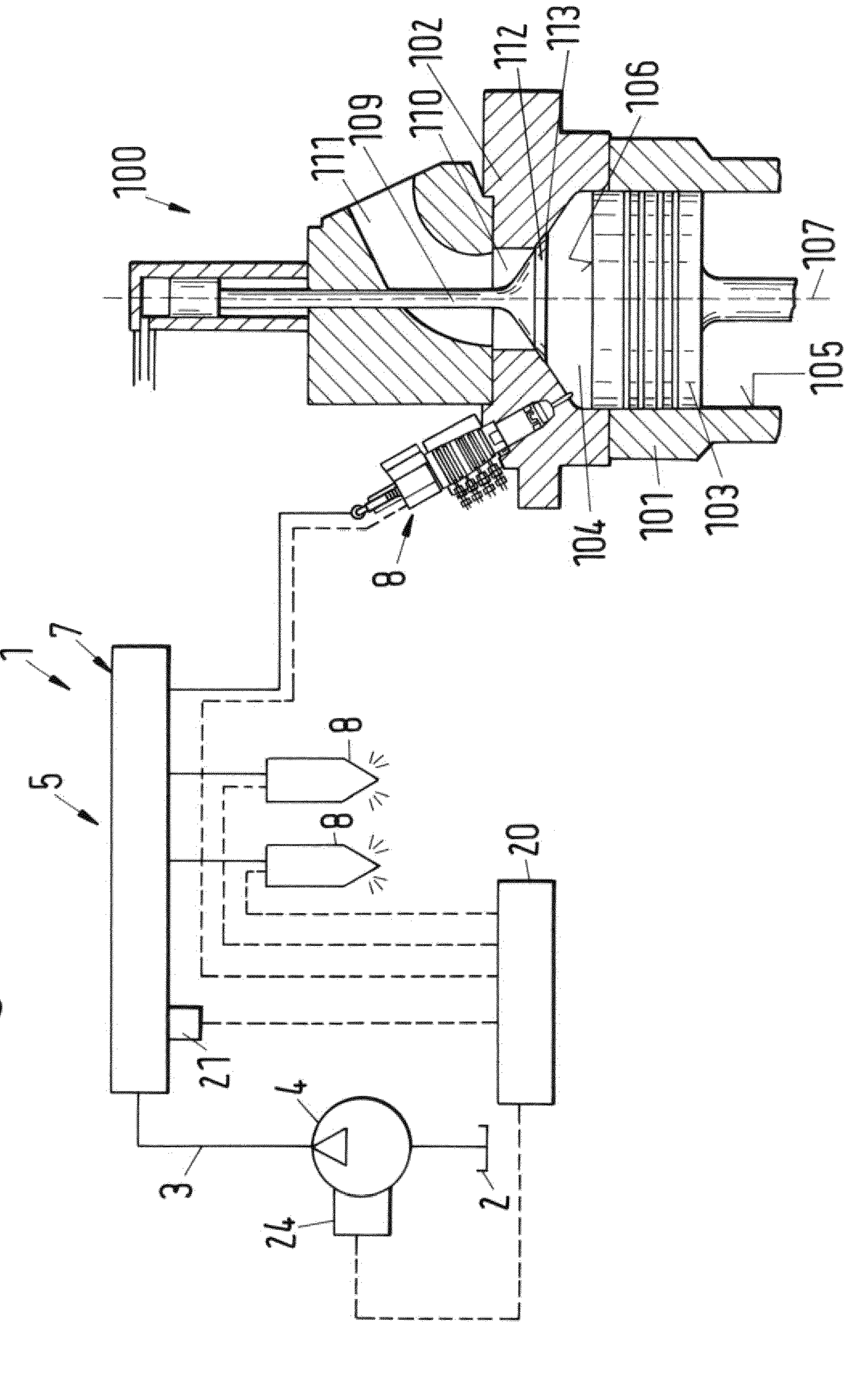
12. A method for manufacturing a high pressure fluid feed device (1) according to any one of claims 1 to 9 comprises the step of milling the radial bore (13).

13. A method for manufacturing a high pressure fluid feed device (1) according to any one of claims 1 to 9 comprises the step of drilling the radial bore (13).

14. A method for manufacturing a high pressure fluid feed device (1) according to claim 13, wherein the drilling is performed by drilling a central radial bore (25) and at least a further side radial bore (26, 27), whereby the drill hole (28) of the central radial bore (25) and the side radial bore (26, 27) is the same on the outer circumferential surface (16) and is inclined with respect to the central radial bore hole at the inner circumferential surface (15).

15. A method for manufacturing a high pressure fluid feed device (1) according to any one of claims 12 to 14, wherein the milling or drilling step is performed from the outer circumferential surface (16) to the inner circumferential surface (15) of the pressure accumulator unit (7), which contains the central bore (13, 25).

Fig.1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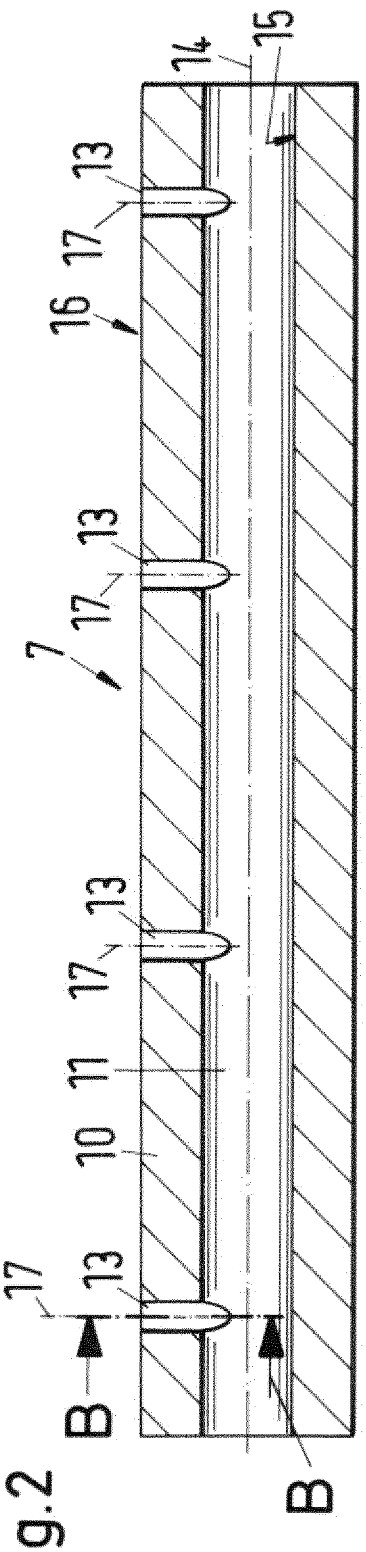


Fig.2

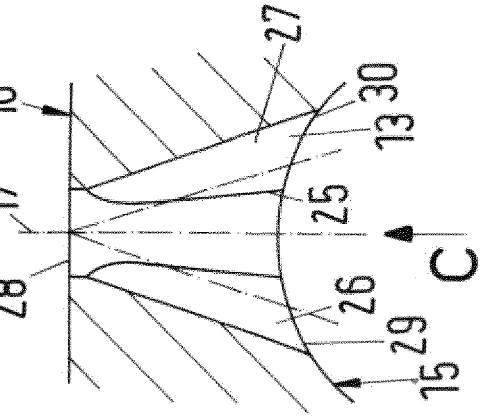


Fig.3 B-B

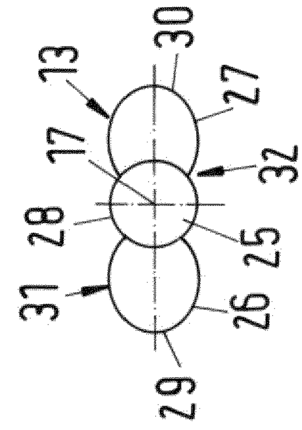


Fig.4 C

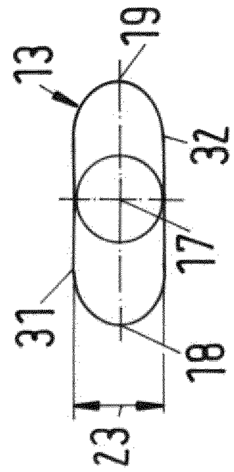


Fig.5

Fig.6
A-A

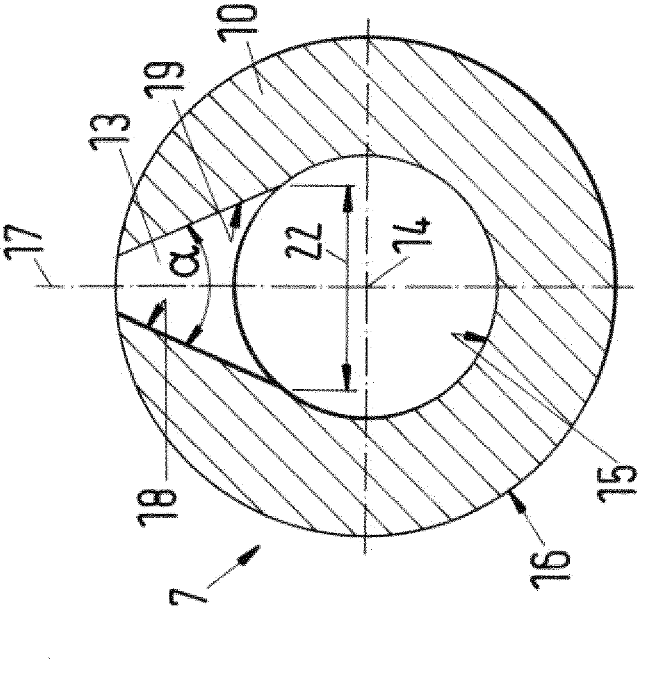


Fig.7

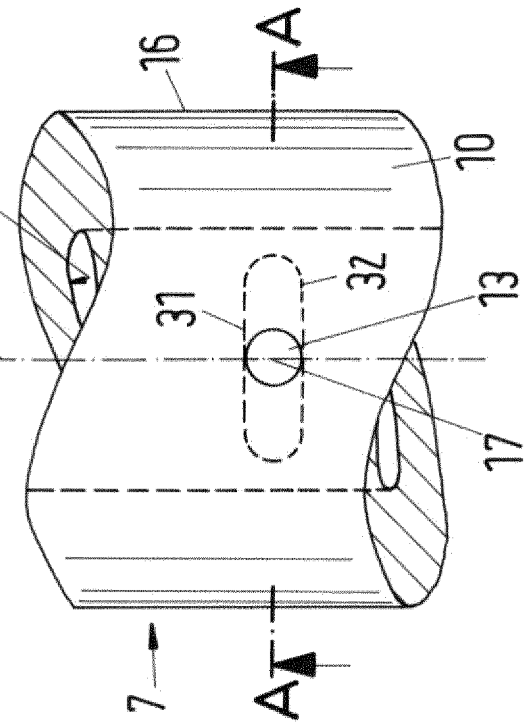


Fig.9

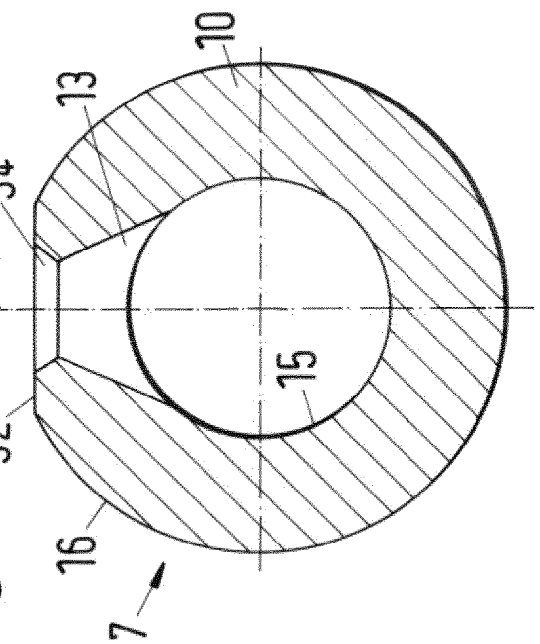


Fig.10

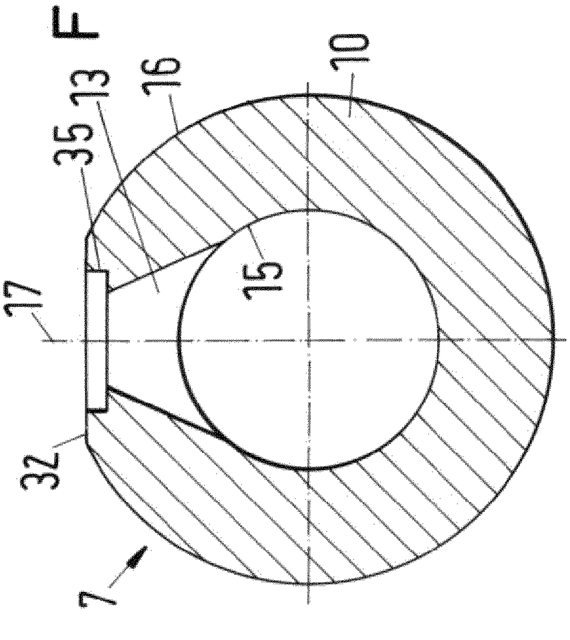


Fig.8

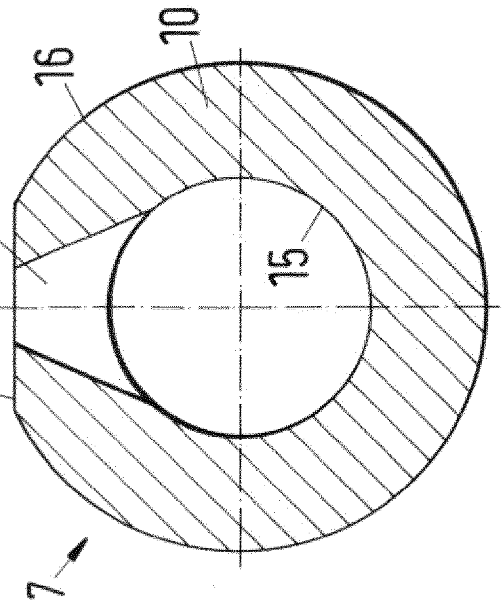


Fig.12

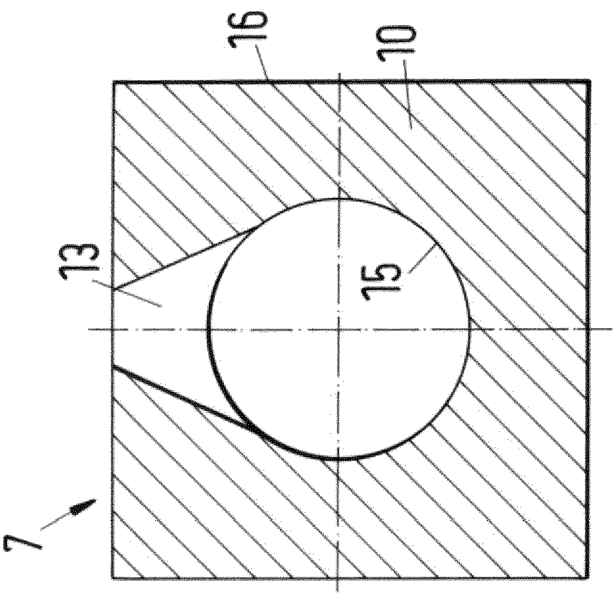
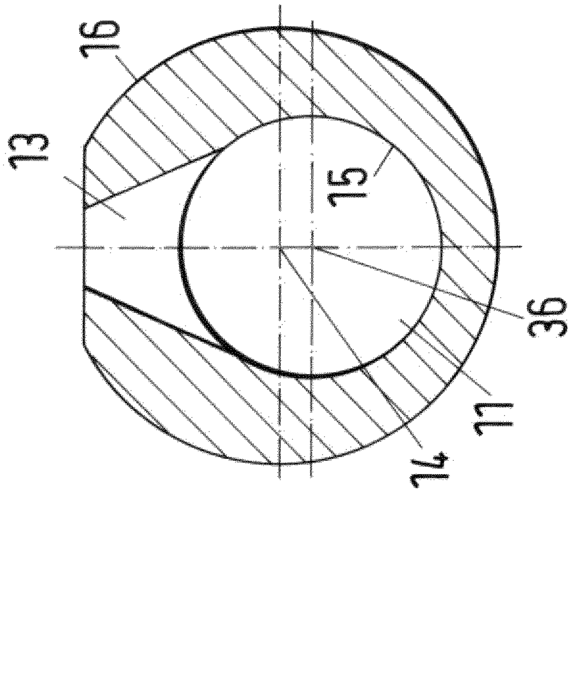


Fig.11



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EUROPEAN SEARCH REPORT

Application Number
EP 12 15 5871

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F02M

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
Munich	19 April 2012	Etschmann, Georg

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