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(54) HIGH-PRESSURE COMMON RAIL SYSTEM FOR LOW-SPEED ENGINE WITH MULTIPLE SAFETY PROTECTION FUNCTIONS

(57) The present solution provides a high-pressure common rail system for a low-speed engine with multiple safety protection functions to achieve the multiple safety protection functions of the high-pressure common rail system. The system comprises: an ECU; an electronic control high-pressure fuel pump provided with an electronic control proportional valve configured to adjust the rate of feed of a low-pressure heavy fuel oil from a fuel tank of the low-speed engine into the electronic control high-pressure fuel pump according to a first instruction from the ECU; a first distribution block connected to the electronic control high-pressure fuel pump via a first high-pressure fuel pipe; a second distribution block connected to the first distribution block via a second high-pressure fuel pipe; and a common rail pipe connected to the second distribution block via a third high-pressure fuel pipe. A sensor connected to the ECU is mounted onto the common rail pipe. A plurality of flow limiting valve components are mounted onto the common rail pipe, and each of the flow limiting valve components is connected to one of electronic control fuel injectors via one of fourth high-pressure fuel pipes. A pressure limiting valve component is further mounted onto the common rail pipe. The first distribution block is equipped with a shut-off valve component and a safety valve component. Circulation valve components are further mounted onto the common rail pipe and onto each electronic control fuel injector.

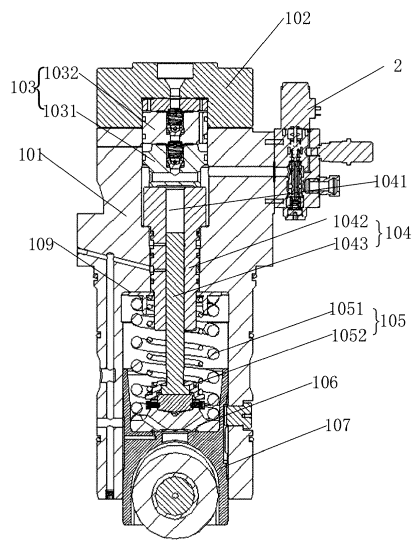

FIG.2

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Description

Technical Field

[0001] The present disclosure relates to the field of high-pressure common rail systems, and specifically to a high-pressure common rail system for a low-speed engine with multiple safety protection functions.

Background Art

[0002] With the increasingly stringent emission regulations and the increasingly severe energy crisis, high-pressure common rail systems have been used more and more in diesel fuel systems. The high-pressure common rail systems meet the increasingly stringent emission regulations and user requirements for fuel economy and emission due to their characteristics such as high injection pressure, high response speed, and flexible control. The high-pressure common rail systems are different from the traditional mechanical fuel injection systems in that the high-pressure common rail systems have always been in a high-pressure state. In this case, there are very strict requirements on the reliability and safety of the parts of the high-pressure common rail systems.

Summary

[0003] The object of the present disclosure includes providing a high-pressure common rail system for a low-speed engine with multiple safety protection functions to achieve the multiple safety protection functions for the high-pressure common rail system.

[0004] The present disclosure provides a high-pressure common rail system for a low-speed engine with multiple safety protection functions, comprising: an ECU (electronic control unit); an electronic control high-pressure fuel pump provided therein with an electronic control proportional valve, wherein the electronic control proportional valve is configured to adjust the proportion of feed of a low-pressure heavy fuel from a fuel tank of the low-speed engine into the electronic control high-pressure fuel pump according to a first instruction from the ECU; a first distribution block connected to the electronic control high-pressure fuel pump via a first high-pressure fuel pipe; a second distribution block connected to the first distribution block via a second high-pressure fuel pipe; and a common rail pipe connected to the second distribution block via a third high-pressure fuel pipe, wherein a sensor configured to detect the fuel pressure of the high-pressure heavy fuel in the common rail pipe is mounted onto the common rail pipe, and the sensor is connected to the ECU; a plurality of flow limiting valve components are mounted onto the common rail pipe, and each of the flow limiting valve components is connected to one of electronic control fuel injectors via one of fourth high-pressure fuel pipes; the flow limiting valve component is configured to be closed when a fuel pressure difference between the fourth high-pressure fuel pipe and the common rail pipe exceeds a set pressure difference; a pressure limiting valve component is further mounted onto the common rail pipe, and the pressure limiting valve component is configured to be opened when the fuel pressure in the common rail pipe exceeds a first set pressure value, so that the fuel pressure in the common rail pipe is stabilized at a target pressure value; the first distribution block is equipped with a shut-off valve component and a safety valve component, wherein the shut-off valve component is configured to perform a pressure relief process according to a second instruction from the ECU; the safety valve component is configured to be opened when the shut-off valve component and the pressure limiting valve component malfunction and the fuel pressure in the common rail pipe exceeds a second set pressure value; circulation valve components are further mounted onto the common rail pipe and the electronic control fuel injectors, and the circulation valve components are configured to be opened when the low-speed engine is stopped, so that a circulation loop is formed, respectively, between the fuel tank of the low-speed engine and the common rail pipe, and between the fuel tank of the low-speed engine and each of the electronic control fuel injectors.

[0005] Optionally, the electronic control high-pressure fuel pump comprises: a pump body, provided with a central hole along its axial direction; a pump cover, mounted onto an upper end surface of the pump body; and a fuel inlet and outlet valve component, a plunger and barrel assembly, a plunger spring, a lower spring seat component, and a guide piston component, each being assembled in the central hole of the pump body; the electronic control proportional valve is assembled on a side of the pump body; the fuel inlet and outlet valve component comprises a fuel inlet valve component and a fuel outlet valve component, wherein the fuel inlet valve component comprises a fuel inlet valve seat, a fuel inlet valve, and a fuel inlet valve spring; the fuel inlet valve is mounted in a central hole of the fuel inlet valve seat; the fuel inlet valve spring is positionally limited between the fuel inlet valve and a wall of the hole of the fuel inlet valve seat; a conical sealing is formed between the fuel inlet valve and the fuel inlet valve seat by being pressed by the fuel inlet valve spring; the fuel outlet valve component comprises a fuel outlet valve seat, a fuel outlet valve, a fuel outlet valve spring, and a fuel outlet valve spring seat, wherein the fuel outlet valve spring seat is mounted at an upper end of the fuel outlet valve seat; the fuel outlet valve is mounted in a central hole of the fuel outlet valve seat; the fuel outlet valve spring is positionally limited between the fuel outlet valve and the fuel outlet valve spring seat; a conical sealing is formed between the fuel outlet valve and the fuel outlet valve seat by being pressed by the fuel outlet valve spring; a high-pressure fuel outlet chamber is formed between the fuel outlet valve seat and the fuel inlet valve seat; a high-pressure fuel chamber is formed in the plunger and barrel assembly, and the high-pressure fuel chamber communicates with the high-pressure fuel outlet chamber via a fuel hole in the fuel inlet valve seat; the electronic control proportional valve communicates with the fuel inlet hole of the fuel inlet valve seat via a first fuel hole in the pump body, and the fuel inlet hole is communicated with or disconnected from the high-pressure fuel chamber; the electronic control proportional valve is provided with a cooling circulation fuel passage, and cooling fuel from a cooling fuel passage of the pump body is injected into the cooling circulation fuel passage and then flows back into the cooling fuel passage of the pump body.

[0006] Optionally, the plunger and barrel assembly comprises: a plunger barrel, disposed at a lower end of the fuel inlet valve seat; and a plunger, slidably inserted into a central hole of the plunger barrel, wherein the high-pressure fuel chamber is formed by the plunger barrel, the plunger, together with the fuel inlet valve seat; an inner wall of the plunger barrel is provided with a first annular groove and a second annular groove; the pump body is provided with a mixed oil outlet passage and a lubricating oil supply passage, the mixed oil outlet passage communicates with the first annular groove via a mixed oil passage in the plunger barrel, and the lubricating oil supply passage communicates with the second annular groove via a lubricating oil passage in the plunger barrel; and the first annular groove is located above the second annular groove.

[0007] Optionally, the lower spring seat component is disposed under the plunger and barrel assembly, and the lower spring seat component comprises: an outer spring seat, being in a boss-type structure thinner on the outer side and thicker in the middle as a whole, wherein the outer spring seat has an upper end surface provided with a third counterbore with a concave spherical surface; an upper sphere, mounted, at its lower part, in the third counterbore, and provided, at its lower end surface, with a convex spherical surface mating with the concave spherical surface; and an inner spring seat, covering an upper part of the upper sphere, wherein the inner spring seat has a first axial through hole penetrating its upper and lower end surfaces; a lower cylindrical head of the plunger is positionally limited in the first axial through hole, and the lower cylindrical head of the plunger has a lower end surface abutting against an upper end surface of the upper sphere.

[0008] Optionally, a spherical hole is provided in the center of the third counterbore, a third annular groove is provided in a lower end surface of the outer spring seat, and the spherical hole communicates with the third annular groove via a lubricating oil inlet conduit; the outer surface of the outer spring seat is formed as a tapered surface, which is provided with a lubricating oil outlet passage communicating with the lower end surface of the outer spring seat; the lubricating oil outlet passage is provided obliquely; a circumferential annular groove is provided in a circumferential direction of the upper sphere; a positioning pin is mounted in the circumferential annular groove through a positioning pin hole of the outer spring seat; and a spacing between upper and lower surfaces of the circumferential annular groove is greater than a cylindrical diameter of a part of the positioning pin that is located in the circumferential annular groove.

[0009] Optionally, the first axial through hole provided inside the inner spring seat comprises: a seventh hole, an eighth hole, and a ninth hole having diameters gradually increasing from top to bottom, wherein a first guide hole with a gradually increasing diameter is provided between the eighth hole and the ninth hole; a second guide hole with a gradually increasing diameter is provided on a side of the ninth hole facing the upper sphere; the walls of the first guide hole and the second guide hole are formed as tapered guide surfaces; the upper part of the upper sphere is positioned partially in the ninth hole through the second guide hole; there is a gap greater than or equal to 1 mm between the upper sphere and the ninth hole; there is a gap greater than or equal to 1 mm between the third counterbore and the upper sphere.

[0010] Optionally, the electronic control high-pressure fuel pump further comprises: an upper spring seat, sleeved on the plunger barrel and located at an upper end of the inner spring seat; and the plunger spring comprises: a first plunger spring, compressively mounted between the upper spring seat and the outer spring seat; and a second plunger spring, compressively mounted between the upper spring seat and the inner spring seat.

[0011] Optionally, the diameters of the concave spherical surface in the outer spring seat and the convex spherical surface of the upper sphere are each 20 to 100 times the diameter of the plunger.

[0012] Optionally, the guide piston component comprises: a guide piston, provided with a first mounting hole at a central position of its upper end surface and provided with a second mounting hole at its lower end surface, wherein the first mounting hole and the second mounting hole communicate with each other via a communicating hole, the lower spring seat component is mounted in the first mounting hole; a roller component, including a roller mounted in the second mounting hole, a roller bushing interference-fitted in the roller, and thrust bearings interference-fitted at both ends of the roller in its axial direction, wherein an annular groove is provided in the axial direction of the roller, and an arc-shaped transitional connection is formed between the bottom of the annular groove and an end surface of the roller in the axial direction; and a roller pin, fitted in the roller bushing with a clearance; a boss is disposed protruding from a wall of the second mounting hole, and the boss is in contact with the thrust bearing; a plurality of first radial oil grooves are uniformly arranged in the boss along its radial direction, and the first radial oil grooves are provided with respect to the thrust bearing.

[0013] Optionally, the outer surface of the roller pin is provided as a cylindrical surface, wherein the cylindrical surface is provided with first waist-shaped grooves and second waist-shaped grooves at its two positions, respectively, and the first waist-shaped grooves and the second waist-shaped grooves are provided in the middle positions of the roller pin, wherein the second waist-shaped groove is provided in the first waist-shaped groove and recessed inward relative to the first waist-shaped groove, and a stepped shape is formed by them together; a small-angle wedge-shaped groove with an angle between 5° and 20° is formed between the first waist-shaped groove and the outer surface of the roller bushing, and a second oil hole is provided in the second waist-shaped groove; the two second oil holes at the two positions communicate with each other via a lubricating oil outlet passage, and the two second oil holes are provided at an angle of 70° to 120°.

[0014] Optionally, the outer surface of the guide piston is provided as a cylindrical surface, wherein the cylindrical surface is provided with a partial circumferential oil groove and a circumferential oil groove, a first axial oil groove, and a vertical groove, wherein the vertical groove is provided in the circumferential oil groove, and the vertical groove communicates with the partial circumferential oil groove via the first axial oil groove; the cylindrical surface is further provided with an inclined hole having two ends communicating with the circumferential oil groove and an inner wall of the second mounting hole, respectively; the cylindrical surface is further provided with a second axial oil groove communicating with the circumferential oil groove; the cylindrical surface is further provided with a first straight hole and a second straight hole connected to each other, wherein the first straight hole communicates with the first axial oil groove, and the second straight hole communicates with the first mounting hole; a lubricating oil inlet passage is provided in the outer cylindrical surface of the roller pin, the lubricating oil inlet passage is provided with respect to the inclined hole, and the lubricating oil inlet passage communicates with the lubricating oil outlet passage.

[0015] Optionally, the outer cylindrical surface of the roller pin is provided with a DLC (diamond-like carbon) coating; the roller bushing is made of a copper alloy; the thrust bearing is made of a copper alloy; forced lubrication and dynamic pressure lubrication are used between the roller pin and the roller bushing; and forced lubrication and dynamic pressure lubrication are used between the thrust bearing and the boss.

[0016] Optionally, the common rail pipe has a fuel inlet conduit and a fuel return conduit penetrating both ends thereof; a fuel inlet end cap is fixed at one end of the common rail pipe, and the fuel inlet end cap is provided with a fuel inlet port communicating with the fuel inlet conduit; an end cover is fixed at the other end of the common rail pipe, the end cover is provided with a fuel outlet port communicating with the fuel inlet conduit, and the circulation valve component is fixed to the end cover; the pressure limiting valve component and the plurality of flow limiting valve components communicate with the fuel inlet conduit, respectively, and the pressure limiting valve component and the plurality of flow limiting valve components communicate with the fuel return conduit, respectively.

[0017] Optionally, the circulation valve component comprises: a first valve body fixed to the end cover, which has a lower end surface provided with a first central hole communicating with the fuel outlet port, and an upper end surface provided with a second central hole, wherein the first central hole communicates with the second central hole; a first valve core, which is slidably inserted into the first central hole from the lower end surface of the first valve body and partially located in the second central hole; a lower spring seat, which is sleeved on the part of the first valve core that is located in the second central hole, and which is fixedly connected to the first valve core, wherein a first cavity is formed between the lower spring seat and the bottom of the second central hole; a gland, which is fixed to the upper end surface of the first valve body, wherein the gland has an upper end surface provided with a threaded hole; a fuel return joint, which is partially fixed in the threaded hole; and a first pressure regulating spring, which is positionally limited between the lower spring seat and the gland; the first valve body is provided with a first fuel return passage communicating with the first central hole, the gland is provided with a second fuel return passage communicating with the first fuel return passage, the fuel return joint is provided with a third fuel return passage communicating with the second fuel return passage, and a fuel circulation passage is formed by the first fuel return passage, the second fuel return passage, and the third fuel return passage; the first valve body is provided with a first air inlet passage communicating with the first cavity, and the gland is provided with a second air inlet passage communicating with the first air inlet passage and an air inlet port communicating with the second air inlet passage; when a spring force from the first pressure regulating spring is less than or equal to a pressure sum of the pressure of a gas introduced into the first cavity and the pressure of fuel fed into the fuel inlet end of the first central hole, a conical seal is formed between the first valve core and the first central hole, and the conical seal is formed at a position below a position where the first fuel return passage is connected to the first central hole.

[0018] Optionally, a first conical seal surface and an external thread are provided at the top of the part of the first valve core that is located in the second central hole, and the external thread is located at an upper end of the first conical seal surface; the lower spring seat passes through the external thread and then is sleeved on the first conical seal surface, and the lower spring seat has a second conical seal surface that forms a conical seal with the first conical seal surface; the lower spring seat is tightly pressed by a nut sleeved on the periphery of the external thread; and the first pressure regulating spring is sleeved on the nut and fixed to the lower spring seat.

[0019] Optionally, the fuel return joint is screwed in the threaded hole; the upper end surface of the gland is provided with a first flat sealing surface at the opening of the threaded hole; and the fuel return joint is provided with a second flat sealing surface that forms a planar seal with the first flat sealing surface.

[0020] Optionally, a third cavity is formed between the lower end surface of the fuel return joint and the bottom of the threaded hole, the third cavity communicates with the second fuel return passage and the third fuel return passage, respectively, and the maximum flow area of the third cavity is greater than the maximum flow area of the second fuel return passage.

[0021] Optionally, when the first valve core moves upward to a top dead center position in the first valve body, a distance H2 between the lower end surfaces of the first valve body and the first valve core is smaller than a distance H1 between the lower end surface of the lower spring seat and the bottom of the second central hole.

[0022] Optionally, the lower spring seat comprises a third central hole configured to limit the position of the first pressure regulating spring and a fourth central hole configured to mate with the first valve core, wherein the fourth central hole has a diameter larger than the diameter of the external thread; and the lower spring seat has an outer diameter the same as the diameter of the second central hole.

[0023] Optionally, the flow limiting valve component comprises a valve seat connected to the high-pressure common rail pipe, a second valve body, a second valve core, and a second pressure regulating spring, wherein the valve seat has a first fuel inlet hole communicating with the high-pressure common rail pipe;

the second valve body has a second axial through hole penetrating its upper and lower end surfaces, and the valve seat is partially press-fitted into the second axial through hole from the lower end surface of the second valve body; the second valve core is mounted in the second axial through hole and disposed above the valve seat; the second valve core has an axial blind hole communicating with the first fuel inlet hole, and a cavity is formed between an upper end of the second valve core and the second axial through hole; the second valve core is provided therein with a transverse flow restricting orifice communicating the axial blind hole with the cavity; the second pressure regulating spring is sleeved on the second valve core and is positionally limited in the cavity; the head of the upper end of the second valve core has a third conical seal surface and a fourth conical seal surface connected to each other; and a first sealing seat surface that can form a conical seal with the third conical seal surface is formed at a wall of the second axial through hole, and a gap may be formed between the first sealing seat surface and the fourth conical seal surface.

[0024] Optionally, the second axial through hole comprises a first hole, a second hole, a third hole, a fourth hole, a fifth hole, and a sixth hole, connected to one another in sequence from top to bottom; the valve seat is partially press-fitted in the first hole; the second valve core is assembled in the second hole, and a cavity is formed between the upper part of the second valve core and the upper part of the second hole; the first sealing seat surface is formed at the wall of the third hole; a second sealing seat surface sealed to the fuel inlet end of the fuel pipe of the fuel injector is formed at the wall of the fifth hole; and the diameter of the sixth hole is larger than the diameter of each of the first hole, the second hole, the third hole, the fourth hole, and the fifth hole.

[0025] Optionally, the second valve body is further provided with a fourth fuel return passage having one end communicating with the sixth hole, and the other end communicating to the lower end surface of the second valve body; the maximum sectional flow area of the fourth fuel return passage is smaller than the maximum sectional flow area of the gap formed between the fourth conical seal surface and the first sealing seat surface; and the maximum sectional flow area of the fourth fuel return passage is smaller than both the maximum sectional flow area of the second axial through hole and the maximum sectional flow area of the fourth hole.

[0026] Optionally, a fifth conical seal surface forming a conical seal with the high-pressure common rail pipe is formed at the head of the lower end of the valve seat; and a first flat portion communicating with the fourth fuel return passage is formed by milling the larger outer cylindrical portion of the lower end of the valve seat.

[0027] Optionally, the pressure limiting valve component comprises a third valve body, a third valve core, a third pressure regulating spring, and a fuel pipe joint, wherein the third valve body is provided therein with a first-stage hole, a second-stage hole, a third-stage hole, and a fourth-stage hole communicating in sequence from bottom to top; the third valve core has a head slidably inserted from the fourth-stage hole through the third-stage hole and partially into the second-stage hole, and a conical seal can be formed between the third valve core and the second-stage hole; a first gap allowing the fuel to pass therethrough is formed between the third valve core and the third-stage hole, and a second gap allowing the fuel to pass therethrough is formed between the third valve core and the fourth-stage hole; the fuel pipe joint is fixed at an upper end of the third valve body, the fuel pipe joint is provided therein with a first counterbore, a second counterbore, and a fuel outlet hole communicating in sequence from bottom to top, and the third pressure regulating spring is positionally limited between the third valve core and the second counterbore.

[0028] Optionally, the diameter of the first-stage hole is larger than the diameter of the second-stage hole, and the diameter of the third-stage hole is larger than the diameter of each of the second-stage hole and the fourth-stage hole.

[0029] Optionally, the head of the third valve core is provided with a first tapered portion and a second tapered portion connected to each other, the first tapered portion has an angle of 120°, and the second tapered portion has an angle of 60°; second flat portions symmetrically arranged are formed by milling the outer cylindrical portion of the middle part of the third valve core, a second gap allowing the fuel to passthrough is formed between the second flat portion and the wall of the fourth-stage hole of the third valve body, and the gap has a maximum fuel flowable area larger than the maximum fuel flowable area in the second-stage hole; a plurality of recesses arranged spaced apart from one another are provided in the outer cylindrical portion of the middle-upper part of the third valve core, and a third gap allowing the fuel to passthrough is formed between each of the recesses and the first counterbore; and the third pressure regulating spring is sleeved on the second smaller outer cylindrical portion of the upper part of the third valve core.

[0030] Optionally, a sealing seat surface forming the conical seal with the second tapered portion is formed at a connection between the second-stage hole and the third-stage hole, and there is an angular deviation less than 1° between the sealing seat surface and the second tapered portion; and the fourth-stage hole of the third valve body has a diameter φ of 5 mm.

[0031] Optionally, a limited distance h1 within which the third valve core is movable is provided between the third valve core and the bottom of the first counterbore of the fuel pipe joint; an overlap region h is provided between the third valve core and the third valve body.

[0032] The present disclosure has the advantageous effect that multiple valves such as a circulation valve component, a pressure limiting valve component, a shut-off valve component, and a safety valve component are involved in the common rail system, and the various types of valves are opened under specific working conditions, so as to achieve multiple safety protection functions of the common rail system.

Brief Description of Drawings

[0033]

FIG. 1 is a schematic structural view of a high-pressure common rail system for a low-speed engine according to an embodiment of the present disclosure;

FIG. 2 is a schematic structural view of an electronic control high-pressure fuel pump in a high-pressure common rail system for a low-speed engine according to an embodiment of the present disclosure;

FIG. 3 is a schematic structural view of a fuel inlet and outlet valve component according to an embodiment of the present disclosure;

FIG. 4 is a schematic structural view of a plunger and barrel assembly in the prior art;

FIG. 5 is a schematic view showing fitting among a plunger and barrel assembly and a pump body, a fuel inlet valve seat, and an upper spring seat according to an embodiment of the present disclosure;

FIG. 6 is a schematic structural view of an angle formed between a plunger and an upper sphere due to uneven forces according to an embodiment of the present disclosure;

FIG. 7 is a schematic structural view of the plunger fitted to the upper sphere adjusted with a spherical surface according to an embodiment of the present disclosure;

FIG. 8 is a schematic structural view of a lower spring seat component fitted with a plunger according to an embodiment of the present disclosure;

FIG. 9 is a schematic sectional view of a lower spring seat component according to an embodiment of the present disclosure;

FIG. 10 is a schematic sectional view of a lower spring seat component according to an embodiment of the present disclosure;

FIG. 11 is a schematic structural view of an inner spring seat according to an embodiment of the present disclosure;

FIG. 12 is a schematic structural view of an upper sphere according to an embodiment of the present disclosure;

FIG. 13 is a schematic sectional view of an outer spring seat according to an embodiment of the present disclosure;

FIG. 14 is a schematic sectional view of an outer spring seat according to an embodiment of the present disclosure;

FIG. 15 is a schematic sectional view of a guide piston component according to an embodiment of the present disclosure;

FIG. 16 is a schematic sectional view of a guide piston according to an embodiment of the present disclosure;

FIG. 17 is a schematic structural view of a guide piston according to an embodiment of the present disclosure;

FIG. 18 is a schematic sectional view of a guide piston according to an embodiment of the present disclosure;

FIG. 19 is a schematic structural view of a roller pin according to an embodiment of the present disclosure;

FIG. 20 is a schematic axially sectional view of a roller pin according to an embodiment of the present disclosure;

FIG. 21 is a schematic radially sectional view of a roller pin according to an embodiment of the present disclosure;

FIG. 22 is a schematic view showing a force distribution on a roller component not provided with an annular groove according to an embodiment of the present disclosure;

FIG. 23 is a schematic view showing a force distribution on a roller component provided with an annular groove according to an embodiment of the present disclosure;

FIG. 24 is a schematic structural view of a common rail pipe and some components thereon according to the present disclosure;

FIG. 25 is a schematic structural view of a common rail pipe and some components thereon according to the present disclosure;

FIG. 26 is a schematic structural view of a common rail pipe and some components thereon according to the present disclosure;

FIG. 27 is a schematic structural view of a flow limiting valve according to an embodiment of the present disclosure;

FIG. 28 is a schematic structural view of a flow limiting valve according to an embodiment of the present disclosure;

FIG. 29 is a schematic structural view of a flow limiting valve according to an embodiment of the present disclosure;

FIG. 30 is a schematic structural view of a pressure limiting valve according to an embodiment of the present disclosure;

FIG. 31 is a schematic structural view of a pressure limiting valve according to an embodiment of the present disclosure;

FIG. 32 is a schematic structural view of a circulation valve according to an embodiment of the present disclosure;

FIG. 33 is a schematic structural view of a second axial through hole according to an embodiment of the present disclosure;

FIG. 34 is a schematic structural view of a circulation valve according to an embodiment of the present disclosure;

FIG. 35A is a front view of a guide piston according to an embodiment of the present disclosure; and FIG. 35B is a left side view of a guide piston according to an embodiment of the present disclosure.

[0034] Description of Reference Signs: 1-electrically controlled high-pressure fuel pump; 101-pump body; 1012-lubricating oil supply passage; 102-pump cover: 103-fuel inlet and outlet valve component; 1031-fuel inlet valve component; 10311-fuel inlet valve seat; 10312-fuel inlet valve; 10313-fuel inlet valve spring; 1032-fuel outlet valve component; 10321-fuel outlet valve seat; 10322-fuel outlet valve; 10323-fuel outlet valve spring; 10324-fuel outlet valve spring seat; 1033-high-pressure fuel outlet chamber; 104-plunger and barrel assembly; 1041-high-pressure fuel chamber; 1042-plunger barrel; 10421-first annular groove; 10422-second annular groove; 10423-mixed oil passage; 10424-lubricating oil passage; 1043-plunger; 10431-lower cylindrical head; 105-plunger spring; 1051-first plunger spring; 1052-second plunger spring; 106-lower spring seat component; 1061-outer spring seat; 10611-third counterbore; 10612-spherical hole; 10613-third annular groove; 10614-lubricating oil inlet conduit: 10615-lubricating oil outlet passage; 10616-positioning pin hole; 1062-upper sphere; 10621-circumferential annular groove: 1063-inner spring seat; 10631-first axial through hole; 10632-seventh hole; 10633-eighth hole; 10634-ninth hole; 10635-first guide hole; 10636-second guide hole; 10637-tapered guide surface; 10638-relief groove; 10639-annular weight-reducing groove; 1064-positioning pin; 107-guide piston component; 1071-guide piston; 10711-first mounting hole; 10726-second chamfer; 10712-second mounting hole; 10713-communicating hole; 10714-boss; 10715-first radial oil groove; 10727-tenth hole; 10713-communicating hole; 10716-first axial oil groove; 10717-vertical groove; 10718-inclined hole; 10719-second axial oil groove; 10720-first straight hole; 10721-second straight hole; 10725-circumferential oil groove; 10729-partial circumferential oil groove; 10741-first chamfer; 1072-roller component; 10728-roller; 10724-annular groove; 10722-roller bushing; 10723-thrust bearing; 1073-roller pin; 10731-first waist-shaped groove; 10732-second waist-shaped groove; 10733-second oil hole; 10735-lubricating oil inlet passage; 10739-third radial oil passage; 10740-axial oil passage; 10736-eleventh hole; 10737-return spring; 10738-stop pin; 2-electrically controlled proportional valve; 109-upper spring seat; 3-first high-pressure fuel pipe; 5-shut-off valve component; 6-first distribution block; 7-safety valve component; 8-second high-pressure fuel pipe; 9-second distribution block; 10-third high-pressure fuel pipe; 14-fourth high-pressure fuel pipe; 15-electronic control fuel injector; 12-common rail pipe; 1201, 505-fuel inlet conduit; 1202-fuel return conduit; 1203-first notch; 1204-second notch; 1205-fuel return hole; 121-fuel inlet end cap; 122-end cover; 123-pressure limiting valve mounting seat; 124-sensor; 125-flow limiting valve mounting seat; 126-bracket; 127-sensor mounting seat; 128-first bolt; 129-screw; 131-third waist-shaped hole; 16-circulation valve component; 161-first valve body; 1601-first central hole; 1602-second central hole; 1605-first fuel return passage; 163-third sealing ring groove; 1619-guide portion; 1620-first sealing ring groove; 1622-second sealing ring groove; 1608-first air inlet passage; 1623-air outlet passage; 162-first valve core; 1611-first conical seal surface; 1612-external thread; 1624-third sealing ring; 164-lower spring seat; 1617-third central hole; 1618-fourth central hole; 1613-second conical seal surface; 1625-fourth sealing ring groove; 1626-mating portion; 1627-fourth sealing ring; 1628-first sealing ring; 167-gland; 1606-second fuel return passage; 1603-threaded hole; 1629-second air inlet passage; 1630-air inlet port; 1615-first flat sealing surface; 168-fuel return joint; 1616-second flat sealing surface; 1607-third fuel return passage; 169-first pressure regulating spring; 1614-nut; 1631-second sealing ring; 1632-screw; 18-pressure limiting valve component; 181-third valve body; 182-third valve core; 183-second O-ring seal; 184-second bolt; 185-third pressure regulating spring; 186-fuel pipe joint; 18604-fifth sealing ring groove; 187-pressure regulating gasket; 18101-first-stage hole; 18102-second-stage hole; 18103-sealing seat surface; 18104-third-stage hole; 18105-fourth-stage hole; 18601-first counterbore; 18602-second counterbore; 18603-fuel outlet hole; 18201-first tapered portion; 18202-second tapered portion; 18203-second flat portion; 18204-third outer cylindrical portion; 18205-recess; 18206-second smaller outer cylindrical portion; h-overlap region; h1-limited distance; 13-flow limiting valve component; 131-valve seat; 132-second valve body; 133-second valve core; 136-first O-ring seal; 135-second pressure regulating spring; 13101-first fuel inlet hole; 13102-fifth conical seal surface; 13103-sunk groove; 13104-first flat portion; 13105-larger end surface; 13106-first smaller outer cylindrical portion; 13201-smaller end surface; 13202-large bevel; 13203-first mating portion; 13204-second mating portion; 13205-third hole; 13206-first sealing seat surface; 13207-fourth hole; 13208-second sealing seat surface; 13209-second axial through hole; 13210-fourth fuel return passage; 13211-screw mounting hole; 13212-first hole; 13213-second hole; 13214-fifth hole; 13215-sixth hole; 13301-first outer cylindrical portion; 13302-axial blind hole; 13303-second outer cylindrical portion; 13304-transverse flow restricting orifice; 13305-third conical seal surface; 13306-fourth conical seal surface.

Detailed Description of Embodiments

[0035] Exemplary embodiments of the present disclosure will be described below in more detail with reference to the accompanying drawings. Although the exemplary embodiments of the present disclosure are shown in the accompanying drawings, it should be understood that the present disclosure may be implemented in various forms and is not intended to be limited by the embodiments set forth herein. Rather, these embodiments are provided to enable a more thorough understanding of the present disclosure and to fully convey the scope of the present disclosure to those skilled in the art.

[0036] It should be noted that when an element is referred to as being "fixed to" another element, it may be directly on the other element or an intervening element may exist therebetween. When an element is considered to be "connected" to another element, it may be connected directly to the other element or an intervening element may be present therebetween. In contrast, when an element is referred to as being "directly on" another element, no intervening element exists therebetween. The terms "vertical", "horizontal", "left", "right", and the like used herein are intended for illustrative purposes only.

[0037] In the present disclosure, terms such as "mounted", "coupled", "connected", and "fixed" should be understood broadly unless otherwise expressly specified or defined. For example, connection may be fixed connection or detachable connection or integral connection, may be mechanical connection or electric connection, or may be direct coupling or indirect coupling via an intermediate medium, or internal communication between two elements or mutual interaction between two elements. The specific meanings of the above-mentioned terms in the present disclosure can be understood by those of ordinary skill in the art according to specific situations.

[0038] In addition, the terms "first" and "second" are used for descriptive purposes only, and should not be understood as an indication or implication of relative importance or an implicit indication of the number of the indicated technical features. Therefore, a feature defined with the terms "first" and "second" may explicitly or implicitly include one or more such features. In the description of the present disclosure, "a plurality" means two or more, unless otherwise expressly and specifically defined.

[0039] Referring to FIG. 1, the present disclosure provides a high-pressure common rail system for a low-speed engine with multiple safety protection functions, comprising: an electronic control unit (ECU); an electronic control high-pressure fuel pump 1, provided therein with an electronic control proportional valve 2, which is configured to adjust the proportion of feed of a low-pressure heavy fuel from a fuel tank of the low-speed engine into the electronic control high-pressure fuel pump 1 according to a first instruction from the ECU, wherein the electronic control high-pressure fuel pump 1 employs a single plunger structure, and the system is provided with three electronic control high-pressure fuel pumps 1, which serve as backups for one another, to prevent the failure of the system caused by a fault in the high-pressure fuel pump; a first distribution block 6 connected to the electronic control high-pressure fuel pumps 1 via corresponding first high-pressure fuel pipes 3; a second distribution block 9 connected to the first distribution block 6 via a second high-pressure fuel pipe 8; and a common rail pipe 12 connected to the second distribution block 9 via a third high-pressure fuel pipe 10. A sensor 17 configured to detect the fuel pressure of the high-pressure heavy fuel in the common rail pipe 12 is mounted onto the common rail pipe 12, and the sensor 17 is electrically connected to the ECU. A plurality of flow limiting valve components 13 are mounted onto the common rail pipe 12, and each of the flow limiting valve components 13 is connected to one of electronic control fuel injectors 15 via one of fourth high-pressure fuel pipes 14. The flow limiting valve component 13 is configured to be closed when a fuel pressure difference between the fourth high-pressure fuel pipe 14 and the common rail pipe 12 exceeds a set pressure difference. The flow limiting valve component 13 is mounted onto the common rail pipe 12 and has an outlet connected to an electronic control fuel injector 15 via a corresponding fourth high-pressure fuel pipe 14. Each cylinder of the low-speed engine has multiple fuel injectors, and each of the fuel injectors is provided with an independent flow limiting valve component 13. When the fourth high-pressure fuel pipe 14 is broken or fuel is abnormally injected from the electronic control fuel injector 15, the flow limiting valve component 13 cuts off the introduction of the high-pressure fuel to the electronic control fuel injector 15, thereby achieving the effect of protecting the low-speed engine and improving its safety. The fourth high-pressure fuel pipe 14 is designed with double walls. High-pressure abnormal leakages from all the double-layered fourth high-pressure fuel pipes 14 communicate with high-pressure abnormal leakages from the components connected thereto such that a fuel return system for the high-pressure abnormal leakage is formed.

[0040] In the field of diesel engines, diesel engines may generally be divided into medium- and high-speed engines and low-speed engines according to their working rotational speed ranges. The low-speed diesel engines are often used for directly driving propellers. Diesel engines are required to have lower rotational speeds in order to allow the propellers to have high propulsion efficiency. Therefore, the low-speed engines are often used as power for various types of ships. Furthermore, heavy fuel is widely used as fuel in the low-speed diesel engines. In the field of marine engines, the engines are generally divided, according to the rotational speeds of the engines, into high-speed engines with a rotational speed range above 3,000 rpm, medium-speed engines at 1,500 to 3,000 rpm, and low-speed engines below 1,500 rpm.

[0041] A pressure limiting valve component 18 is further mounted onto the common rail pipe 12. The pressure limiting valve component 18 is configured to be opened when the fuel pressure in the common rail pipe 12 exceeds a first set pressure value, so that the fuel pressure in the common rail pipe 12 is stabilized to a target pressure value. The pressure limiting valve component 18 is mounted onto the common rail pipe 12, and the pressure limiting valve component 18 employs a mechanical structure and has the ability to maintain an operation under fault pressure after it is opened at certain pressure. In an optional embodiment, when the pressure in the system exceeds 125 MPa, the pressure limiting valve component 18 is automatically opened and changed into a fault mode, so as to keep the system operating safely at the pressure required in the fault mode.

[0042] The first distribution block 6 is equipped with a shut-off valve component 5 and a safety valve component 7. The shut-off valve component 5 is configured to perform a process of relieving pressure from the first high-pressure fuel pipe 3 according to a second instruction from the ECU. The safety valve component 7 is configured to be opened when the shut-off valve component 5 and the pressure limiting valve component 18 malfunction and the fuel pressure in the common rail pipe 12 exceeds a second set pressure value. The shut-off valve component 5 employs an air-controlled structure and is mounted onto the first distribution block 6 to facilitate rapid relief of pressure from the system. When the low-speed engine or the fuel system needs to be stopped emergently in an emergency, the shut-off valve component 5 is quickly opened as a secondary pressure protection measure to quickly relieve pressure from the system. The shut-off valve component 5 employs an independent fuel return structure, which is separated from the fuel return system for the high-pressure abnormal leakage. The safety valve component 7 employs a mechanical structure and is mounted onto the first distribution block 6. It is opened at a pressure of preferably 150 MPa, and in general, it is opened at a pressure ranging from 150 MPa to 155 MPa. When the system is out of control and both the shut-off valve component 5 and the pressure limiting valve component 18 malfunction, the safety valve component 7 is opened as the secondary protector to ensure the safety of the entire system.

[0043] Circulation valve components 16 are further mounted onto the common rail pipe 12 and the electronic control fuel injectors 15. The circulation valve components 16 are configured to be opened when the low-speed engine is stopped, so that a circulation loop is formed between the fuel tank of the low-speed engine and the common rail pipe 12, and between the fuel tank of the low-speed engine and each of the electronic control fuel injectors 15, respectively. The circulation valve components 16 are mounted onto the end cover 122 and onto the respective electronic control fuel injectors 15, respectively. When the low-speed engine is started, there is no fuel in the common rail pipe 12 and in the respective electronic control fuel injectors 15. At this time, the valve core in each circulation valve component 16 is at the bottom dead center position, and the circulation valve component 16 is in an open circulation state. In order to allow the common rail system to quickly build up pressure and work, compressed air is introduced into the circulation valve component 16, so that the circulation valve component 16 quickly closes the system and starts to build up pressure. When the pressure exceeds a preload force on the spring of the circulation valve component 16, the circulation valve is closed mainly by fuel pressure. When the low-speed engine is stopped, the system pressure is reduced by the shut-off valve component 5, and then the circulation valve component 16 automatically opens to achieve the low-pressure automatic circulation function and ensure the safety of the heavy fuel in the system. After the low-speed engine is stopped to relieve pressure, the circulation valve component 16 is opened. Considering that the heavy fuel in the system which is cooled and solidified may corrode the parts of the system and hence reduce the service life of the parts, the system is brought into a low-pressure circulation mode at this time. Low-pressure fuel enters the common rail pipe 12 through the first high-pressure fuel pipe 3, the second high-pressure fuel pipe 8, and the third high-pressure fuel pipe 10, and then the fuel is divided into two branches. One of the branches flows to the fuel tank from the circulation valve component 16 mounted onto the end cover 122. The other branch enters the electronic control fuel injector 15 through the flow limiting valve component 13 and flows out to the fuel tank through the circulation valve component 16 mounted to the electronic control fuel injector 15. The flow area provided in the circulation valve component 16 on the end cover 122 must be smaller than the flow area of the circulation valve component 16 on the electronic control fuel injector 15. If the flow area of the circulation valve component 16 on the end cover does not match the flow area of the circulation valve component on the electronic control fuel injector 15, the low-pressure fuel will flow out from the circulation valve component 16 at the end cover 122, while no fuel flows out from the circulation valve component 16 of the electronic control fuel injector 15 and the fuel circulation therethrough cannot be achieved. Therefore, it is important to match the sizes of the flow areas of the two circulation valve components 16. The flow area of each of the circulation valve components 16 is determined by the lift of the valve core.

[0044] In the fuel return system for the high-pressure abnormal leakage of the entire common rail system, the electronic control fuel injector 15, the fourth high-pressure fuel pipe 14, the flow limiting valve component 13, the common rail pipe 12, the fuel inlet end cap 121, the third high-pressure fuel pipe 10, the second distribution block 9, the second high-pressure fuel pipe 8, the first distribution block 6, the first high-pressure fuel pipe 3, and the electronic control high-pressure fuel pump 1 communicate with one another, and the fuel is returned from the leakage outlet of the most-downstream electronic control high-pressure fuel pump 1.

[0045] Referring to FIGS. 2 and 3, the electronic control high-pressure fuel pump 1 is a marine single-piece high-pressure fuel pump. The fuel pump specifically comprises: a pump body 101, provided with a central hole along its axial direction; a pump cover 102, mounted to an upper end surface of the pump body 101; and a fuel inlet and outlet valve component 103, a plunger and barrel assembly 104, a plunger spring 105, a lower spring seat component 106, and a guide piston component 107, each of which is assembled in the central hole of the pump body 101. The electronic control proportional valve 2 is assembled on a side of the pump body 101. The fuel inlet and outlet valve component 103 comprises a fuel inlet valve component 1031 and a fuel outlet valve component 1032. The fuel inlet valve component 1031 comprises a fuel inlet valve seat 10311, a fuel inlet valve 10312, and a fuel inlet valve spring 10313. The fuel inlet valve 10312 is mounted in a central hole of the fuel inlet valve seat 10311. The fuel inlet valve spring 10313 is positionally limited between the fuel inlet valve 10312 and a wall of the hole of the fuel inlet valve seat 10311. A conical seal is formed between the fuel inlet valve 10312 and the fuel inlet valve seat 10311 by being pressed against each other by the fuel inlet valve spring 10313. The fuel outlet valve component 1032 comprises a fuel outlet valve seat 10321, a fuel outlet valve 10322, a fuel outlet valve spring 10323, and a fuel outlet valve spring seat 10324. The fuel outlet valve spring seat 10324 is mounted at an upper end of the fuel outlet valve seat 10321. The fuel outlet valve 10322 is mounted in a central hole of the fuel outlet valve seat 10321. The fuel outlet valve spring 10323 is positionally limited between the fuel outlet valve 10322 and the fuel outlet valve spring seat 10324. A conical seal is formed between the fuel outlet valve 10322 and the fuel outlet valve seat 10321 by being pressed against each other by the fuel outlet valve spring 10323. A high-pressure fuel outlet chamber 1033 is formed between the fuel outlet valve seat 10321 and the fuel inlet valve seat 10311. A high-pressure fuel chamber 1041 is formed in the plunger and barrel assembly 104, and the high-pressure fuel chamber 1041 communicates with the high-pressure fuel outlet chamber 1033 via a fuel hole in the fuel inlet valve seat 10311. The electronic control proportional valve 2 communicates with the fuel inlet hole of the fuel inlet valve seat 10311 via a first fuel hole in the pump body 101, and the fuel inlet hole is communicated with or disconnected from the high-pressure fuel chamber 1041. The electronic control proportional valve 2 is provided with a cooling circulation fuel passage, and cooling fuel from a cooling fuel passage of the pump body 101 is injected into the cooling circulation fuel passage and then flows back into the cooling fuel passage of the pump body 101.

[0046] As shown in FIGS. 2 and 3, the central hole provided in the pump body 101 is a through hole penetrating the upper and lower end surfaces of the pump body 101. The pump cover 102 is fixed to the upper end surface of the pump body 101. A mounting hole corresponding to the central hole of the pump body 101 is provided in the direction of the pump cover 102 facing the pump body 101. The fuel outlet valve seat 10321 is mounted in the central hole of the pump body 101 and in the mounting hole of the pump body 101. It can be seen from FIG. 2 that the fuel outlet valve component 1032 is mounted above the fuel inlet valve component 1031, and the upper part of the pump cover 102 has a fuel passage communicating with the fuel outlet valve component 1032. Finally, the high-pressure heavy fuel pumped from the high-pressure fuel pump of the present disclosure is discharged through the fuel passage in the pump cover 102.

[0047] In the description of the present disclosure, it should be noted that orientation or positional relationships indicated by the terms such as "up", "down", "left", "right", "vertical", "horizontal", "inside", and "outside" in the specification of the present disclosure are the orientation or positional relationships shown based on the drawings, and these terms are intended only to facilitate the description of the present disclosure and simplify the description, but not intended to indicate or imply that the referred devices or elements must be in a particular orientation, or constructed or operated in the particular orientation, and therefore should not be construed as limiting the present disclosure.

[0048] The electronic control proportional valve 2, which is a hydraulic control device, has the effect of throttling a flow of fuel to be fed. The electronic control proportional valve 2 is mainly used for regulating feed of a light fuel (such as gasoline or light diesel) or the like. In the prior art, there is no scheme to use the electronic control proportional valve 2 in the regulation of feed of a heavy fuel, because the heavy fuel may reach a temperature up to 160 ºC during working, which has exceeded the limit working temperatures of electronically controlled elements such as armatures and coils of the prior electronic control proportional valves 2. In the prior art, a traditional mechanical design is employed for throttling adjustment for the feed of a heavy fuel to the high-pressure fuel pump. In other words, the fuel quantity is controlled by a governor and a spiral groove in the upper part of the plunger. Such fuel feed adjustment method is disadvantageous, for example, in that the fuel quantity is adjusted with low accuracy and is adjusted and controlled in slow response, and the magnitude of the fuel quantity is dependent on the rotational speed of the governor.

[0049] In an embodiment of the present disclosure, the problems of insufficient flexibility and high temperature in the prior mechanical adjustment method can be solved by adjusting feed of the heavy fuel using the electronic control proportional valve 2. In view of this, a cooling circulating fuel passage is provided inside the electronic control proportional valve 2, through which the cooling fuel flowing in the pump body 101 enters the electronic control proportional valve 2, so as to cool the electronically controlled elements in the electronic control proportional valve 2 in a targeted manner, so that the electronically controlled elements of the electronic control proportional valve 2 are maintained within the normal temperature range. The cooling circulation fuel passage designed in the electronic control proportional valve 2 should meet the following requirements: (1) the cooling circulation fuel passage should be as close as possible to the electronically controlled elements such as a coil and an armature of the electronic control proportional valve; (2) the flow of the cooling fuel introduced into the cooling circulation fuel passage should enable the temperatures of the electronically controlled elements such as the coil and the armature to be reduced within the working temperature range. In order to enable the cooling circulation fuel passage to meet the requirements, it is necessary to perform simulation calculations and experiments for different types of armatures in advance to determine the specific parametric information such as the spatial arrangement and size of the cooling circulation fuel passage in each type. Here, the simulation calculations and experiments for different types of armatures may be performed with reference to the methods in the prior art and therefore will not be described in detail here. The above-mentioned design is advantageous in that the temperatures of the armature and the coil of the electronic control proportional valve 2 in the working state are reduced by the cooling circulation fuel passage provided in the electronic control proportional valve 2, so that the electronically controlled elements work in the normal temperature range, thereby permitting the use of the electronic control proportional valve 2 for throttling the feed of fuel to the pump. The disadvantages of the mechanical fuel quantity adjustment are overcome by using the electronic control proportional valve 2, and the fuel supply flow is adjusted with increased accuracy, flexibility, and response speed, thereby achieving a more accurate match between the fuel supply quantity from the pump and the working condition of the diesel engine. Thus, the performance degradation caused by insufficient fuel supply is avoided. Also, an excess flow is reduced during working, thereby reducing the actual load on the pump.

[0050] As shown in FIG. 3, in the fuel feeding stage, the fuel inlet valve 10312 is configured to be opened under the action of the fuel feed pressure from the electronic control proportional valve 2, and the fuel outlet valve 10322 is configured to be sealed to the fuel outlet valve seat 10321 under the action of the back pressure from the fuel flow flowing out therethrough. The electronic control proportional valve 2 for feeding a low-pressure heavy fuel starts charging the fuel into the high-pressure fuel chamber 1041 from the fuel inlet hole. The quantity of fuel to be fed is controlled by adjusting the opening degree of the electronic control proportional valve 2, so as to meet the requirements in different working conditions of the high-pressure fuel pump. Here, in the fuel pumping stage, the guide piston component 107 moves upward, the plunger 1043 compresses the heavy fuel in the high-pressure fuel chamber 1041, and thus the pressure of the heavy fuel gradually increases. When the fuel pressure in the high-pressure fuel chamber 1041 is greater than the fuel feed pressure, the fuel inlet valve 10312 is closed. Because the high-pressure fuel outlet chamber 1033 is connected to the high-pressure fuel chamber 1041, when the fuel pressure in the high-pressure fuel chamber 1041 exceeds the back pressure and a spring force exerted on the fuel outlet valve, the fuel outlet valve 10322 is opened, and the high-pressure fuel is discharged from the central hole of the pump cover 102 through the fuel outlet valve spring seat 10324. FIG. 4 shows the structure of the plunger and barrel assembly in the prior art. As shown in FIG. 4, the heavy fuel pump for the high-pressure common rail system in the prior art employs a mechanical design. A fuel inlet conduit 505 is provided in the plunger barrel 1042, the plunger 1043 is slidably inserted into the plunger barrel 1042, and no fuel inlet valve component is disposed. During working, when the fuel is sucked and compressed alternately, part of the pressurized fuel will flow from the fuel inlet conduit 505 back to the low-pressure fuel inlet passage, which will result in a large change in pressure in the fuel inlet conduit 505. As a result, cavitation erosion is likely to occur at the related position(s) in the fuel inlet conduit 505. This is also one of the main forms of damage of the plunger and barrel assemblies which have been observed and recorded in the actual experiments for ships. Compared with the structure of the high-pressure fuel pump in the prior art, the fuel inlet valve component 1031 is additionally provided in the high-pressure fuel pump 1 in the present disclosure, so that when the high-pressure fuel chamber of the plunger barrel 1042 is changed from the fuel suction state to the fuel compression state, the fuel inlet valve component is quickly closed to ensure stable pressure at the related position(s) in the fuel inlet passage of the fuel inlet valve seat 10311, thereby effectively preventing cavitation erosion.

[0051] FIG. 5 shows the fitting relationship between the plunger and barrel assembly 104 and the pump body 101, and between the fuel inlet valve seat 10311 and the upper spring seat 109. Referring to FIG. 5, the plunger and barrel assembly 104 comprises: a plunger barrel 1042, provided in the lower end of the fuel inlet valve seat 10311; and a plunger 1043, slidably inserted into the central hole of the plunger barrel 1042. The high-pressure fuel chamber 1041 is formed by the plunger barrel 1042, the plunger 1043, together with the fuel inlet valve seat 10311. The inner wall of the plunger barrel 1042 is provided with a first annular groove 10421 and a second annular groove 10422. The pump body 101 is provided with a mixed oil outlet passage (not shown in the drawings) and a lubricating oil supply passage 1012. The mixed oil outlet passage communicates with the first annular groove 10421 via a mixed oil passage 10423 in the plunger barrel 1042. The mixed oil formed at the first annular groove 10421 flows out to a waste oil tank (not shown in the drawings) through the mixed oil outlet passage and through the mixed oil passage 10423. The lubricating oil supply passage 1012 communicates with the second annular groove 10422 via a lubricating oil passage 10424 in the plunger barrel 1042. The first annular groove 10421 is located above the second annular groove 10422. The lubricating oil entering the second annular groove 10422 includes the following functions. (1) It has the effect of isolating the fuel entering the gap between the plunger 1043 and the plunger barrel 1042, from the high-pressure fuel chamber 1041 above the plunger 1043, thereby preventing flow of the fuel into the transmission parts under the plunger 1043 to avoid contamination of the lubricating oil system of the whole machine by the fuel intruding into the transmission parts under the plunger 1043. (2) It allows all the friction surfaces under the plunger 1043 to be in a state of being lubricated by clean lubricating oil, thereby improving the friction state of the plunger 1043. The lubricating oil is cleaner than the heavy fuel above thereof, and the lubricating oil contains an additive for improving friction and can form a better oil film than lubrication by the heavy fuel.

[0052] The low-speed engine in the prior art permits the heavy fuel to leak below the plunger 1043, and then the leaked heavy fuel is collected separately. However, the leaked heavy fuel has the risk of corroding the plunger spring 105 and other parts under the plunger 1043. In the high-pressure fuel pump 1 of the present disclosure, the leakage of the heavy fuel can be completely prevented by using a small amount of lubricating oil in the second annular groove 10422 of the plunger barrel 1042 of the plunger and barrel assembly 104, which prevents corrosion of important parts such as the plunger spring 105 under the plunger barrel 1042 by the leaked heavy fuel. Furthermore, because the guide piston under the plunger of the low-speed engine in the prior art is provided with a complicated dynamic sealing mechanism, the high-pressure fuel pump has drawbacks such as having a larger overall vertical height and high cost. In the high-pressure fuel pump 1 of the present disclosure, the vertical height of the guide piston 1071 can be effectively shortened by sealing the heavy fuel with a small amount of lubricating oil (the traditional heavy fuel guide piston has a long heavy fuel sealing section), so that the vertical height of the pump of the high-pressure fuel pump is reduced, and the overall weight of the high-pressure fuel pump is reduced. It is experimentally known that the vertical height of the high-pressure fuel pump in the technical solution of the present disclosure is reduced by 1/3, as compared with the high-pressure fuel pump of the low-speed engine in the prior art.

[0053] Referring to FIG. 2 and FIGS. 7 to 14, the lower spring seat component 106 is disposed under the plunger and barrel assembly 104. The lower spring seat component 106 comprises: an outer spring seat 1061, having a boss-type structure thinner on the outside and thicker in the middle as a whole. The outer spring seat 1061 is mainly subjected to a pressure transmitted from the plunger 1043 to the upper sphere during working. The pressure results in a stress field distributed in a tapered shape in the outer spring seat 1061. The outer spring seat 1061 is provided in a boss shape corresponding thereto, so that the mass of the outer spring seat 1061 can be reduced on the premise of having satisfactory strength, thereby reducing the mass of the outer spring seat 1061 in movement. The thicker part in the middle of the boss also provides space for designing a spherical surface and a fuel passage in its middle.

[0054] The upper end surface of the outer spring seat 1061 is provided with a third counterbore 10611 with a concave spherical surface. The lower spring seat component 106 further comprises an upper sphere 1062. The lower part of the upper sphere 1062 is mounted in the third counterbore 10611, and the lower end surface of the upper sphere 1062 is provided with a convex spherical surface mating with the concave spherical surface.

[0055] The lower spring seat component further comprises an inner spring seat 1063, sleeved on the upper part of the upper sphere 1062. The inner spring seat 1063 has a first axial through hole 10631 penetrating its upper and lower end surfaces.

[0056] The lower cylindrical head 10431 of the plunger 1043 is positionally limited in the first axial through hole 10631, and the lower cylindrical head 10431 of the plunger 1043 has a lower end surface abutting against the upper end surface of the upper sphere 1062. As experimentally known, when the plunger 1043 is working, a flat surface at the tail of the plunger 1043 may be partially subjected to an excessively large force when being pressed (as shown in FIG. 6, an angle with a degree of β is formed between the upper sphere 1062 and the plunger 1043 in FIG. 6), because of an error in parallelism between the flat surface at the tail thereof and a corresponding surface against which it is pressed (a surface of the guide piston or the spring seat). The uneven force distribution will create an additional moment of force on the plunger 1043, which will cause additional load and energy loss to the system, thereby affecting the dynamic characteristics of the system. When the lower spring seat component 106 with a spherical surface is disposed between the plunger 1043 and the guide piston 1071, even if there is a large parallelism error between the upper end surface of the guide piston 1071 and the end surface at the tail of the plunger 1043, the angle can be automatically adjusted by the spherical surface, so that the contact surfaces between the upper sphere 1062 and the plunger 1043 are kept in full contact with each other (the state of FIG. 6 is changed to the state of FIG. 7, in FIG. 7, the two contact surfaces of the upper sphere 1062 and the plunger 1043 are attached to each other). In this way, partial contact is eliminated, the overall force application is balanced, and the tendency of excessive local stress is alleviated. Moreover, a supporting force exerted on the plunger 1043 by the upper sphere 1062 passes through the center of the spherical surface, to eliminate the additional bending moment, thereby optimizing the dynamic characteristics and improving the load-bearing capacity of the system.

[0057] Optionally, referring to FIGS. 9 to 14, a spherical hole 10612 is provided at the center of the third counterbore 10611, a third annular groove 10613 is provided in the lower end surface of the outer spring seat 1061, and the spherical hole 10612 communicates with the third annular groove 10613 via a lubricating oil inlet conduit 10614. The lubricating oil inlet conduit 10614 communicates with a piston oil passage in the guide piston 1071. The lubricating oil passing through the lubricating oil inlet conduit 10614 forms an oil film at the concave spherical surface of the outer spring seat 1061, which can effectively prevent fretting damage between the convex spherical surface of the upper sphere 1062 and the concave spherical surface of the outer spring seat 1061. The spherical hole 10612 supplies the lubricating oil to the spherical surfaces for lubricating the spherical surfaces. The lubricating oil creates an elastohydrodynamic lubrication effect on the spherical surfaces, so as to reduce the wear rate, reduce contact stress, reduce fretting damage, and increase the load-bearing capacity and fatigue strength of the spherical surfaces.

[0058] As shown in FIG. 9, the outer surface of the outer spring seat 1061 is formed as a tapered surface, which is provided with a lubricating oil outlet passage 10615. The lubricating oil outlet passage 10615 communicates with the lower end surface of the outer spring seat 1061. The lubricating oil outlet passage 10615 is provided obliquely relative to the bottom surface of the outer spring seat 1061. The lubricating oil outlet passage 10615 can communicate the regions above and below the outer spring seat 1061 with each other, so that the lubricating oil upstream of the outer spring seat 1061 smoothly flows downward, to prevent additional load caused by compression of the lubricating oil upon the lubricating oil chamber above the outer spring seat 1061 is filled with the lubricating oil. The lubricating oil outlet passage 10615 is provided in the outer tapered surface of the outer spring seat 1061, which can avoid blockage of the flow area of the lubricating oil outlet passage 10615 by the plunger spring 105, so that the flow area is not affected by the position of the plunger spring 105.

[0059] Optionally, as shown in FIG. 10, eight lubricating oil outlet passages 10615 may be provided. The eight lubricating oil outlet passages 10615 communicate to the bottom end surface of the outer spring seat 1061, respectively, so that it can be ensured that the lubricating oil can smoothly flow out from the space above the outer spring seat 1061 to avoid additional load caused by the accumulation of the lubricating oil. Moreover, the lubricating oil outlet passages 10615 are provided obliquely in the tapered surface of the outer spring seat 1061, which also prevents the unsmooth flow and hence accumulation of the lubricating oil caused by blockage of the lubricating oil outlet passages 10615 by the plunger spring 105. As shown in FIG. 12, a circumferential annular groove 10621 is provided in the circumferential direction of the upper sphere 1062. As shown in FIG. 13, a positioning pin 1064 is mounted in the circumferential annular groove 10621 through a positioning pin hole 10616 of the outer spring seat 1061. Here, the spacing between the upper and lower surfaces of the circumferential annular groove 10621 is larger than the cylindrical diameter of a part of the positioning pin 1064 that is located in the circumferential annular groove 10621. Optionally, the upper sphere 1062 and the outer spring seat 1061 are connected by using a threaded positioning pin 1064. The positioning pin 1064 is fixed to the outer spring seat 1061 by means of threads, and the head of the positioning pin 1064 is provided as a cylindrical surface and serves as a positioning part. Optionally, the upper sphere 1062 is provided with a corresponding circumferential annular groove 10621, in which the head of the positioning pin 1064 is mounted. The circumferential annular groove 10621 in the upper sphere 1062 may also be replaced with a round hole. The positioning pin 1064 can substantially position the upper sphere 1062 and the outer spring seat 1061, so as to prevent the upper sphere 1062 from being detached from the outer spring seat 1061 during reciprocating movement when the plunger 1043 is separated from the upper sphere 1062.

[0060] Optionally, as shown in FIG. 11, the first axial through hole 10631 provided inside the inner spring seat 1063 comprises: a seventh hole 10632, an eighth hole 10633, and a ninth hole 10634 having diameters gradually increasing from top to bottom. A first guide hole 10635 having a diameter gradually increasing from top to bottom in the vertical direction is provided between the eighth hole 10633 and the ninth hole 10634. A second guide hole 10636 having a diameter gradually increasing from top to bottom in the vertical direction is provided on a side of the ninth hole 10634 facing the upper sphere 1062. The walls of the first guide hole 10635 and the second guide hole 10636 are formed as tapered guide surfaces 10637. The upper part of the upper sphere 1062 is positioned partially in the ninth hole 10634 through the second guide hole 10636. Here, as shown in FIG. 9, the upper end surface of the lower cylindrical head 10431 of the plunger 1043 abuts against the upper end surface of the eighth hole 10633. The wall of the eighth hole 10633 is attached to the lower cylindrical head 10431 of the plunger 1043 in an annular surface. As shown in FIG. 9, the walls of the first guide hole 10635 and the second guide hole 10636 are formed as tapered guide surfaces 10637. Therefore, if the plunger 1043 and the upper sphere 1062 are separated or the inner spring seat 1063 and the plunger 1043 are separated, when the plunger 1043 impacts on the upper sphere 1062 again, the tapered guide surfaces 10637 will automatically align the plunger 1043 with the upper sphere 1062, align the plunger 1043 with the inner spring seat 1063, and align the inner spring seat 1063 with the upper sphere 1062, so as to prevent a large angular deviation and radial displacement between the plunger 1043 and the inner spring seat 1063 and between the inner spring seat 1063 and the upper sphere 1062. Thus, it can be ensured that the system is in a proper position even when an impact occurs, so that the overall force application is balanced. Specifically, when the plunger 1043 is jammed, the inner spring seat 1063 is relatively stationary (i.e., jammed at the top dead center), and the outer spring seat 1061 and the upper sphere 1062 will impact thereon up and down. The inner spring seat 1063 and the plunger 1043 may not be centered relative to the upper sphere 1062 during impact, which may cause a force to be applied locally during the impact. The inner spring seat 1063 is provided with the tapered guide surfaces 10637, which can improve the centering of the spring seat 1063 and the plunger 1043 relative to the upper sphere 1062. Even when an impact occurs, the inner spring seat 1063 and the upper sphere 1062 can be automatically aligned with each other to improve the tendency of uneven force application.

[0061] Optionally, there is a gap greater than or equal to 1 mm between the upper sphere 1062 and the ninth hole 10634. Optionally, there is a gap of 1 mm between the outer cylindrical surface of the upper sphere 1062 and the wall of the ninth hole 10634. The outer spring seat 1061 and the upper sphere 1062 are fitted to each other at their spherical surfaces and have a relatively large gap (of the order of millimeter) and thus are freely slidable relative to each other. Therefore, during the working process of the plunger 1043, if there is an angular error between the lower end surface of the lower cylindrical head 10431 of the plunger 1043 and the upper end surface of the upper sphere 1062, when the plunger 1043 moves downward and impacts on the upper sphere 1062, the upper sphere 1062 will slide relative to the outer spring seat 1061 to automatically compensate for the angular error. In this way, when additional load is applied to the plunger 1043, the force is uniformly applied to the lower end surface of the lower cylindrical head 10431 of the plunger 1043 and to the upper end surface of the guide piston 1071, whereby excessive local stress can be effectively prevented.

[0062] Optionally, there is a gap greater than or equal to 1 mm between the third counterbore 10611 and the upper sphere 1062. Optionally, there is a gap of 1 mm between the outer cylindrical surface of the upper sphere 1062 and the cylindrical surface of the third counterbore 10611. In other words, there is a relatively large gap (of 1 mm) between the upper sphere 1062 and the inner spring seat 1063, and there is a relatively large gap (of 1 mm) between the positioning pin 1064 and the upper sphere 1062, between the positioning pin 1064 and the outer spring seat 1061, and between the upper sphere 1062 and the inner spring seat 1063, respectively. Thus, it is ensured that effective degrees of freedom of rotation of the upper sphere 1062 will not be restricted by the positioning pin 1064 during its radial movement, and effective degrees of freedom of rotation of the plunger 1043 and the upper sphere 1062 will not be restricted by the inner spring seat 1063 during their radial movements, which prevents additional radial load on the plunger 1043.

[0063] Optionally, there are gaps of the order of millimeter between the upper sphere 1062 and the ninth hole 10634 and between the third counterbore 10611 and the upper sphere 1062, respectively. The upper sphere 1062 is permitted to have a macroscopic angular error relative to the outer spring seat 1061, thereby achieving the technical effects of preventing jamming of the spherical surfaces, eliminating partial contact, balancing the overall force application, and alleviating the tendency of excessive local stress. Furthermore, as shown in FIG. 11, optionally, relief grooves 10638 are formed in both the outer circumferential surface of the inner spring seat 1063 and the wall of the eighth hole 10633. An annular weight-reducing groove 10639 is provided around the central axis of the inner spring seat 1063, in the upper end surface of the inner spring seat 1063.

[0064] Referring to FIG. 2, the high-pressure fuel pump 1 in the present disclosure further comprises: an upper spring seat 109, sleeved on the plunger barrel 1042 and located at the upper end of the inner spring seat 1063. The plunger spring 105 comprises: a first plunger spring 1051, compressively mounted between the upper spring seat 109 and the outer spring seat 1061; and a second plunger spring 1052, compressively mounted between the upper spring seat 109 and the inner spring seat 1063. Optionally, the diameters of the concave spherical surface in the outer spring seat 1061 and the convex spherical surface of the upper sphere 1062 are each 20 to 100 times the diameter of the plunger 1043. There is a parallelism error of low order of magnitude, generally of the order of magnitude of 0.01 mm, between the tail of the plunger 1043 and the upper end surface of the guide piston 1071, which has low requirements for angle adjustability of the spherical surfaces. Therefore, spherical surface adjustment at a small angle can also meet the angle adjustment requirements. When the concave spherical surface in the outer spring seat 1061 and the convex spherical surface of the upper sphere 1062 are relatively large, only a small part of forces acting on the two surfaces when being pressed is transformed into tensile stress. A metal material generally has higher compressive strength than tensile strength, and the compressive stress is not likely to cause fatigue. Therefore, the proportion of tensile stress can be reduced by choosing a large spherical surface, so as to increase the load-bearing capacity and fatigue strength of the material.

[0065] Optionally, referring to FIGS. 15 to 21, the guide piston component 107 comprises: a guide piston 1071, provided with a first mounting hole 10711 at a central location of its upper end surface and provided with a second mounting hole 10712 at its lower end surface, wherein the first mounting hole 10711 and the second mounting hole 10712 communicate with each other via a communicating hole 10713, and the lower spring seat component 106 is mounted in the first mounting hole 10711; a roller component 1072, including a roller 10728 mounted in the second mounting hole 10712, a roller bushing 10722 interference-fitted in the roller 10728, and thrust bearings 10723 interference-fitted at both ends of the roller 10728 in its axial direction, wherein an annular groove 10724 is provided in the axial direction of the roller 10728, and an arc-shaped transitional connection is formed between the bottom of the annular groove 10724 and an end surface of the roller 10728 in the axial direction; and a roller pin 1073, fitted with a clearance in the roller bushing 10722. A boss 10714 is disposed protruding from a wall of the second mounting hole 10712, and the boss 10714 is in contact with the thrust bearing 10723. A plurality of first radial oil grooves 10715 are uniformly arranged in the boss 10714 along its radial direction, and the first radial oil grooves 10715 are provided with respect to the thrust bearing 10723. Optionally, four first radial oil grooves 10715 may be provided.

[0066] The roller bushing 10722, the thrust bearing 10723, and the roller 10728 are interference-fitted to one another, so as to reduce movable surfaces and to increase the movement speeds of friction surfaces among the three parts. According to the theory of dynamic pressure lubrication, a friction coefficient decreases within a certain range as a relative movement speed between the friction surfaces increases. Therefore, the effect of dynamic pressure lubrication can be enhanced by increasing the relative movement speed, so that a thicker dynamic pressure oil film is formed between the corresponding friction surfaces, to avoid contact of solids and to reduce the friction coefficient and abrasion.

[0067] The following effects can be achieved by providing the communicating hole. (1) When the lubricating oil upstream of the guide piston 1071 is flowing down from the communicating hole 10713, the lubricating oil is evenly distributed in the middle right above the second mounting hole 10712 of the roller 10728, the lubricating oil is evenly distributed on generatrix of the roller 10728, and the lubricating oil is distributed on the surface of the roller 10728 without being affected by the forward or reverse rotation (namely, the lubricating oil can be evenly distributed on the surface of the roller 10728 regardless of the forward or reverse rotation of the roller 10728). (2) The vertical force distribution on the guide piston 1071 is improved. In other words, the pressure from the plunger 1043 is distributed onto thicker positions around the communicating hole 10713. In this way, the overall force application is balanced, the maximum stress is reduced, and the load-bearing capacity and reliability of the system are increased. In the guide piston in the prior art, the communicating hole is provided around the center of the guide piston, and the communicating hole is at a physical position which is thinner and subjected to larger stress. (3) In the pump assembly, when the guide piston 1071 is fitted with the outer spring seat 1061, the lubricating oil outlet passage 10615 of the outer spring seat 1061 communicates with the plunger and barrel assembly 104, so that the lubricating oil leaking from the upper part of the plunger and barrel assembly 104 can be discharged therethrough, thereby increasing the actual area of flow of the lubricating oil.

[0068] Optionally, the annular groove 10724 is formed by machining after the inner hole and the outer cylindrical portion of the roller 10728 are accurately machined. As shown in FIGS. 22 and 23, the annular groove 10724 is provided to reduce the rigidity of both ends of the roller 10728. When the surface of the roller 10728 is subjected to radial pressure, the roller 10728 may be automatically deformed in the vicinity of the annular groove 10724. Moreover, after the annular groove 10724 is machined, the outer cylindrical portion and the inner hole of the roller 10728 automatically collapse, so that microscopic arcuate surfaces are formed at both ends of the inner hole and the outer cylindrical portion of the roller 10728, to reduce the geometric stress concentration at both ends of the roller 10728, thereby balancing the force applied to the surface of the roller 10728. Here, the geometric stress concentration herein means that contact stress at both ends of the generatrix of the roller 10728 is significantly greater than contact stress in the middle thereof when the surface of the roller 10728 is stressed. The wall of the annular groove 10724 is formed in an arcuate shape, which can effectively weaken the geometric stress concentration at the outer cylindrical surface of the roller 10728 and the side pressure effect of the inner hole of the roller 10728 during rotation of the roller 10728. This results in a balanced stress distribution on the inner and outer working surfaces of the roller component 1072, thereby reducing the probability of jamming between the roller component 1072 and the roller pin 1073. Here, the side pressure effect mentioned above specifically refers to a phenomenon in which a shaft, fitted in a hole, is closer to one side of the hole and farther away from the other side of the hole and thus exerts more force on the side closer thereto and less force on the side farther away therefrom during working, because there must be a certain angular error between the shaft and the hole during working, namely, their axes are not parallel to each other.

[0069] A thrust bearing model is formed by the boss 10714 and a corresponding friction surface (an end surface of the roller component 1072). In other words, the first radial oil grooves 10715 are filled with lubricating oil to supply sufficient lubricating oil to the moving surface (the end surface of the roller component 1072). A dynamic pressure oil film is formed on the end surface of the roller 10728 with the speed of movement of the end surface of the roller 10728, to separate the boss 10714 of the guide piston 1071 from the end surface of the roller component 1072, which can reduce wear and reduce the friction coefficient. In the case where the first radial oil grooves 10715 are provided in the boss 10714 of the guide piston 1071, the guide piston 1071 will not rotate relatively, high-pressure and low-pressure oil film regions are relatively stationarily distributed on the friction surface, and thus the roller component 1072 is relatively stationary in the axial direction, as compared with a case where the first radial oil grooves 10715 are provided in the roller component 1072. If the first radial oil grooves 10715 are provided in a moving part (the end surface of the roller component 1072), the relative movement of the first radial oil grooves 10715 relative to the guide piston 1071 will result in a relative moving distribution of the oil film, which will lead to an unnecessary additional vibration of the roller 10728 in the axial direction and hence to reduced overall dynamic performance of the roller component 1072.

[0070] With regard to the roller 10728, the roller 10728 is designed with end portions grooved and deformed to reduce boundary stress. Specifically, during machining of the roller 10728, the outer cylindrical portion of the roller 10728 and the inner hole of the roller 10728 are finished first, and then its two end surfaces in the axial direction are grooved with the annular grooves 10724. After the grooving process is completed, the generatrices of the outer cylindrical portion and the inner hole of the roller 10728 are naturally deformed into arcs, which can effectively weaken the geometric stress concentration at the outer cylindrical surface of the roller and the side pressure effect of the inner hole of the roller 10728 during rotation of the roller 10728. This results in a balanced stress distribution on the inner and outer working surfaces of the roller component 1072, thereby reducing the probability of jamming between the roller component 1072 and the roller pin 1073. The roller bushing 10722 and the roller 10728 are fitted in an interference fit manner, to increase the relative speed of the moving surface of the roller bushing 10722 relative to the roller pin 1073. The end surface of the roller bushing 10722 moves at a high speed so that an effective dynamic pressure oil film is formed between it and the roller pin 1073, thereby improving the dynamic pressure lubrication effect and reducing the probability of jamming between the roller bushing 10722 and the roller pin 1073. The roller 10728 and the thrust bearing 10723 are fitted in an interference fit manner, to increase the relative speed of the moving surface of the thrust bearing 10723 relative to the boss 10714. The end surface of the thrust bearing 10723 moves at a high speed so that an effective dynamic pressure oil film is formed between it and the boss 10714, which can prevent attachment of the boss 10714 to the thrust bearing 10723, so as to avoid excessive wear of the end surface of the thrust bearing 10723 caused by insufficient oil supply thereto. The dynamic pressure oil film is formed to improve the dynamic pressure lubrication effect and to reduce the possibility of jamming between the thrust bearing 10723 and the boss 10714.

[0071] Optionally, as shown in FIG. 19, the outer surface of the roller pin 1073 is provided as a cylindrical surface, which is provided with first waist-shaped grooves 10731 and second waist-shaped grooves 10732 at its two positions, respectively. The first waist-shaped grooves 10731 and the second waist-shaped grooves 10732 are provided in the middle positions of the roller pin 1073, wherein the second waist-shaped groove 10732 is provided in the first waist-shaped groove 10731 and recessed inward relative to the first waist-shaped groove 10731, and a stepped shape is formed by them together. The first waist-shaped groove 10731 and the second waist-shaped groove 10732 each are in an elongated shape. There is a large contact area between the lubricating oil in the first waist-shaped groove 10731 and in the second waist-shaped groove 10732 and the corresponding friction surface. With full use of the moving speed of the corresponding moving surface, more lubricating oil is brought onto the load-bearing surface to form a dynamic pressure oil film, so as to form a thicker lubricating oil film. The two edges of the first waist-shaped groove 10731 are provided in a waist shape to reduce stress concentration caused by the groove provided in the surface of the roller pin 1073. The flow of the lubricating oil on the surface can be increased by providing the first waist-shaped groove 10731 and the second waist-shaped groove 10732. Optionally, a small-angle wedge-shaped groove with an angle ranging from 5° to 10° is formed between the first waist-shaped groove 10731 located on an outer layer and the outer surface of the roller bushing 10722, and a second oil hole 10733 is provided in each second waist-shaped groove 10732 located in an inner layer. An angle of 70° to 120° (the actual value may be determined according to the simulation calculation results and may be selected to be 90° in an example) is formed between the two second waist-shaped grooves in a plane perpendicular to the axial direction of the roller pin, and the two second waist-shaped grooves are located directly above the pressure-bearing region. This reduces the influence of the waist-shaped grooves provided in the surface on the area of the pressure-bearing region while ensuring sufficient oil supply to the friction surface, so that the pressure-bearing region has a larger angle, and the oil film on the pressure-bearing region has a smaller average pressure. A small-angle convergent wedge shape is formed between the first waist-shaped groove 10731 and the corresponding friction surface, which strengthens the squeezing effect in dynamic pressure lubrication. The second waist-shaped groove 10732 is mainly used for storing more lubricating oil to ensure sufficient oil supply to the friction surface. Thus, the lubrication of the surface of the roller pin will not be affected even if it is supplied with the lubricating oil poorly for a short time, thereby reducing the probability of jamming of the system when there is a problem in the lubrication system. The two second oil holes 10733 at the two positions communicate with each other via a lubricating oil outlet passage, and the two second oil holes 10733 are provided at an angle of 90°.

[0072] Optionally, as shown in FIGS. 17, 18, and 34, the outer surface of the guide piston 1071 is provided as a cylindrical surface, which is provided with a partial circumferential oil groove 10729 and a circumferential oil groove 10725, a first axial oil groove 10716, and a vertical groove 10717. The vertical groove 10717 is provided in the circumferential oil groove 10725, and the vertical groove 10717 communicates with the partial circumferential oil groove 10729 via the first axial oil groove 10716. Chamfers of 1° to 10° are provided between the upper and lower edges of the circumferential oil groove 10725 and the upper and lower ends of the guide piston 1071, respectively. A small-angle convergent wedge shape is formed between the chamfer and the corresponding moving surface during movement, to strengthen the squeezing effect in dynamic pressure lubrication. This improves the state of lubrication of the surface of the guide piston 1071, so that a thicker dynamic pressure oil film is established to reduce friction and reduce the probability of jamming. According to related data and experiments, when the chamfer is too large (such as at 45° or 90°), the chamfer cannot enhance the lubrication, but rather has a scraping effect on the corresponding friction surface and will scrap off the lubricating oil from the surface, resulting in a reduced lubrication effect. Referring to FIG. 16, optionally, the cylindrical surface is further provided with an inclined hole 10718 having two ends communicating with the circumferential oil groove 10725 and the inner wall of the second mounting hole 10712, respectively. Referring to FIG. 18, optionally, the cylindrical surface is further provided with a second axial oil groove 10719 communicating with the circumferential oil groove 10725. The cylindrical surface is further provided with a first straight hole 10720 and a second straight hole 10721 connected to each other, the first straight hole 10720 communicates with the first axial oil groove 10716, and the second straight hole 10721 communicates with the first mounting hole 10711. The first straight hole 10720 and the second straight hole 10721 supply lubricating oil to the lower spring seat component 106 inside the guide piston 1071, so as to reduce the wear of the corresponding moving surface. Referring to FIG. 15, optionally, a lubricating oil inlet passage 10735 is provided in the outer cylindrical surface of the roller pin 1073. The lubricating oil inlet passage 10735 is provided with respect to the inclined hole 10718, and the lubricating oil inlet passage 10735 communicates with a lubricating oil outlet passage.

[0073] Optionally, the outer cylindrical surface of the roller pin 1073 is provided with a DLC (Diamond-Like Carbon) coating. The DLC coating has the characteristics of high hardness, low friction coefficient, wear resistance, and high temperature resistance. When poor lubrication occurs between the roller bushing 10722 and the roller pin 1073, a friction pair consisting of the DLC coating and the roller bushing 10722 made of a copper alloy can still operate well, which can further reduce the probability of jamming between the roller pin 1073 and the roller bushing 10722. The roller bushing 10722 is made of a copper alloy. The thrust bearing 10723 is made of a copper alloy. Forced lubrication and dynamic pressure lubrication are used between the roller pin 1073 and the roller bushing 10722. Forced lubrication and dynamic pressure lubrication are used between the thrust bearing 10723 and the boss 10714. The roller bushing 10722 and the thrust bearing 10723 are made of bronze alloy. The bronze alloy having a low friction coefficient, good wear resistance, self-lubricating property and impact resistance allows for improved friction characteristics when solid friction occurs between the inner hole and end surfaces of the roller component 1072 and the corresponding moving surfaces, and allows for a reduced friction coefficient, increased impact resistance, and increased load-bearing capacity. It should be noted that the forced lubrication is a lubrication mode, in which the pressure of the lubricating oil is forcibly increased by an external force in order to establish a thicker lubricating film among contact surfaces of the respective parts. The dynamic pressure lubrication is a lubrication mode, in which the lubricating oil is brought onto a friction surface with the movement of a moving surface to form a dynamic pressure oil film. In general, and at low speed, the parts are forcibly lubricated mainly by using the pressure generated by the lubricating oil pump. At high speed, the parts are lubricated mainly by using the dynamic pressure oil film generated by the movement of the parts.

[0074] As shown in FIGS. 15, 16, 18, and 20, optionally, a first chamfer 10741 is provided at each of the outer circumference of the upper end surface, the outer circumference of the lower end surface, and the circumferential oil groove 10725 of the guide piston 1071. A second chamfer 10726 is provided in the wall of the first mounting hole 10711. A tenth hole 10727 is provided in the wall of the second mounting hole 10712. An eleventh hole 10736 is provided in the outer cylindrical surface of the roller pin 1073. A return spring 10737 and a stop pin 10738 are sequentially placed in the eleventh hole 10736, and the stop pin 10738 partially extends into the tenth hole 10727. The eleventh hole 10736 is provided for mounting the stop pin 10738, so that the roller pin 1073 is stationary relative to the guide piston 1071, to reduce the number of the relative moving surfaces of the roller pin 1073 and the roller 10728, thereby increasing the speeds of the relative moving surfaces and hence enhancing the dynamic pressure lubrication effect. Its principle is the same as that of the interference fit of the roller and the bushing. Optionally, when assembling the roller component 1072 and the roller pin 1073 into the guide piston 1071, first the roller bushing 10722 and the thrust bearing 10723 are mounted to the roller 10728 by means of cold assembling. Then, the return spring 10737 and the stop pin 10738 are sequentially placed in the eleventh hole 10736 of the roller pin 1073. Next, the roller component 1072 is placed in the lower part of the guide piston 1071, with the roller pin 1073 passing sequentially through one side of the tenth hole 10727 at the lower end of the guide piston 1071, the inner hole of the roller component 1072 (specifically the inner hole of the roller bushing 10722), and the other side of the tenth hole 10727 at the lower end of the guide piston. Then, the stop pin 10738 is pressed with a hand so that its height is lower than that of the second mounting hole 10712. At the same time, the roller pin 1073 is pushed until the stop pin 10738 springs into the tenth hole 10727 of the guide piston 1071 under the action of the return spring 10737. In other words, the lubricating oil flowing out of the pump body 101 of the fuel injection pump flows into the circumferential oil groove 10725 through the second axial oil groove 10719 and then flows into the partial circumferential oil groove 10729 through the vertical groove 10717 and the first axial oil groove 10716, so as to achieve the lubrication between the guide piston 1071 and the pump body 101 of the fuel injection pump. Moreover, because the gap between the guide piston 1071 and the central hole of the pump body 101 in which it is fitted is small, the lubricating oil entering the second axial oil groove 10719 and the circumferential oil groove 10725 of the guide piston 1071 is maintained at a certain pressure, and a lubricating oil film can be formed between the outer cylindrical portion of the guide piston 1071 and the central hole of the pump body 101. Meanwhile, the lubricating oil in the second axial oil groove 10719 partially flows into the lubricating oil inlet passage 10735 and is deep into the roller pin 1073, through the inclined hole 10718, and then flows out to the outer cylindrical surface of the roller pin 1073 through the second oil holes 10733, so as to deeply lubricate the roller pin 1073 and the roller bushing 10722 and form a lubricating oil film between the roller pin 1073 and the rolling bushing 10722.

[0075] Optionally, as shown in FIG. 18, the first chamfer 10741 and the second chamfer 10726 each have an angle ranging from 1° to 10°. A small-angle convergent wedge shape can be formed when the guide piston 1071 is fitted in the central hole in the pump body 101, so as to strengthen the squeezing effect in dynamic pressure lubrication and to increase the thickness of the oil film on the surface of the guide piston 1071 during operation, thereby reducing the probability of jamming between the guide piston 1071 and the pump body 101.

[0076] Optionally, as shown in FIGS. 20 and 21, the lubricating oil inlet passage 10735 comprises: a third radial oil passage 10739 provided along the radial direction of the roller pin 1073 and an axial oil passage 10740 provided along the axial direction of the roller pin 1073. The third radial oil passage 10739 is connected to the axial oil passage 10740. The axial oil passage 10740 is connected to the second oil hole 10733 in the second waist-shaped groove 10732.

[0077] The high-pressure fuel pump 1 described above has the following effects.

(1) The technical problem of insufficient flexibility in the prior mechanical adjustment method can be solved by adjusting feed of the heavy fuel using the electronic control proportional valve 2. In view of this, a cooling circulating fuel passage is provided inside the electronic control proportional valve 2, through which the cooling fuel flowing in the pump body 101 enters the electronic control proportional valve 2 to cool the electronically controlled elements in the electronic control proportional valve 2 in a targeted manner, so that the electronically controlled elements of the electronic control proportional valve 2 are maintained within the normal temperature range, thereby permitting the use of the electronic control proportional valve 2 for throttling the feed of fuel to

the high-pressure fuel pump. The disadvantages of the mechanical fuel quantity adjustment are overcome by the electronic control proportional valve 2, and the fuel supply flow is adjusted with increased accuracy, flexibility, and response speed, thereby achieving a more accurate match between the fuel supply quantity from the pump and the working condition of the diesel engine. Thus, the performance degradation caused by insufficient fuel supply is avoided. Also, an excessive flow is reduced during working, thereby reducing the actual load on the high-pressure fuel pump 1.

(2) The fuel inlet valve component 1031 is additionally provided so that when the high-pressure fuel chamber 1041 of the plunger barrel 1042 is changed from the fuel suction state to the fuel compression state, the fuel inlet valve component is quickly closed to ensure stable pressure at the related position(s) in the fuel inlet passage of the fuel inlet valve seat 10311, thereby effectively preventing cavitation erosion.

(3) The leakage of the heavy fuel can be completely prevented by using a small amount of lubricating oil in the second annular groove 10422 of the plunger and barrel assembly 104, which prevents corrosion of important parts such as the plunger spring 105 under the plunger barrel 1042 by the leaked heavy fuel. Furthermore, in the present disclosure, the vertical height of the guide piston 1071 can be effectively shortened by sealing the heavy fuel with a small amount of lubricating oil (it is unnecessary to provide the guide piston 1071 with a long sealing section as in the low-speed engine in the prior art), thereby reducing the vertical height of the pump of the high-pressure fuel pump 1 and reducing the overall weight of the high-pressure fuel pump 1. It is experimentally known that the vertical height of the high-pressure fuel pump 1 in the solution of the present disclosure is reduced by 1/3 as compared with the high-pressure fuel pump in the prior art.

(4) The outer spring seat 1061 has a boss-type structure thinner on the outside and thicker in the middle as a whole. The outer spring seat 1061 is mainly subjected to a pressure transmitted from the plunger 1043 to the upper sphere 1062 during working. The pressure results in a stress field distributed in a tapered shape in the outer spring seat 1061. The outer spring seat 1061 is provided in a boss shape corresponding thereto, so that the mass of the outer spring seat 1061 can be reduced on the premise of having satisfactory strength, thereby reducing its mass in movement. The thicker part in the middle of the boss also provides space for designing a spherical surface and a fuel passage in its middle.

(5) The outer spring seat 1061 and the upper sphere 1062 are fitted to each other in spherical surface. When the lower spring seat component 106 with a spherical surface is disposed between the plunger 1043 and the guide piston 1071, even if there is a large parallelism error between the upper end surface of the guide piston 1071 and the end surface at the tail of the plunger 1043, the angle can be automatically adjusted by the spherical surface, so that the contact surfaces between the upper sphere 1062 and the plunger 1043 are kept in full contact with each other. In this way, partial contact is eliminated, the overall force application is balanced, and the tendency of excessive local stress is alleviated. Moreover, a supporting force exerted on the plunger 1043 by the upper sphere 1062 passes through the center of the spherical surface to eliminate the additional bending moment, thereby optimizing the dynamic characteristics and improving the load-bearing capacity of the system.

(6) The spherical hole 10612 supplies lubricating oil to the spherical surfaces for lubricating the spherical surfaces. The lubricating oil creates an elastohydrodynamic lubrication effect on the spherical surfaces, so as to reduce the wear rate, reduce contact stress, reduce fretting damage, and increase the load-bearing capacity and fatigue strength of the spherical surfaces.

(7) The outer surface of the outer spring seat 1061 is formed as a tapered surface. The lubricating oil outlet passage 10615 is provided in the outer tapered surface of the outer spring seat 1061, which can avoid blockage of the flow area of the lubricating oil outlet passage 10615 by the plunger spring 105, so that the flow area is not affected by the position of the plunger spring 105.

(8) The inner spring seat 1063 is provided with the tapered guide surfaces 10637, which can improve the centering. Even when an impact occurs, the inner spring seat 1063 and the upper sphere 1062 can be automatically aligned with each other to improve the tendency of uneven force application.

(9) A communicating hole 10713 is provided inside the guide piston 1071. Thus, when the lubricating oil upstream of the guide piston 1071 is flowing down from the communicating hole 10713, the lubricating oil is evenly distributed in the middle right above the second mounting hole 10712 of the roller 10728, the lubricating oil is evenly distributed on generatrix of the roller, and the lubricating oil is distributed (can be evenly distributed) on the surface of the roller 10728 without being affected by the forward or reverse rotation. The vertical force distribution on the guide piston is improved. In other words, the pres-

sure from the plunger 1043 is distributed onto thicker positions around the communicating hole 10713. In this way, the overall force application is balanced, the maximum stress is reduced, and the load-bearing capacity and reliability of the system are increased. In the pump assembly, when the guide piston 1071 is fitted with the outer spring seat 1061, the lubricating oil outlet passage 10615 of the outer spring seat 1061 communicates with the lubricating oil leaking from the upper part of the plunger and barrel assembly 104, so as to increase the area of flow of the lubricating oil.

(10) The first radial oil grooves 10715 provided in the boss 10714 are filled with lubricating oil to supply sufficient lubricating oil to the moving surface (the end surface of the roller component 1072). A dynamic pressure oil film is formed on the end surface of the roller 10728 with the speed of movement of the end surface of the roller 10728, to separate the boss 10714 of the guide piston 1071 from the end surface of the roller component 1072, which can reduce wear and reduce the friction coefficient. In the case where the first radial oil grooves 10715 are provided in the boss 10714 of the guide piston 1071, the guide piston 1071 will not rotate relatively, high-pressure and low-pressure oil film regions are relatively stationarily distributed on the friction surface, and thus the roller component 1072 is relatively stationary in the axial direction, as compared with a case where the first radial oil grooves 10715 are provided in the roller component 1072.

(11) The first waist-shaped groove 10731 and the second waist-shaped groove 10732 provided in the roller pin 1073 form an angle of 70° to 120° and are located directly above the pressure-bearing region. This reduces the influence of the waist-shaped grooves provided in the surface on the area of the pressure-bearing region while ensuring sufficient oil supply to the friction surface, so that the pressure-bearing region has a larger angle, and the oil film on the pressure-bearing region has a smaller average pressure. A small-angle convergent wedge shape is formed between the first waist-shaped groove 10731 on the outer layer and the corresponding friction surface, which strengthens the squeezing effect in dynamic pressure lubrication. The second waist-shaped groove 10732 on the inner layer is mainly used for storing more lubricating oil to ensure sufficient oil supply to the friction surface. Thus, the lubrication of the surface of the roller pin will not be affected even if it is supplied with the lubricating oil poorly for a short time, thereby reducing the probability of jamming of the system when there is a problem in the lubrication system.

[0078] Referring to FIGS. 24 and 25, the high-pressure common rail system for a low-speed engine according to the present disclosure is suitable for working at a temperature of 200 °C and at a pressure of 150 MPa, and has a low-pressure circulation function and pressure limiting and pressure maintaining functions. The high-pressure common rail system for a low-speed engine of the present disclosure comprises an integrated common rail for a low-speed engine. The integrated common rail for a low-speed engine mainly comprises parts such as a common rail pipe 12, a flow limiting valve component 13, a fuel inlet end cap 121, an end cover 122, a circulation valve component 16, a pressure limiting valve component 18, a sensor mounting seat 127, a sensor 124, a flow limiting valve mounting seat 125, and a bracket 126. The common rail pipe 12 employs a cylindrical structure. A plurality of first notches 1203 are provided in a side of the outer circumferential surface of the common rail pipe 12, and the corresponding flow limiting valve component 13, the sensor mounting seat 127, and the pressure limiting valve component 18 are mounted into the plurality of first notches 1203, respectively. The specific number of the plurality of first notches 1203 described above may be adjusted according to the number of various components to be mounted correspondingly. A second notch 1204 into which the bracket 126 is mounted is provided in the other side of the common rail pipe 12 opposite to the first notch 1203. Moreover, the end cover 122 and the fuel inlet end cap 121 are mounted on the two sides of the common rail pipe 12.

[0079] Referring to FIG. 26, in the integrated common rail for a low-speed engine according to the present disclosure, first bolts 128 are mounted by using two third waist-shaped holes 131 provided in the bracket 126. The first bolts 128 are fixed to the low-speed engine through the waist-shaped holes so that the integrated common rail is mounted to the low-speed engine. In addition, a low-speed diesel engine has a relatively large number of cylinders, usually, 6 cylinders, 8 cylinders, 12 cylinders, or the like. Accordingly, there are a relatively large number of corresponding brackets 126. In this embodiment, six brackets 126 are disposed, and each of the brackets 126 is provided with third waist-shaped holes 131 to facilitate the mounting and adjustment of the brackets 126 and prevent interference therebetween. Moreover, the bracket 126 is fixed to the common rail pipe 12 by using four screws 130 and is equipped with cylindrical pins to facilitate the positioning of the common rail pipe 12.

[0080] The common rail pipe 12 in this embodiment employs an integral structure, in which a fuel inlet conduit 1201 penetrating both ends of the common rail pipe 12 is provided. This facilitates machining of both sides of the common rail pipe 12 and reduces the machining difficulty. The end cover 122 and the fuel inlet end cap 121 for sealing the common rail pipe 12 are mounted on the two sides thereof. Moreover, the end cover 122 and the fuel inlet end cap 121 are sealed to the common rail pipe 12 by using tapered surfaces, respectively. The end cover 122 is fixed to the common rail pipe 12 by means of screwing and the fuel inlet end cap 121 is fixed to the common rail pipe 12 by means of screwing, whereby the sealing performance of the common rail pipe 12 can be improved.

[0081] The integrated common rail for a low-speed engine according to the present disclosure has the pressure limiting and pressure maintaining functions. The pressure limiting valve component 18 is mounted into the first notch 1203 provided in the common rail pipe 12. The pressure limiting valve component 18 is configured such that when the pressure in the fuel inlet conduit 1201 of the common rail pipe 12 exceeds the limit pressure of the pressure limiting valve component 18, the pressure limiting valve component 18 is opened by overcoming a preload force on a second pressure regulating spring 185 provided therein, thereby achieving the self-adjustment function of the common rail pipe 12. This function enables the pressure in the common rail pipe 12 to be maintained at a certain pressure value when the pressure limiting valve component 185 is stabilized. It can be ensured that the diesel engine operates at low speed and at low torque.

[0082] Further, considering that inferior or low-quality fuel is used as a fuel medium in the integrated common rail for a low-speed engine according to the present disclosure, an air-controlled circulation valve component 16 is mounted onto the end cover 122 in order to prevent solidification of the fuel in the common rail system for a low-speed engine after the low-speed engine is stopped. As shown in FIGS. 27 and 28, optionally, the circulation valve component 16 comprises a first valve core 161, a first valve body 162, a gland 167, a third sealing ring 1624, a fourth sealing ring 1627, a first sealing ring 1628, a second sealing ring 1631, a lower spring seat 164, a fuel return joint 168, a nut 1614, and a first pressure regulating spring 169, wherein the gland 167 is fixed to the first valve body 162 by means of screws 1632. After the low-speed engine is stopped, the pressure in the fuel inlet conduit 1201 of the common rail pipe 12 is reduced to be lower than a spring force from the first pressure regulating spring 169. The spring force pushes the first valve core 161 to move downward. When the first valve core 161 moves to the bottom dead center, the conical seal between the first valve core 161 and the first valve body 162 in the circulation valve component 16 is unsealed, and then the heavy fuel in the fuel inlet conduit 1201 of the common rail pipe 12 can flow back into the fuel tank through the first fuel return passage 1605, the second fuel return passage 1606, and the third fuel return passage 1607, so as to achieve fuel circulation. When the low-speed engine is just started, compressed air is introduced into an air inlet port 1630 in the circulation valve component 16. After the compressed air enters the first cavity, the first valve core 162 is pushed to move toward the direction of the top dead center by the action of the compressed air. When the first valve core 162 moves upward to the top dead center, the conical seal is formed between the first valve core 162 and the first valve body 161, so that the pressure in the common rail pipe 12 is built up.

[0083] While the integrated common rail of the present disclosure is working, the common rail pipe 12 is always in a high-pressure state. In order to ensure the safety of the operator and the low-speed engine, the common rail pipe 12 is equipped with a sensor 124 for detecting the pressure in the fuel inlet conduit 1201 in the common rail pipe 12, which is configured to monitor the pressure in the fuel inlet conduit 1201 in real time. Two sensors 124 are provided as a backup for each other in order to ensure accurate detection.

[0084] In addition, referring to FIGS. 24 and 30, optionally, a flow limiting valve component 13 is disposed between the common rail pipe 12 and the electronic control fuel injector 15. The flow limiting valve component 13 mainly comprises a second valve core 133, a valve seat 131, a second valve body 132, a second pressure regulating spring 135, and a first O-ring seal 136. When the high-pressure fuel pipe between the flow limiting valve component 13 and the electronic control fuel injector 15 is broken or excessive fuel is injected from the electronic control fuel injector 15, the second valve core 133 in the flow limiting valve component 13 adheres to the second valve body 132 by overcoming a spring force from the second pressure regulating spring 135 by the pressure difference, so that the high-pressure fuel in the fuel inlet conduit 1201 of the common rail pipe 12 is hindered from flowing into the electronic control fuel injector 15.

[0085] In addition, the common rail system of the present disclosure works under a working condition at high temperature and at high pressure using inferior fuel. Considering that high-pressure fuel may possibly leak from the high-pressure sealing surfaces of the common rail pipe 12 and the respective parts, and thus cause environmental pollution and endanger the personal safety of workers, fuel return holes 1205 are provided at the respective high-pressure seals for collecting returned fuel. Specifically, a fuel return conduit 1202 is provided in the common rail pipe 12. The fuel return conduit 1202 communicates with the first notches 1203 for collecting the returned fuel leaking from the high-pressure seals of the common rail pipe 12, the flow limiting valve component 13, and the pressure limiting valve component 18.

[0086] Optionally, referring to FIG. 24, optionally, the flow limiting valve component 13 is mounted at one of the first notches 1203 by means of the flow limiting valve mounting seat 125, the pressure limiting valve component 18 is mounted at one of the first notches 1203 by means of the pressure limiting valve mounting seat 123, and the sensor 124 is mounted at one of the first notches 1203 by means of the sensor mounting seat 127. A sealing ring is disposed on each of the flow limiting valve mounting seat 125, the pressure limiting valve mounting seat 123, and the sensor mounting seat 127, so as to prevent fuel leakage. Similarly to the principle of mounting of the pressure limiting valve component 18, the sensor mounting seat 127 and the flow limiting valve mounting seat 125 are first fixed to the common rail pipe 12 by means of screw fixing, and then the sensor 124 and the flow limiting valve component 13 are mounted by means of screw connection.

[0087] Referring to FIG. 24 and FIGS. 26 to 29, optionally, the circulation valve component 16 comprises: a first valve body 161 fixed to the end cover 122, a first valve core 162, a nut 1614, a lower spring seat 164, a sealing ring, a gland 167, and a fuel return joint 168. The first valve body 161 is provided therein with an axial central hole and two first fuel return passages 1605. The axial central hole communicates with the fuel inlet conduit 1201 in the common rail pipe 12, so that the high-pressure fuel in the fuel inlet conduit 1201 in the common rail pipe 12 can enter the third axial through hole of the circulation valve component 16. Two first sealing ring grooves 1620 are provided at positions of the top of the first valve body 161 where the two first fuel return passages 1605 are provided, and the first valve body 161 is further provided with a first air inlet passage 1608 communicating with the axial central hole. A second sealing ring groove 1622 is provided in the circumferential direction of the first air inlet passage 1608, and a second sealing ring 1631 for improving the sealing effect is mounted in the second sealing ring groove 1622. The first valve core 162 is disposed in the axial central hole of the first valve body 161, the top of the first valve core 162 is provided with a first conical seal surface 1611, and the upper part of the first conical seal surface 1611 is further provided with an external thread 1612. The lower spring seat 164 passes through the external thread 1612 and then is sleeved on the first conical seal surface 1611. The lower part of the lower spring seat 164 is provided with a second conical seal surface 1613 mating with the first conical seal surface 1611. The nut 1614 is sleeved on the external thread 1612 and presses against the lower spring seat 164. The first pressure regulating spring 169 is sleeved on the nut 1614 and fixed to the lower spring seat 164. The gland 167, which is disposed on the top of the first valve body 161, is provided with two second fuel return passages 1606 communicating with the two first fuel return passages 1605 and is further provided, in its middle part, with a threaded hole 1603 in which the fuel return joint 168 is mounted. The orifice of the threaded hole 1603 is provided with a first flat sealing surface 1615. The gland 167 is provided, at its bottom surface, with a second air inlet passage 1629 communicating with the first air inlet passage 1608 and provided, in its circumferential direction, with an air inlet port 1630 communicating with the second air inlet passage 1629. The fuel return joint 168 is provided in the threaded hole 1603 and provided with a second flat sealing surface 1616 mating with the first flat sealing surface 1615. It should be understood by those skilled in the art that another number of first fuel return passages 1605 may be provided as required, in the technical solution of the present disclosure, and the numbers of the first sealing ring grooves 1620 and the second fuel return passages 1606 may be adjusted correspondingly according to the number of the first fuel return passages.

[0088] As shown in FIGS. 28 and 29, when compressed air is introduced into the air inlet port 1630 upon the low-speed engine is just started, the first valve core 162 is moved upward, so that there is a maximum distance H2 between the bottom end surface of the first valve body 161 and the bottom end surface of the first valve core 162. At this time, no fuel is returned into the first fuel return passages 1605. When the fuel feed pressure at the bottom of the first valve core 162 is lower than the pressure for opening the first pressure regulating spring 169, the first valve core 162 is moved downward under the action of spring force from the first pressure regulating spring 169. At this time, the distance H2 between the bottom end surface of the first valve body 161 and the bottom end surface of the first valve core 162 is 0. Return of fuel into the first fuel return passages 1605 is started, thereby realizing the circulation flow of fuel at low pressure. The high-pressure heavy fuel in the common rail pipe 12 flows back to the fuel tank through the first fuel return passages 1605, the second fuel return passages 1606, and the third fuel return passage 1607. It can be understood that the above-mentioned distance H2 between the bottom end surface of the first valve body 161 and the bottom end surface of the first valve core 162 corresponds to the lift of the first valve core 162.

[0089] Optionally, referring to FIG. 28, optionally, the axial central hole of the first valve body 161 comprises: a first central hole 1601 mating with the outer circumferential surface of the first valve core 162. A third sealing ring groove 163, in which a third sealing ring 1624 is mounted, is provided in the wall of the first central hole 1601. In the present disclosure, it can be determined based on the above-mentioned principle that the first valve core 162 is moved down to the bottom dead center position under the action of the first pressure regulating spring 169 before air is introduced into the circulation valve component 16. At this time, the first valve core 162 and the first valve body 161 are not sealed to each other. At the initial moment at which the low-speed engine is just started, the high-pressure fuel pump 1 is rotating at a low speed, and the high-pressure fuel is pumped from the high-pressure fuel pump 1 into the common rail pipe 12 at a small flow rate. Because no conical seal is formed between the first valve core 162 and the first valve body 161, pressure cannot be quickly built up in the fuel inlet conduit 1201 of the common rail pipe 12. Therefore, it is necessary to form a conical seal between the first valve core 162 and the first valve body 161, in other words, it is necessary to enable a forward movement of the first valve core 162, in order to quickly build up the pressure in the common rail pipe 12. At this time, it is necessary to apply an external forward force to the first valve core 162. The "forward movement" described in this embodiment refers to an upward movement in the vertical direction according to the direction shown in the drawings, and vice versa. In this example, compressed air is used as a power source of the external forward force. Therefore, it is necessary to design a closed volume chamber (first cavity) in the first valve body 161 for accommodating compressed air. As shown in FIGS. 28 and 29, the first valve core 162 is provided with a first conical seal surface 1611 and an external thread 1612. The lower spring seat 164 is pressed against the first conical seal surface 1611 by tightening the nut 1614 onto the external thread 1612. The first pressure regulating spring 169 is mounted between the nut 1614 and the lower spring seat 164. A first cavity for accommodating compressed air is formed between the lower spring seat 164 and the first central hole 1601 of the first valve body 161. When the pressure of compressed air is large enough to be greater than the preload force on the first pressure regulating spring 169, the first valve core 162 is moved upward. At this time, the distance H2 between the bottom end surface of the first valve body 161 and the bottom end surface of the first valve core 162 has the maximum value, and no fuel is returned into the first fuel return passage 1605. When the fuel pressure at the lower part of the first valve core 162 is large enough, the introduction of compressed air is stopped. At this time, the distance H2 between the bottom end surfaces of the first valve core 162 and the first valve body 161 is maintained mainly by the fuel feed pressure.

[0090] Here, a third sealing ring 1624 for preventing leakage of compressed air into the first fuel return passages 1605 is mounted in the third sealing ring groove 163 provided in the first central hole 1601 mating with the outer circumferential surface of the first valve core 162, in order to enable the first cavity to form a closed volume chamber. Similarly, a fourth sealing ring groove 1625 in which a fourth sealing ring 1627 is mounted is provided in a wall of the lower spring seat 164 that mates with the axial central hole. Furthermore, the lower part of the lower spring seat 164 is provided with a second conical seal surface 1613. The second conical seal surface 1613 mates with the first conical seal surface 1611, so as to prevent leakage of compressed air.

[0091] In the present disclosure, compressed air is introduced into the first cavity through the air inlet port 1630, the second air inlet passage 1629, and the first air inlet passage 1608. A first sealing ring 1628 is disposed at the junction between the first air inlet passage 1608 and the second air inlet passage 1629 in order to prevent air leakage.

[0092] As shown in FIGS. 28 and 29, after compressed air is introduced into the first cavity, the lower spring seat 164 is moved forward. The lower spring seat 164 is pressed against the first valve core 162 by means of the nut 1614, so that the lower spring seat 164 is integrated with the first valve core 162, in order to ensure a forward movement of the lower spring seat 164 together with the first valve core 162. In addition, the diameter of the fourth central hole 1618 in the lower spring seat 164 must be larger than the outer diameter of the external thread 1612, in order to prevent damage of the second conical seal surface 1613 of the lower spring seat 164 caused by its contact with the external thread 1612, when the lower spring seat 164 is passing through the outer thread 1612 of the first valve core 162.

[0093] An air outlet passage 1623 communicates with a second cavity formed between the third central hole 1617 in the lower spring seat 164 and the second central hole 1602 in the first valve body 161 for discharging air from the second cavity, in order to reduce the movement resistance applied to the first valve core 162 during its forward movement.

[0094] Furthermore, considering that a mating portion 1626 of the lower spring seat 164 is equipped with a fourth sealing ring 1627, a guide portion 1619 having a full angle generally set at 30° to 40° is provided at the opening of the second central hole 1602, so that the lower spring seat 164 is conveniently mounted in the second central hole 1602 of the first valve body 161 without damaging the fourth sealing ring 1627. The full angle of the guide portion 1619 described above is defined as an angle between the two edges of the guide portion 1619, and the half angle is an angle between one edge of the guide portion 1619 and its central line.

[0095] As shown in FIG. 28, optionally, a first sealing ring groove 1620 in which a first sealing ring 1628 is mounted is provided at the junction between the first fuel return passage 1605 and the second fuel return passage 1606 to prevent leakage of low-pressure fuel. Similarly, the fuel return joint 168 is provided with a second flat sealing surface 1616 mating with the first flat sealing surface 1615 of the gland 167, in order to prevent leakage of fuel between the fuel return joint 168 and the gland 167.

[0096] A third cavity is formed between the threaded hole 1603 provided in the gland 167 and the bottom surface of the fuel return joint 168. The third cavity communicates with the third fuel return passage 1607 provided in the fuel return joint 168 (the third fuel return passage 1607 communicates to the fuel tank via a pipeline) and with the second fuel return passages 1606. The third cavity has a flow area larger than that of the second fuel return passage 1606, so that the fuel can be discharged in time.

[0097] When the low-speed engine is stopped and when the fuel pressure at the bottom of the first valve core 162 (i.e., the fuel pressure in the fuel inlet conduit 1201 of the common rail pipe 12) decreases to the pressure for opening the first pressure regulating spring 169, the first valve core 162 is moved in a reverse direction. At this time, the distance H2 is 0. Since the distance H1 between the lower end surface of the lower spring seat and the bottom of the second central hole is set to be greater than H2, the lower spring seat 164 will not collide with the second central hole 1602. At this time, the first cavity has a certain volume. In this way, when compressed air is introduced into the first cavity during starting of the low-speed engine, the first valve core 162 can be moved forward quickly, whereby the fuel pressure at the bottom of the first valve body 162 can be built up quickly.

[0098] Here, the circulation valve component 16 is used in a high-pressure common rail system for a low-speed engine, in which a combustible medium is often inferior fuel. In order to ensure the service life and reliable function of the circulation valve, the first valve core 162 is made of a high-speed tool steel material, the mating section of the first valve core 162 is plated with DLC, and the first valve body 161 is made of high-strength structural steel subjected to nitriding treatment. The high temperature resistance and corrosion resistance of the circulation valve are ensured by using such materials and heat treatment method.

[0099] The circulation valve component 16 for the common rail pipe 12 described above is mounted onto the high-pressure common rail pipe 12. The main principle involves achieving the upward pushing and downward pushing of the first valve core 162 by using compressed air and the spring force from the first pressure regulating spring 169, respectively. When compressed air is introduced into the air inlet port 1630, the first valve core 162 and the lower spring seat 164 are connected as a whole by means of the nut 1614 under the action of air pressure so as to move upward together. At this time, the distance H2 has the maximum value, no fuel is returned into the first fuel return passages 1605, and the pressure of the fuel at the bottom of the first valve core 162 is built up during the movement of the high-pressure fuel pump. When the fuel pressure reaches a certain level, the compressed air is cut off. At this time, the distance H2 is maintained at the maximum value mainly by the fuel feed pressure exerted on the common rail pipe 12. When the low-speed engine is stopped and when the pressure in the common rail pipe 12 decreases below the spring force from the first pressure regulating spring 169, the first valve core 162 is pushed in a reverse direction. At this time, the low-pressure fuel enters the first fuel return passages 1605 and then flows back into the fuel tank sequentially from the second fuel return passages 1606 and the third fuel return passage 1607. Thus, the fuel circulation effect is achieved, solidification of the fuel is avoided, and corrosion of the parts can be prevented.

[0100] Specifically, referring to FIGS. 32 and 34, optionally, the pressure limiting valve component 18 in this embodiment comprises: a third valve body 181, a third valve core 182, a second O-ring seal 183, a second bolt 184, a third pressure regulating spring 185, a fuel pipe joint 186, a pressure regulating gasket 187 as parts. Here, the third valve core 182 is mounted in a fourth-stage hole 18105 in the third valve body 181. A second tapered portion 18202 of the third valve core 182 is pressed against a sealing seat surface 18103 (referring to a wall of a second-stage hole 18102) of the third valve body 181. The third valve core 182 is pressed by the fuel pipe joint 186 and by the third pressure regulating spring 185. The lower part of the fuel pipe joint 186 is provided with a fifth sealing ring groove 18604, in which a second O-ring seal 183 is mounted to improve the sealing effect and achieve the effect of preventing fuel leakage. The middle part of the fuel pipe joint 186 is provided as a multi-stage counterbore structure, including a first counterbore 18601, a second counterbore 18602, and a fuel outlet hole 18603. The outer circumferential surface of the upper part of the fuel pipe joint 186 is provided with a thread. The fuel pipe joint 186 is mounted onto the third valve body 181 by means of the second bolts 184. The pressure regulating gasket 187 and the third pressure regulating spring 185 are mounted in the second counterbore 18602 in the middle part of the fuel pipe joint 186.

[0101] Optionally, the central hole of the third valve body 181 is designed as a multi-stage hole structure, including a first-stage hole 18101, a second-stage hole 18102, a third-stage hole 18104, and a fourth-stage hole 18105, wherein the second-stage hole 18102 is a small throttling hole, and the diameter of the first-stage hole 18101 is larger than the diameter of the second-stage hole 18102, so that the depth of the throttling hole, i.e., the second-stage hole 18102, can be decreased to reduce the overall machining difficulty. An angle of 59° is provided between the second-stage hole 18102 and the third-stage hole 18104 of the third valve body 181 (namely, the sealing seat surface 18103 of the third valve body is provided at an angle of 59°) to ensure a linear seal between the sealing seat surface of the third valve body and the second tapered portion 18202 of the third valve core 182 within a deviation range of 1° to achieve good sealing performance. The diameter of the third-stage hole 18104 of the third valve body 181 is set to be larger than both the diameter of the second-stage hole 18102 and the diameter of the fourth-stage hole 18105 for storing fuel pressure. Optionally, the diameter φ of the fourth-stage hole 18105 of the third valve body 181 is set at 5 mm. If the diameter of the fourth-stage hole 18105 is too small, it is machined with great difficulty and its accuracy cannot be guaranteed in use. If the diameter of the fourth-stage hole 18105 is too large, the depth of the fourth-stage hole 18105 will be increased due to its length required to mate with the third valve core 182, which leads to increased difficulty in machining the sealing seat surface 18103. As a result, its accuracy cannot be guaranteed, and a difficult measurement problem will also be caused. The third valve body 181 is made of a high-strength structural steel material subjected to nitriding treatment. The high-pressure resistance of the third valve body 181 can be guaranteed by selecting a high-strength material. The nitriding treatment can ensure that the third valve body 181 can work in an environment of a low-speed engine using a heavy fuel oil and have corrosion resistance.

[0102] Referring to FIG. 34, optionally, the head of the third valve core 182 is provided as a structure with two tapered portions, including a first tapered portion 18201 and a second tapered portion 18202, wherein the first tapered portion 18201 has a taper angle set to an obtuse angle of 120°. On the one hand, this allows for increased reliability of the third valve core 182. On the other hand, this allows for an increased flow area of the head of the third valve core 182 and reduced occurrence of cavitation. The second tapered portion 18202 of the third valve core 182 has a taper angle set to an acute angle of 60°. The taper angle of the second tapered portion 18202 of the third valve core 182 is generally set at two angles, namely, 60° or 90°, in order to ensure good sealing performance. Considering that the third valve core which is a needle valve has a shorter lift at 60° than at 90° under the same flow area, there is less difficulty in designing the third pressure regulating spring 185.

[0103] Referring to FIG. 34, optionally, two symmetrical second flat portions 18203 are formed by milling the third outer cylindrical portion 18204 of the middle part of the third valve core 182. The second flat portion 18203 has a slightly larger flow area than that of the second-stage hole 18102 in the third valve body 181. A second gap allowing the fuel to passthrough is formed between the second flat portion 18203 of the third valve core 182 and the fourth-stage hole 18105. The second gap has a fuel flowable area larger than the fuel flow area in the second-stage hole 18102. As shown in FIG. 34, a second smaller outer cylindrical portion 18206 configured for positioning of the third pressure regulating spring 185 is provided at the upper part of the third valve core 182, and the third pressure regulating spring 185 is sleeved around the second smaller outer cylindrical portion 18206. It should be explained that, in the present disclosure, the term "flow area" generally refers to a fuel passing area of a corresponding gap or hole.

[0104] The third valve core 182 is made of a high-speed tool steel material. The third valve core 182 has a third outer cylindrical portion 18204 mating with the fourth-stage hole 18105 of the third valve body 1, which is correspondingly plated with a DLC layer to ensure the required strength of the third valve core 182 and the corrosion resistance in a heavy fuel oil environment. The DLC plating can also make the third valve core 182 more wear-resistant.

[0105] Optionally, referring to FIG. 32, a limited distance h1 is provided between the third valve core 182 and the third valve body 181 to ensure a movement of the third valve core 182 within a certain range. Otherwise, the third pressure regulating spring 185 will be compressed to a solid state, so that the third pressure regulating spring 185 cannot be restored to its original position. Here, the compression of the third pressure regulating spring 185 to a solid state refers to compression of the spring to a state where the coils touch each other.

[0106] Optionally, referring to FIG. 32, an overlap region h is provided between the third valve core 182 and the third valve body 181. While the third valve core 182 is being closed from an opened state, there is a moment at which an overlap region h occurs between the third valve core 182 and the third valve body 181, at which the fuel cannot flow out through the second flat portions 18203, and pressure is formed in the third-stage hole 18104 such that a force opposite to the movement direction (i.e., the opening or closing direction) is exerted on the third valve core 182. This reduces an impact force applied to the sealing seat surface 18103 of the third valve body 181 from the third valve core 182, thereby increasing the service life of the third valve body 181 and the third valve core 182.

[0107] The pressure limiting valve component 18 works based on the following principle. The pressure limiting valve component 18 is used in the high-pressure common rail system described above, wherein the high-pressure common rail system may be a high-pressure common rail fuel injection system for a marine low-speed diesel engine, and the pressure limiting valve component 18 can work at a high pressure of 150 MPa and at a high temperature of 200 °C. As a safety protection part of the common rail system, the pressure limiting valve component 18 is usually in an inoperative state and thus is also called a pressure safety valve. When the rail pressure in the fuel inlet conduit 1201 of the common rail pipe 12 is controlled abnormally and thus exceeds the pressure P_L for opening the pressure limiting valve component 18, the pressure limiting valve component 18 is opened to relieve pressure. Specifically, as high-pressure fuel supplied into the fuel inlet conduit 1201 of the common rail pipe 12 flows into the first-stage hole 18101 and then into the second-stage hole 18102, the third valve core 182 is pushed by the fuel pressure to move upward so as to release the seal between the third valve core 182 and the third valve body 181. The third valve core 182 is moved upward under the action of the fuel pressure, until the tapered surface seal between the second tapered portion 18202 and the sealing seat surface 18103 is released. The high-pressure fuel in the fuel inlet conduit 1201 of the common rail pipe 12 partially passes through the first-stage hole 18101, the second-stage hole 18102, the third-stage hole 18104, the fourth-stage hole 18105, the recesses 18205, the first counterbore 18601, and the second counterbore 18602, and finally enters the fuel return pipe from the fuel outlet hole 18603 and ultimately flows back to the fuel tank, so that the pressure in the fuel inlet conduit 1201 of the common rail pipe 12 is reduced. After the third valve core 182 is opened, the pressure in the fuel inlet conduit 1201 of the common rail pipe 12 is reduced, so that the amount of fuel fed into the fuel system (i.e., the amount of fuel pumped from the high-pressure fuel pump 1 into the common rail pipe 12) and the amount of fuel discharged through the pressure limiting valve component 18 gradually reach a stable state, and the system pressure gradually approaches a stable pressure Ps. It is ensured that the diesel engine works in a failure mode at the stable pressure Ps, so that the ship limps back to port. The pressure limiting valve component 18 is closed only when the pressure in the common rail pipe 12 decreases to a certain level (lower than the pressure P_L for opening the pressure limiting valve component 18).

[0108] In this embodiment, a flow limiting valve component 13 is mounted between the common rail pipe 12 and a corresponding electronic control fuel injector 15 and can cut off the fuel supply into the electronic control fuel injector 15 to prevent problems such as fire and explosion and personnel safety issues when fuel leaks from the high-pressure fuel pipe, or fuel leaks or is abnormally injected from the electronic control fuel injector 15.

[0109] Optionally, as shown in FIGS. 30 and 31, the flow limiting valve component 13 comprises: a valve seat 131 having a larger outer cylindrical portion provided with a first flat portion 13104 for returning fuel, provided in its middle part with a first fuel inlet hole 13101 communicating with the fuel inlet conduit 1201 of the common rail pipe 12, and provided with a fifth conical seal surface 13102 at its bottom; a second valve body 132 having a smaller end surface 13201 fixed to a larger end surface 13105 of the valve seat 131, the second valve body 132 being provided with a second axial through hole 13209 in the axial direction which extends through the two upper and lower end surfaces of the second valve body 132 and in which the valve seat 131 is partially press-fitted in a press fit manner; and a second valve core 133 fixed to the upper end surface of the first smaller outer cylindrical portion 13106 of the valve seat 131 through the second axial through hole 13209. The second valve core 133 is provided with an axial blind hole 13302 communicating with the first fuel inlet hole 13101 and four transverse flow restricting orifices 13304 communicating with the axial blind hole 13302. The upper part of the second valve core 133 is provided with a third conical seal surface 13305 and a fourth conical seal surface 13306. The second valve body 132 has four fourth fuel return passages 13210 communicating with the first flat portion 13104. The second valve body 132 is provided with a first sealing seat surface 13206 mating with the third conical seal surface 13305 and is provided with a fourth hole 13207 and a second sealing seat surface 13208 in this order at a position close to the first sealing seat surface 13206. The second valve body 132 comprises a first mating portion 13203 mating with the first smaller outer cylindrical portion 13106 of the valve seat 131 and a second mating portion 13204 mating with the first outer cylindrical portion 13301 of the second valve core 133. A cavity communicating the transverse flow restricting orifices 13304 with the axial blind hole 13302 is formed between the second mating portion 13204 and the second valve core 133. A second pressure regulating spring 135 is inserted in the cavity and fixed to the upper end surface of the first outer cylindrical portion 13301 of the second valve core 133.

[0110] Optionally, as shown in FIG. 33, in this embodiment, the second axial through hole 13209 comprises a first hole 13212, a second hole 13213, a third hole 13205, a fourth hole 13207, a fifth hole 13214, and a sixth hole 13215 that are connected to each other in this order from top to bottom. The valve seat 131 is partially press-fitted into the first hole 13212. The second valve core 133 is assembled in the second hole 13213, and a cavity is formed between the upper part of the second valve core 133 and the upper part of the second hole 13213. The third conical seal surface 13305 is formed at the wall of the third hole 13205. A second sealing seat surface 13208 for being sealed to the fuel inlet end of the fuel pipe of the electronic control fuel injector 15 is formed at the wall of the fifth hole 13214. The diameter of the sixth hole 13215 is larger than the diameter of each of the first hole 13212, the second hole 13213, the third hole 13205, the fourth hole 13207, and the fifth hole 13214.

[0111] The fourth hole 13207 is used as a fuel outlet hole and is configured such that the fuel in the first fuel inlet hole 13101 flows out into the fourth hole 13207 and further flows out therefrom when the first sealing seat surface 13206 and the third conical seal surface 13305 are unsealed from each other.

[0112] When the fuel feed pressure in the transverse flow restricting orifice 13304 on the side of the axial blind hole 13302 exceeds a certain value of the pressure in the cavity between the second mating portion 13204 and the second valve core 133, the second valve core 133 is moved upward under the action of the fuel feed pressure, so that the third conical seal surface 13305 is brought into contact with the first sealing seat surface 13206 and a tapered surface seal is formed therebetween to block flow of fuel into the fuel outlet hole.

[0113] Optionally, as shown in FIG. 30, the second valve core 133 has a third conical seal surface 13305, and the second valve body 132 has a first sealing seat surface 13206 that can form a tapered surface seal with the third conical seal surface 13305. When the fuel feed pressure in the transverse flow restricting orifice 13304 on the side of the axial blind hole 13302 exceeds a certain value of the cavity pressure, a tapered surface sealing relationship is formed between the third conical seal surface 13305 and the first sealing seat surface 13206 to block flow of fuel into the fuel outlet hole 13207. On the contrary, when the fuel feed pressure in the transverse flow restricting orifice 13304 on the side of the axial blind hole 13302 does not exceed a certain value of the cavity pressure, the tapered surface seal between the third conical seal surface 13305 and the first sealing seat surface 13206 is released. With regard to the fourth conical seal surface 13306, during the upward movement of the second valve core 133 under the pressure from the fuel fed therein, the fourth conical seal surface 13306 is first brought into contact with the first sealing seat surface 13206, but there is a gap between the third sealing seat surface 13305 and the first sealing seat surface 13206. The gap may be filled with a certain amount of fuel to cushion the rapid movement of the second valve core 133 to prevent a large impact on the second valve body 132 due to an excessively rapid movement of the second valve core 133.

[0114] Optionally, as shown in FIG. 25, the flow limiting valve component 13 is mounted onto the common rail pipe 12 by means of a corresponding flow limiting valve mounting seat 125, and the flow limiting valve mounting seat 125 is fixed to the common rail pipe 12 by screws. Screw mounting holes 13211 are designed in the second valve body 132. The second valve body 132 of the flow limiting valve component 13 is fixed to the flow limiting valve mounting seat 125 by means of the screws passing through the screw mounting holes 13211. The flow limiting valve mounting seat 125 is provided with a fuel hole communicating with the first fuel inlet hole 13101, through which the high-pressure fuel in the fuel inlet conduit 1201 of the high-pressure common rail pipe 12 can enter the first fuel inlet hole 13101. The valve seat 131 is designed with a fifth conical seal surface 13102 for sealing the high-pressure fuel between the flow limiting valve component 13 and the high-pressure common rail pipe 12. The second valve body 132 is designed with a second sealing seat surface 13208 for allowing tapered surface sealing and connection between the flow limiting valve component 13 and the fuel pipe of the electronic control fuel injector 15.

[0115] Optionally, as shown in FIG. 30, the outer surface of the second valve body 132 is provided with a sixth sealing ring groove in which a first O-ring seal 136 is mounted, wherein the first O-ring seal 136 is provided in the outer surface of the second valve body 132 for the purpose of enhancing the sealing performance between the second valve body 132 and an external component when the second valve body 132 is assembled into the external component.

[0116] Optionally, as shown in FIG. 30, the first valve seat 131 is provided with a first flat portion 13104. The first flat portion 13104 communicates with the four fourth fuel return passages 13210 for collecting fuel leaking from the fifth conical seal surface 13102, from between the larger end surface 13105 of the valve seat 131 and the smaller end surface 13201 of the second valve body 132, and from the second sealing seat surface 13208 of the second valve body 132.

[0117] Optionally, as shown in FIG. 30, in order to ensure the sealing between the smaller end surface 13201 of the second valve body 132 and the larger end surface 13105 of the valve seat 131, a large bevel 13202 is provided for reducing the contact area between the larger end surface 13105 and the smaller end surface 13201 to enhance the sealing therebetween.

[0118] Optionally, the first smaller outer cylindrical portion 13106 of the valve seat 131 and the first mating portion 13203 of the second valve body 132 have the same diameter and are mated with each other in a transitional manner, in order to facilitate the mounting of the flow limiting valve component 13 onto the high-pressure common rail pipe 12 as a whole. A sunk groove 13103 is provided at the intersection of the larger end surface 13105 and the first smaller outer cylindrical portion 13106.

[0119] Optionally, as shown in FIG. 30, four transverse flow restricting orifices 13304 are provided in the second valve core 133 in order to achieve the function of the flow limiting valve. The four flow restricting orifices have an area smaller than the area of the axial blind hole 13302, the area of the fourth hole (the fuel outlet hole) 13207, and the flow area formed between the fourth conical seal surface 13306 and the first sealing seat surface 13206.

[0120] Optionally, as shown in FIG. 30, the fourth conical seal surface 13306 of the second valve core 133 is provided at the tail of the second valve core 133 adjacent to the third conical seal surface 13305, and the fourth conical seal surface 13306 has an angle larger than the angle of the third conical seal surface 13305, so that the flow area between the fourth conical seal surface 13306 and the first sealing seat surface 13206 can be increased. It should be noted that the angle of each of the above tapered surfaces refers to an angle between the tapered surface and a generatrix in the same cross-section.

[0121] The gap between the first outer cylindrical portion 13301 and the second mating portion 13204 is formed as small as possible in order to prevent leakage of fuel from the cavity formed between the second valve core 133 and the second mating portion 13204 into the first mating portion 13203. However, the gap should not be too small, otherwise the operation of the second valve core 133 is hindered, and the flow limiting valve malfunctions. Optionally, the gap between the first outer cylindrical portion 13301 and the second mating portion 13204 is between 0.02 mm and 0.04 mm.

[0122] The flow limiting valve component 13 is used in a high-pressure common rail system for a low-speed engine, in which a combustible medium is often inferior fuel. In order to ensure the service life and reliable function of the flow limiting valve, the second valve core 133 is made of a high-speed tool steel material and plated with DLC at its sections mating with the second valve body 132 (the first outer cylindrical portion 13301, the second outer cylindrical portion 13303, the third conical seal surface 13305, and the fourth conical seal surface 13306), and the second valve body 132 is made of high-strength structural steel subjected to nitriding treatment. The high temperature resistance and corrosion resistance of the flow limiting valve are ensured by using such materials and heat treatment method.

[0123] In this embodiment, the second valve core 133 is pushed forward by using the fuel feed pressure from fuel and the spring force from the second pressure regulating spring 135. When the second valve core 133 is pushed forward, the second valve core 133 seals the fourth hole 13207 serving as a fuel outlet hole to block entry of fuel thereinto, so that the fuel is cut off to avoid excessive fuel injection from the electronic control fuel injector 15 or fuel leakage from the high-pressure fuel pipe, thereby preventing damage to the diesel engine or personal injury and also preventing pollution of the environment.

[0124] Only one or more of some embodiments of the present disclosure are described in the foregoing embodiments, but it will be appreciated by those of ordinary skill in the art that the present disclosure may be implemented in many other forms without departing from its spirit and scope. Therefore, the examples and embodiments shown are considered to be illustrative rather than limitative, and the present disclosure may encompass various modifications and alternatives without departing from the spirit and scope of the present disclosure as defined by the appended claims.

Claims

1. A high-pressure common rail system for a low-speed engine with multiple safety protection functions, **characterized by** comprising:

   an electronic control unit ECU;
   an electronic control high-pressure fuel pump (1), provided with an electronic control proportional valve (2), wherein the electronic control proportional valve (2) is configured to adjust a proportion of feed of a low-pressure heavy fuel from a fuel tank of the low-speed engine into the electronic control high-pressure fuel pump (1) according to a first instruction from the ECU;
   a first distribution block (6) connected to the electronic control high-pressure fuel pump (1) via a first high-pressure fuel pipe (3);
   a second distribution block (9) connected to the first distribution block (6) via a second high-pressure fuel pipe (8); and
   a common rail pipe (12) connected to the second distribution block (9) via a third high-pressure fuel pipe (10),
   wherein a sensor (17) configured to detect a fuel pressure of the high-pressure heavy fuel in the common rail pipe (12) is mounted onto the common rail pipe (12), and the sensor (17) is connected to the ECU;
   a plurality of flow limiting valve components (13) are mounted onto the common rail pipe (12), and each of the flow limiting valve components (13) is connected to one of electronic control fuel injectors (15) via one of fourth high-pressure fuel pipes (14); and each of the flow limiting valve components (13) is configured to be closed when a fuel pressure difference between corresponding one of the fourth high-pressure fuel pipes (14) and the common rail pipe (12) exceeds a set pressure difference;
   a pressure limiting valve component (18) is further mounted onto the common rail pipe (12), and the pressure limiting valve component (18) is configured to be opened when a fuel pressure in the common rail pipe (12) exceeds a first set pressure value, so that the fuel pressure in the common rail pipe (12) is stabilized at a target pressure value;
   the first distribution block (6) is equipped with a shut-off valve component (5) and a safety valve component (7), wherein the shut-off valve component (5) is configured to perform a pressure relief process according to a second instruction from the ECU; and the safety valve component (7) is configured to be opened when the shut-off valve component (5) and the pressure limiting valve component (18) malfunction and a fuel pressure in the common rail pipe (12) exceeds a second set pressure value; and
   circulation valve components (16) are further mounted onto the common rail pipe (12) and the electronic control fuel injectors (15), and each of the circulation valve components (16) is configured to be opened when the low-speed engine is stopped, so that a circulation loop is formed between a fuel tank of the low-speed engine and the common rail pipe (12), and between the fuel tank of the low-speed engine and each of the electronic control fuel injectors (15).

2. The system according to claim 1, wherein the electronic control high-pressure fuel pump (1) comprises:

   a pump body (101), provided with a central hole along an axial direction of the pump body (101);
   a pump cover (102), mounted onto an upper end surface of the pump body (101); and
   a fuel inlet and outlet valve component (103), a plunger and barrel assembly (104), a plunger spring (105), a lower spring seat component (106), and a guide piston component (107), each being assembled in the central hole of the pump body (101),
   wherein the electronic control proportional valve (2) is assembled on a side of the pump body (101);
   the fuel inlet and outlet valve component (103) comprises a fuel inlet valve component (1031) and a fuel outlet valve component (1032),
   wherein the fuel inlet valve component (1031) comprises a fuel inlet valve seat (10311), a fuel inlet valve (10312), and a fuel inlet valve spring (10313),
   wherein the fuel inlet valve (10312) is mounted in a central hole of the fuel inlet valve seat (10311); the fuel inlet valve spring (10313) is positionally limited between the fuel inlet valve (10312) and a wall of the central hole of the fuel inlet valve seat (10311); and a conical seal is formed between the fuel inlet valve (10312) and the fuel inlet valve seat (10311) by being pressed by the fuel inlet valve spring (10313); and
   the fuel outlet valve component (1032) comprises a fuel outlet valve seat (10321), a fuel outlet valve (10322), a fuel outlet valve spring (10323), and a fuel outlet valve spring seat (10324),

wherein the fuel outlet valve spring seat (10324) is mounted at an upper end of the fuel outlet valve seat (10321); the fuel outlet valve (10322) is mounted in a central hole of the fuel outlet valve seat (10321); the fuel outlet valve spring (10323) is positionally limited between the fuel outlet valve (10322) and the fuel outlet valve spring seat (10324); and a conical seal is formed between the fuel outlet valve (10322) and the fuel outlet valve seat (10321) by being pressed by the fuel outlet valve spring (10323),
a high-pressure fuel outlet chamber (1033) is formed between the fuel outlet valve seat (10321) and the fuel inlet valve seat (10311);
a high-pressure fuel chamber (1041) is formed in the plunger and barrel assembly (104), wherein the high-pressure fuel chamber (1041) communicates with the high-pressure fuel outlet chamber (1033) via a fuel hole in the fuel inlet valve seat (10311);
the electronic control proportional valve (2) communicates with the fuel inlet hole of the fuel inlet valve seat (10311) via a first fuel hole in the pump body (101), and the fuel inlet hole is communicated with or disconnected from the high-pressure fuel chamber (1041); and
the electronic control proportional valve (2) is provided with a cooling circulation fuel passage, and cooling fuel from a cooling fuel passage of the pump body (101) is injected into the cooling circulation fuel passage and then flows back into the cooling fuel passage of the pump body (101).

3. The system according to claim 2, wherein the plunger and barrel assembly (104) comprises:

a plunger barrel (1042), disposed at a lower end of the fuel inlet valve seat (10311); and
a plunger (1043), slidably inserted into a central hole of the plunger barrel (1042), wherein the high-pressure fuel chamber (1041) is formed by the plunger barrel (1042), the plunger (1043) and the fuel inlet valve seat (10311) together, wherein
an inner wall of the plunger barrel (1042) is provided with a first annular groove (10421) and a second annular groove (10422);
the pump body (101) is provided with a mixed oil outlet passage and a lubricating oil supply passage (1012), wherein the mixed oil outlet passage communicates with the first annular groove (10421) via a mixed oil passage (10423) in the plunger barrel (1042), and the lubricating oil supply passage (1012) communicates with the second annular groove (10422) via a lubricating oil passage (10424) in the plunger barrel (1042);
the first annular groove (10421) is located above the second annular groove (10422); and the lower spring seat component (106) is disposed under the plunger and barrel assembly (104),
wherein the lower spring seat component (106) comprises:

an outer spring seat (1061), of a boss-type structure as a whole with a central portion thicker and an outer portion thinner, wherein the outer spring seat (1061) has an upper end surface provided with a third counterbore (10611) with a concave spherical surface;
an upper sphere (1062), having lower part mounted in the third counterbore (10611), wherein a lower end surface of the upper sphere (1062) is provided with a convex spherical surface mating with the concave spherical surface; and
an inner spring seat (1063), sleeved on an upper part of the upper sphere (1062), wherein the inner spring seat (1063) has a first axial through hole (10631) penetrating its upper and lower end surfaces,
wherein a lower cylindrical head (10431) of the plunger (1043) is positionally limited in the first axial through hole (10631), and the lower cylindrical head (10431) of the plunger (1043) has a lower end surface abutting against an upper end surface of the upper sphere (1062).

4. The system according to claim 3, wherein

a spherical hole (10612) is provided in a center of the third counterbore (10611), a third annular groove (10613) is provided in a lower end surface of the outer spring seat (1061), and the spherical hole (10612) communicates with the third annular groove (10613) via a lubricating oil inlet conduit (10614);
an outer surface of the outer spring seat (1061) is formed as a tapered surface, which is provided with a lubricating oil outlet passage (10615), wherein the lubricating oil outlet passage (10615) communicates with the lower end surface of the outer spring seat (1061); and the lubricating oil outlet passage (10615) is provided obliquely;
a circumferential annular groove (10621) is arranged in a circumferential direction of the upper sphere (1062);
a positioning pin (1064) is mounted in the circumferential annular groove (10621) through a positioning pin hole (10616) of the outer spring seat (1061);
a spacing between upper and lower surfaces of the circumferential annular groove (10621) is

greater than a cylindrical diameter of a part of the positioning pin (1064) that is located in the circumferential annular groove (10621);
the first axial through hole (10631) provided inside the inner spring seat (1063) comprises:

a seventh hole (10632), an eighth hole (10633), and a ninth hole (10634) having diameters gradually increasing from top to bottom, wherein
a first guide hole (10635) with a gradually increasing diameter is provided between the eighth hole (10633) and the ninth hole (10634);
a second guide hole (10636) with a gradually increasing diameter is provided on a side of the ninth hole (10634) facing the upper sphere (1062), wherein

walls of the first guide hole (10635) and the second guide hole (10636) are formed as tapered guide surfaces (10637);
an upper part of the upper sphere (1062) is positioned partially in the ninth hole (10634) through the second guide hole (10636);
a gap greater than or equal to 1 mm is formed between the upper sphere (1062) and the ninth hole (10634); and
a gap greater than or equal to 1 mm is formed between the third counterbore (10611) and the upper sphere (1062).

5. The system according to any one of claims 2 to 4, wherein the electronic control high-pressure fuel pump (1) further comprises: an upper spring seat (109), sleeved on the plunger barrel (1042) and located at an upper end of the inner spring seat (1063);
the plunger spring (105) comprises: a first plunger spring (1051), compressively mounted between the upper spring seat (109) and the outer spring seat (1061); and a second plunger spring (1052), compressively mounted between the upper spring seat (109) and the inner spring seat (1063);
preferably, diameters of the concave spherical surface in the outer spring seat (1061) and the convex spherical surface of the upper sphere (1062) are each 20 to 100 times a diameter of the plunger (1043).

6. The system according to any one of claims 2 to 5, wherein the guide piston component (107) comprises:

a guide piston (1071), provided with a first mounting hole (10711) at a central position of its upper end surface and provided with a second mounting hole (10712) at its lower end surface, wherein the first mounting hole (10711) and the second mounting hole (10712) communicate with each other via a communicating hole (10713), and the lower spring seat component (106) is mounted in the first mounting hole (10711);
a roller component (1072), comprising a roller (10728) mounted in the second mounting hole (10712), a roller bushing (10722) interference-fitted in the roller (10728), and thrust bearings (10723) interference-fitted at both ends of the roller (10728) in an axial direction, wherein an annular groove (10724) is provided in the axial direction of the roller (10728), and an arc-shaped transitional connection is formed between a bottom of the annular groove (10724) and an end surface of the roller (10728) in the axial direction; and
a roller pin (1073), fitted in the roller bushing (10722) with a clearance therebetween, wherein a boss (10714) is disposed protruding from a wall of the second mounting hole (10712), and the boss (10714) is in contact with the thrust bearing (10723); and
a plurality of first radial oil grooves (10715) are uniformly arranged in the boss (10714) along its radial direction, and the first radial oil grooves (10715) are provided with respect to the thrust bearing (10723).

7. The system according to claim 6, wherein

an outer surface of the roller pin (1073) is provided as a cylindrical surface, which is provided with first waist-shaped grooves (10731) and second waist-shaped grooves (10732) at two positions, respectively, wherein the first waist-shaped grooves (10731) and the second waist-shaped grooves (10732) are provided in the middle positions of the roller pin (1073);
a small-angle wedge-shaped groove with an angle between 5° and 20° is formed between each of the first waist-shaped grooves (10731) located on an outer layer and an outer surface of the roller bushing (10722), and a second oil hole (10733) is provided in each of the second waist-shaped grooves (10732) located in an inner layer,
wherein two second oil holes (10733) at the two positions communicate with each other via a lubricating oil outlet passage, and the two second oil holes (10733) are provided at an angle of 70° to 120°;
an outer surface of the guide piston (1071) is provided as a cylindrical surface, which is provided with a plurality of partial circumferential oil grooves (10729) and circumferential oil grooves

(10725), a first axial oil groove (10716) and a vertical groove (10717), wherein the vertical groove (10717) is provided in each of the circumferential oil grooves (10725), and the vertical groove (10717) communicates with the partial circumferential oil grooves (10729) via the first axial oil groove (10716);

the cylindrical surface is further provided with an inclined hole (10718) having two ends communicating with one of the circumferential oil grooves (10725) and an inner wall of the second mounting hole (10712), respectively;

the cylindrical surface is further provided with a second axial oil groove (10719) communicating with one of the circumferential oil grooves (10725);

the cylindrical surface is further provided with a first straight hole (10720) and a second straight hole (10721) connected to each other, the first straight hole (10720) communicates with the first axial oil groove (10716), and the second straight hole (10721) communicates with the first mounting hole (10711); and

a lubricating oil inlet passage (10735) is provided in the outer cylindrical surface of the roller pin (1073), wherein the lubricating oil inlet passage (10735) is provided with respect to the inclined hole (10718), and the lubricating oil inlet passage (10735) communicates with the lubricating oil outlet passage.

8. The system according to claim 7, wherein the outer cylindrical surface of the roller pin (1073) is provided with a diamond-like carbon DLC coating; the roller bushing (10722) is made of a copper alloy; the thrust bearing (10723) is made of a copper alloy; forced lubrication and dynamic pressure lubrication are used between the roller pin (1073) and the roller bushing (10722); and forced lubrication and dynamic pressure lubrication are used between the thrust bearing (10723) and the boss (10714).

9. The system according to any one of claims 1 to 8, wherein

the common rail pipe (12) has a fuel inlet conduit (1201) and a fuel return conduit (1202) penetrating both ends thereof;

a fuel inlet end cap (121) is fixed at one end of the common rail pipe (12), wherein the fuel inlet end cap (121) is provided with a fuel inlet port communicating with the fuel inlet conduit (1201);

an end cover (122) is fixed at the other end of the common rail pipe (12), wherein the end cover (122) is provided with a fuel outlet port communicating with the fuel inlet conduit (1201), and the circulation valve component (16) is fixed to the end cover (122);

the pressure limiting valve component (18) and the plurality of flow limiting valve components (13) communicate with the fuel inlet conduit (1201), respectively, and the pressure limiting valve component (18) and the plurality of flow limiting valve components (13) communicate with the fuel return conduit (1202), respectively.

10. The system according to claim 9, wherein the circulation valve component (16) comprises:

a first valve body (161) fixed to the end cover (122), which has a lower end surface provided with a first central hole (1601) communicating with the fuel outlet port, and an upper end surface provided with a second central hole (1602), wherein the first central hole (1601) communicates with the second central hole (1602);

a first valve core (162), which is slidably inserted into the first central hole (1601) from the lower end surface of the first valve body (161) and partially located in the second central hole (1602);

a lower spring seat (164), which is sleeved on the part of the first valve core (162) that is located in the second central hole (1602), and which is fixedly connected to the first valve core (162), wherein a first cavity is formed between the lower spring seat (164) and a bottom of the second central hole (1602);

a gland (167), which is fixed to the upper end surface of the first valve body (161), wherein the gland (167) has an upper end surface provided with a threaded hole (1603);

a fuel return joint (168), which is partially fixed in the threaded hole (1603); and

a first pressure regulating spring (169), which is positionally limited between the lower spring seat (164) and the gland (167), wherein

the first valve body (161) is provided with a first fuel return passage (1605) communicating with the first central hole (1601), the gland (167) is provided with a second fuel return passage (1606) communicating with the first fuel return passage (1605), the fuel return joint (168) is provided with a third fuel return passage (1607) communicating with the second fuel return passage (1606), and a fuel circulation passage is formed by the first fuel return passage (1605), the second fuel return passage (1606) and the third fuel return passage (1607);

the first valve body (161) is provided with a first air inlet passage (1608) communicating with the first cavity, and the gland (167) is provided with a second air inlet passage (1629) communicating with the first air inlet passage (1608) and an air inlet port (1630) communicating with the second air inlet passage (1629); and

when a spring force from the first pressure regulating spring (169) is less than or equal to a pressure sum of a pressure of a gas introduced into the first cavity and a pressure of fuel fed into a fuel inlet end of the first central hole (1601), a conical seal is formed between the first valve core (162) and the first central hole (1601), and the conical seal is formed at a position below a position where the first fuel return passage (1605) is connected to the first central hole (1601).

11. The system according to claim 10, wherein a first conical seal surface (1611) and an external thread (1612) are provided at a top of a part of the first valve core (162) that is located in the second central hole (1602), wherein the external thread (1612) is located at an upper end of the first conical seal surface (1611); the lower spring seat (164) passes through the external thread (1612) and then is sleeved on the first conical seal surface (1611), and the lower spring seat (164) has a second conical seal surface (1613) that forms a conical seal with the first conical seal surface (1611); the lower spring seat (164) is pressed by a nut (1614) sleeved on a periphery of the external thread (1612); and the first pressure regulating spring (169) is sleeved on the nut (1614) and fixed to the lower spring seat (164).

12. The system according to claim 10 or 11, wherein the fuel return joint (168) is screwed fixedly in the threaded hole (1603); an upper end surface of the gland (167) is provided with a first flat sealing surface (1615) at an opening of the threaded hole (1603); and the fuel return joint (168) is provided with a second flat sealing surface (1616) that forms a planar seal with the first flat sealing surface (1615); and
a third cavity is formed between a lower end surface of the fuel return joint (168) and a bottom of the threaded hole (1603), wherein the third cavity communicates with the second fuel return passage (1606) and the third fuel return passage (1607), respectively, and a maximum flow area of the third cavity is greater than a maximum flow area of the second fuel return passage (1606).

13. The system according to any one of claims 10 to 12, wherein when the first valve core (162) moves upward to a top dead center position in the first valve body (161), a distance H2 between lower end surfaces of the first valve body (161) and the first valve core (162) is smaller than a distance H1 between a lower end surface of the lower spring seat (164) and the bottom of the second central hole (1602);
the lower spring seat (164) comprises a third central hole (1617) configured to limit a position of the first pressure regulating spring and a fourth central hole (1618) configured to mate with the first valve core (162), wherein the fourth central hole (1618) has a diameter larger than a diameter of the external thread (1612); and the lower spring seat (164) has an outer diameter the same as a diameter of the second central hole (1602).

14. The system according to any one of claims 1 to 13, wherein each of the flow limiting valve components (13) comprises a valve seat (131) connected to the common rail pipe (12), a second valve body (132), a second valve core (133), and a second pressure regulating spring (135), wherein

the valve seat (131) has a first fuel inlet hole (13101) communicating with the common rail pipe (12);
the second valve body (132) has a second axial through hole (13209) penetrating its upper and lower end surfaces, and the valve seat (131) is partially press-fitted into the second axial through hole (13209) from a lower end surface of the second valve body (132);
the second valve core (133) is mounted in the second axial through hole (13209) and disposed above the valve seat (131);
the second valve core (133) has an axial blind hole (13302) communicating with the first fuel inlet hole (13101), and a cavity is formed between an upper end of the second valve core (133) and the second axial through hole (13209);
the second valve core (133) is provided therein with a transverse flow restricting orifice (13304) communicating the axial blind hole (13302) with the cavity;
the second pressure regulating spring (135) is sleeved on the second valve core (133) and is positionally limited in the cavity;
a head of the upper end of the second valve core (133) has a third conical seal surface (13305) and a fourth conical seal surface (13306) connected to each other; and
a first sealing seat surface (13206) that is able to form a conical seal with the third conical seal surface (13305) is formed at a wall of the second axial through hole (13209), and a gap is able to be formed between the first sealing seat surface (13206) and the fourth conical seal surface (13306).

15. The system according to any one of claims 1 to 14, wherein the pressure limiting valve component (18) comprises a third valve body (181), a third valve core (182), a third pressure regulating spring (185), and a fuel pipe joint (186), wherein

the third valve body (181) is provided therein with a first-stage hole (18101), a second-stage hole (18102), a third-stage hole (18104), and a fourth-stage hole (18105) communicating in sequence from bottom to top;

the third valve core (182) has a head, which is slidably inserted from the fourth-stage hole (18105), through the third-stage hole (18104) and then partially located inside the second-stage hole (18102), and a conical seal is able to be formed between the third valve core (182) and the second-stage hole (18102);

a first gap allowing a fuel to pass therethrough is formed between the third valve core (182) and the third-stage hole (18104), and a second gap allowing the fuel to pass therethrough is formed between the third valve core (182) and the fourth-stage hole (18105); and

the fuel pipe joint (186) is fixed at an upper end of the third valve body (181), the fuel pipe joint (186) is provided therein with a first counterbore (18601), a second counterbore (18602), and a fuel outlet hole (18603) communicating in sequence from bottom to top, and the third pressure regulating spring (185) is positionally limited between the third valve core (182) and the second counterbore (18602).

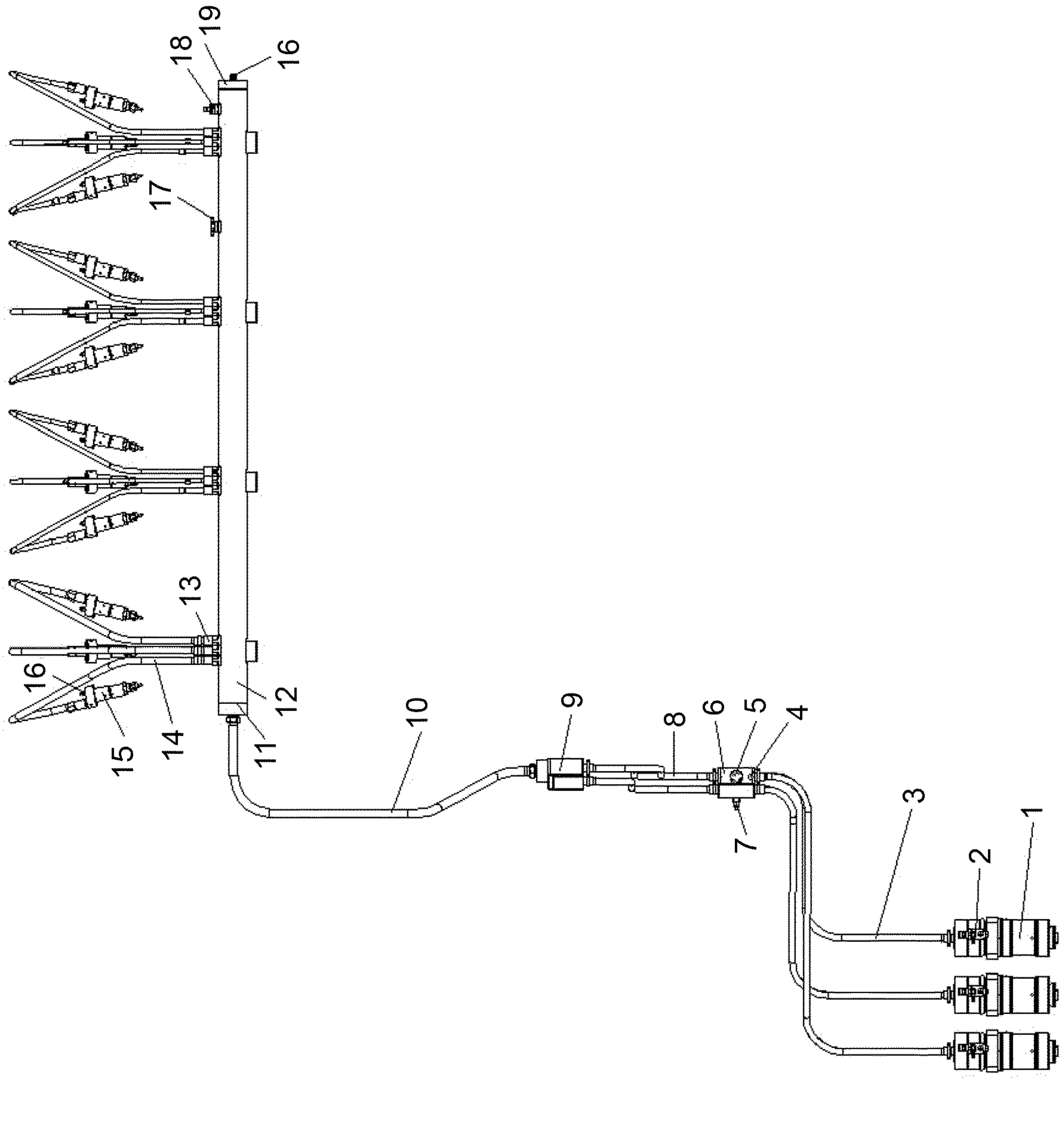

FIG.1

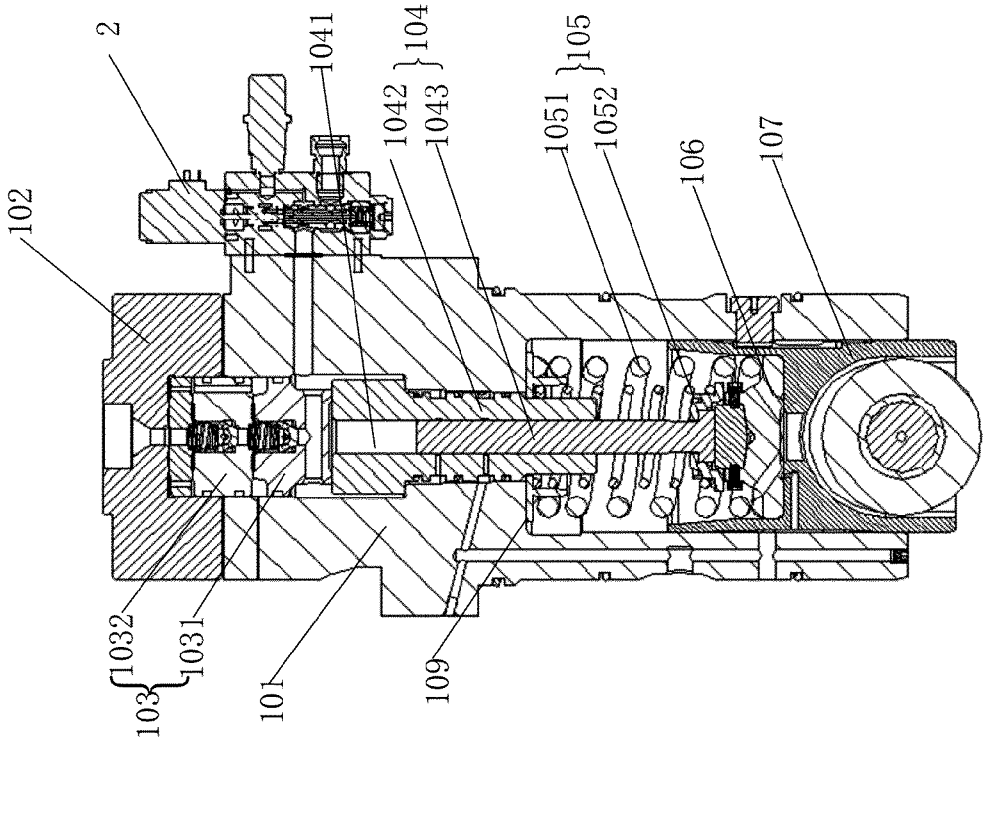

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FIG.2

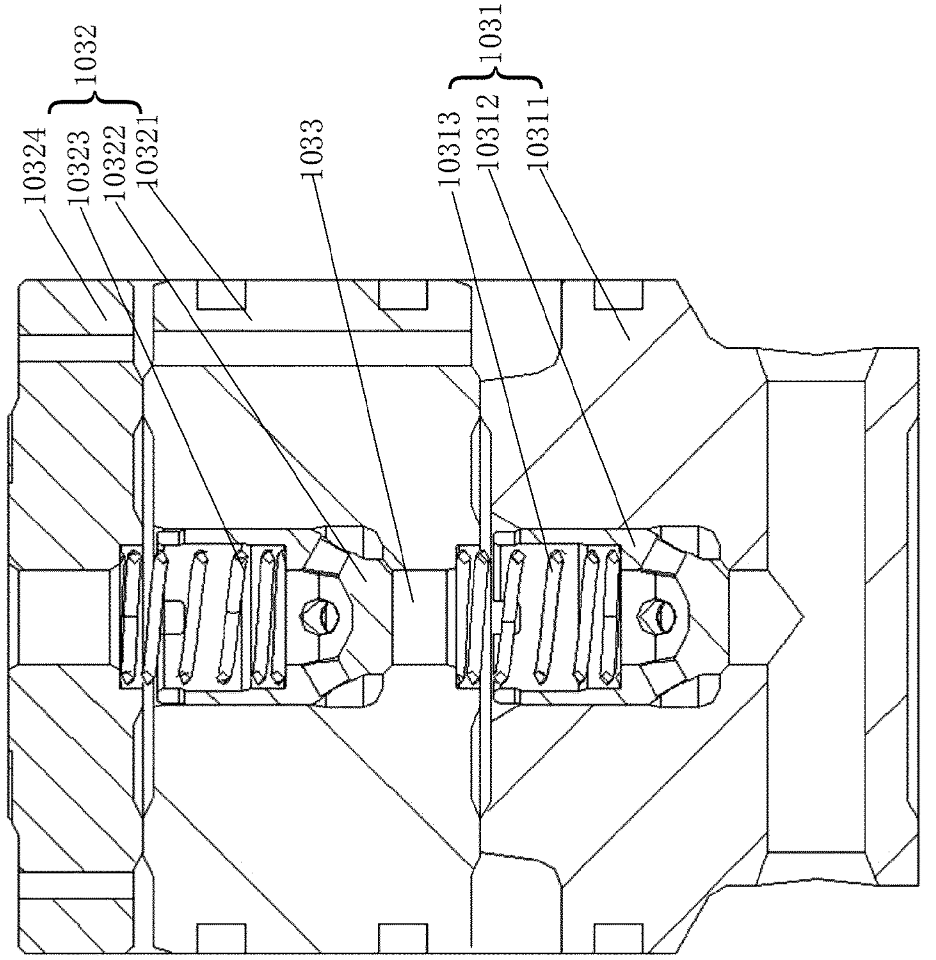

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FIG.3

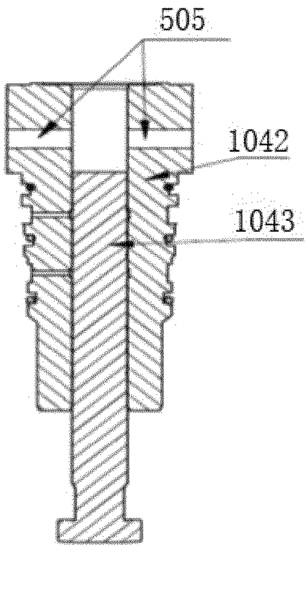

FIG.4

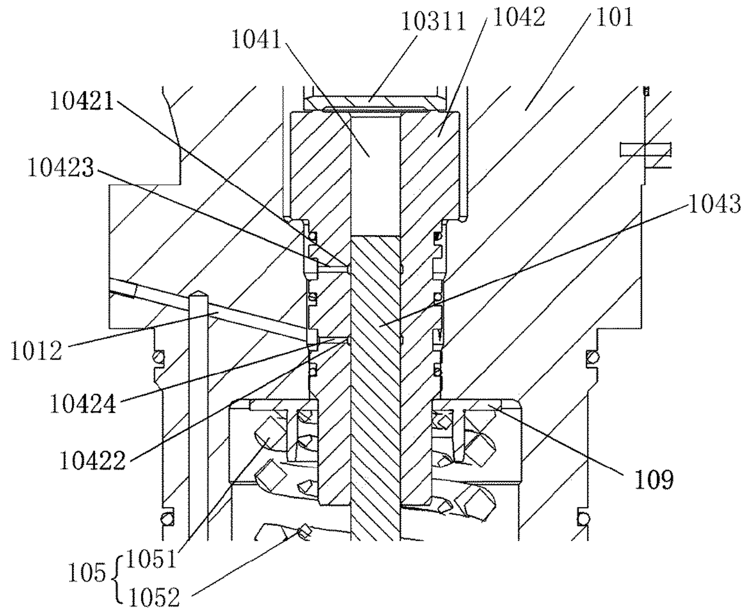

FIG.5

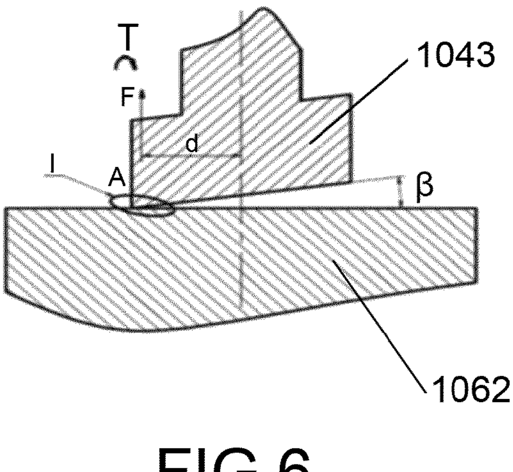

FIG.6

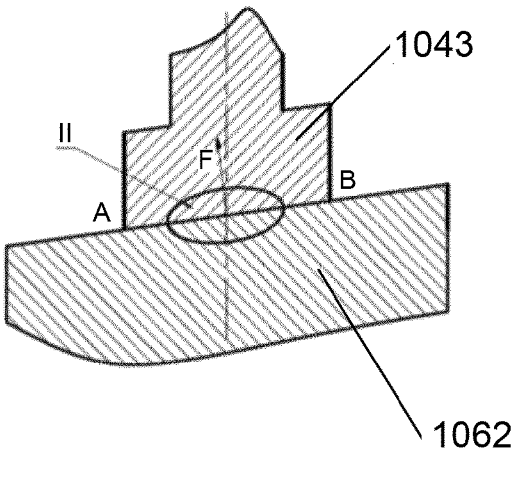

FIG.7

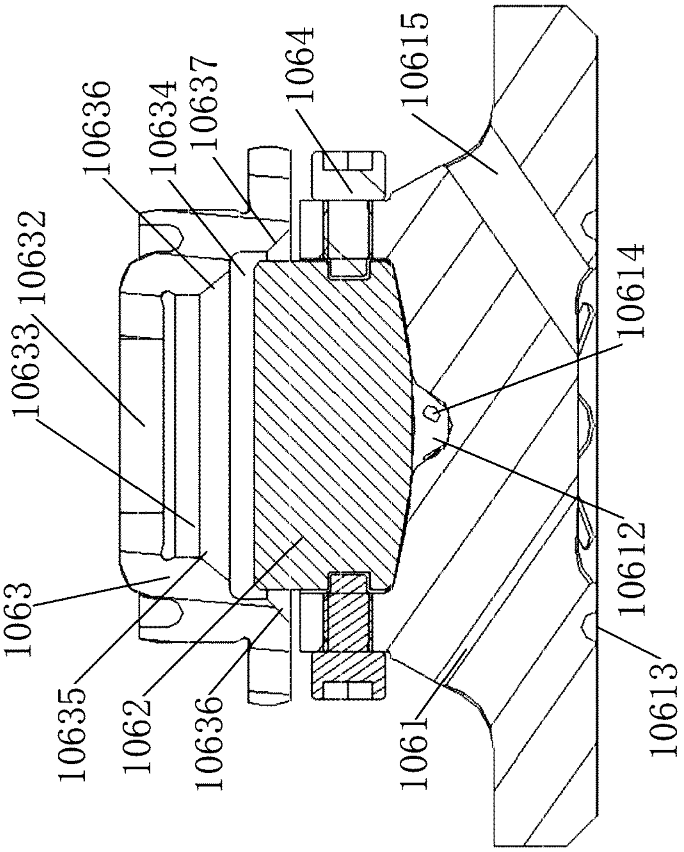
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FIG.9

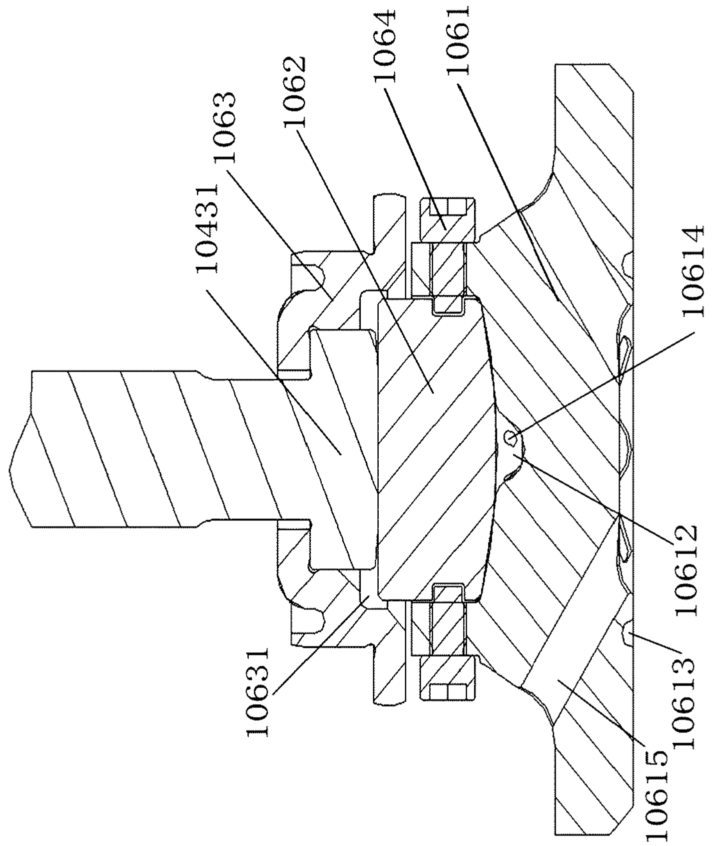
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FIG.8

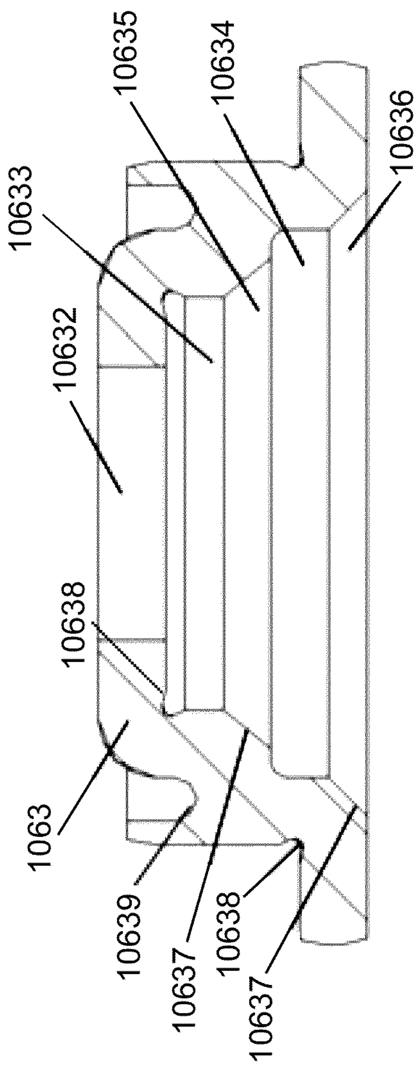

FIG.11

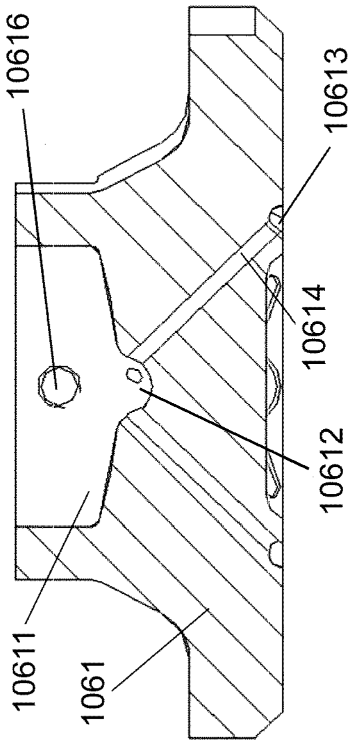

FIG.13

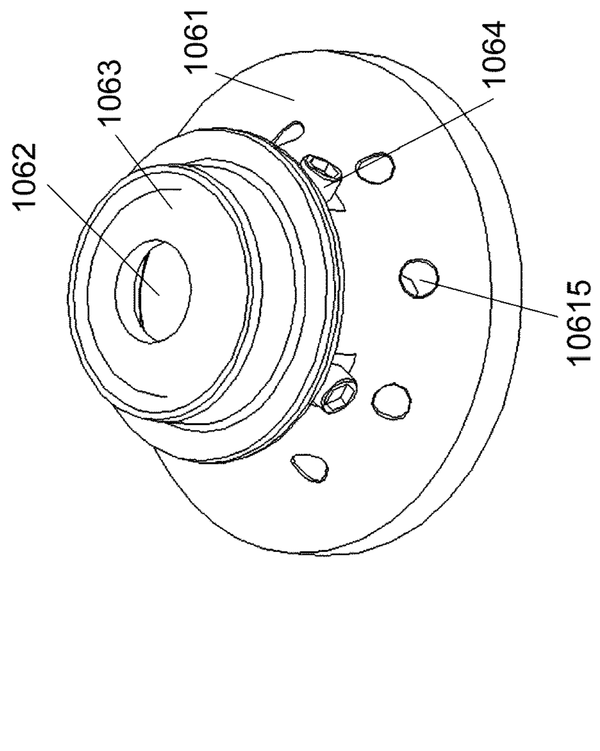

FIG.10

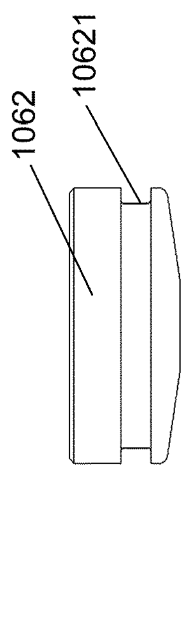

FIG.12

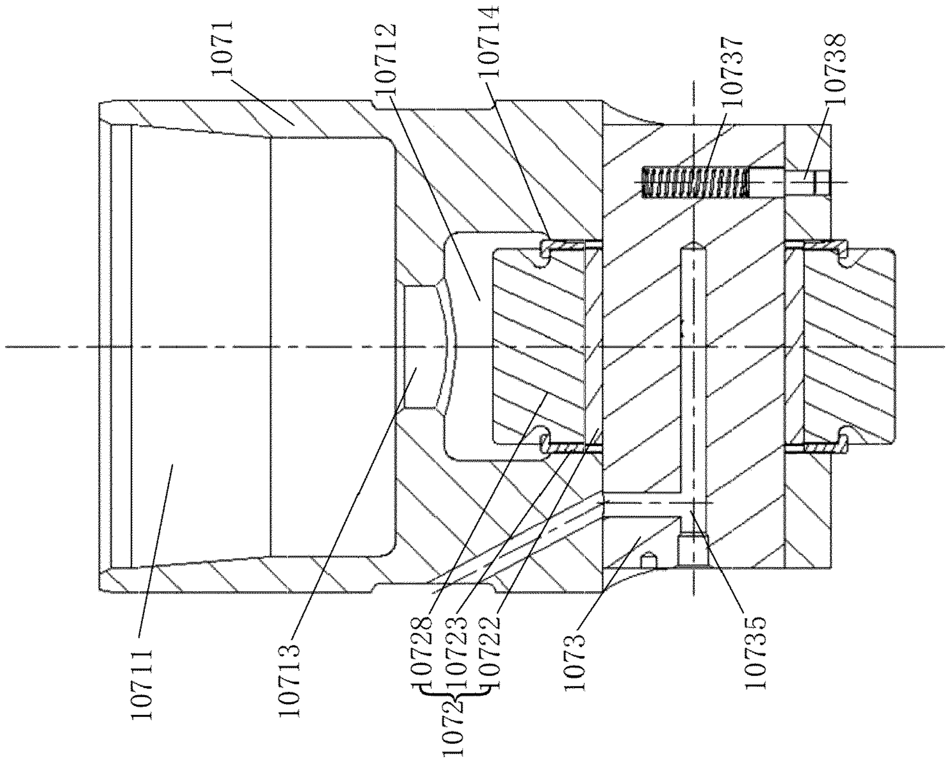
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FIG.15

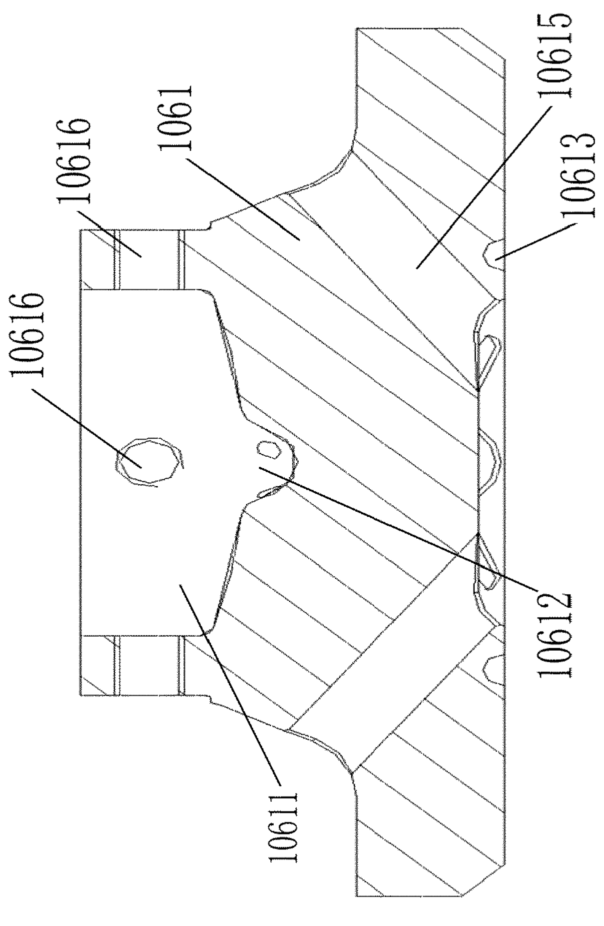
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FIG.14

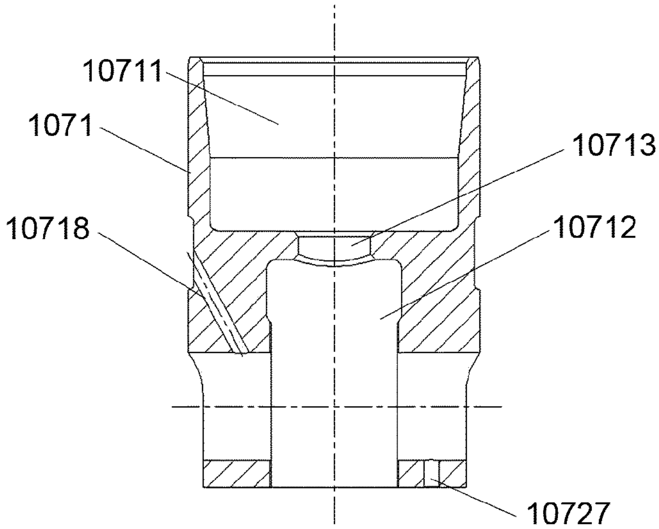

FIG.16

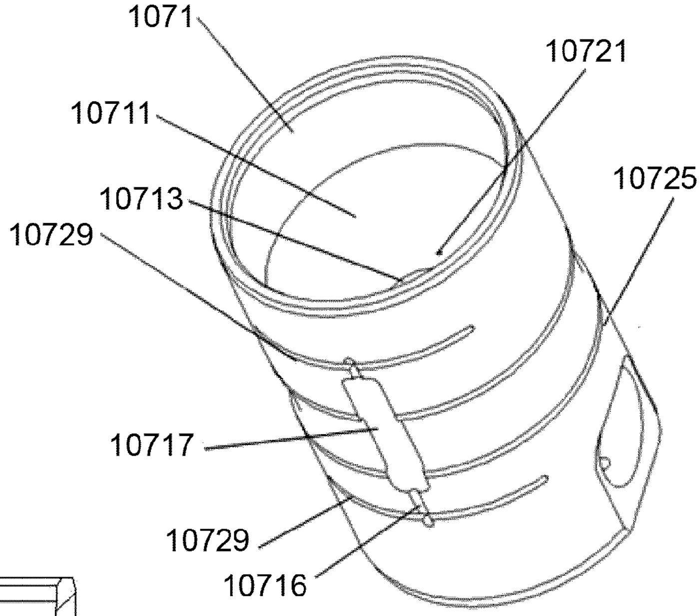

FIG.17

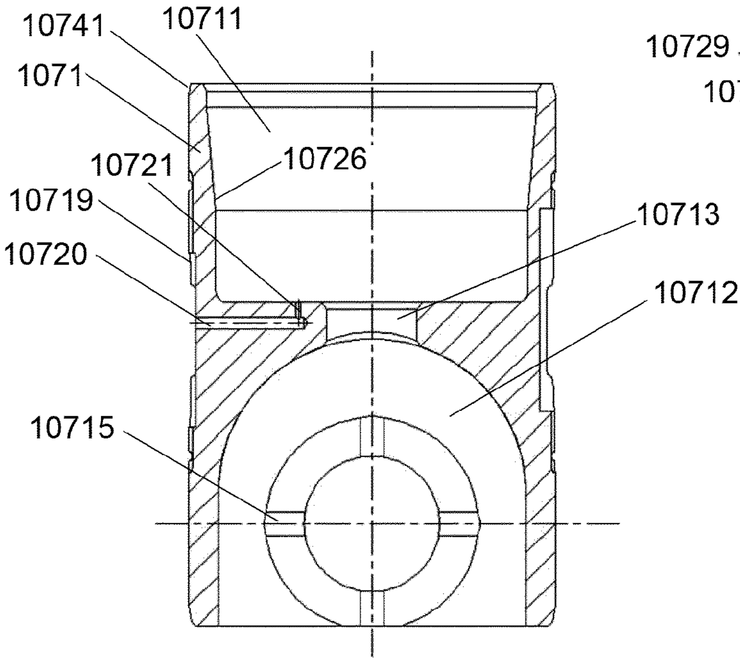

FIG.18

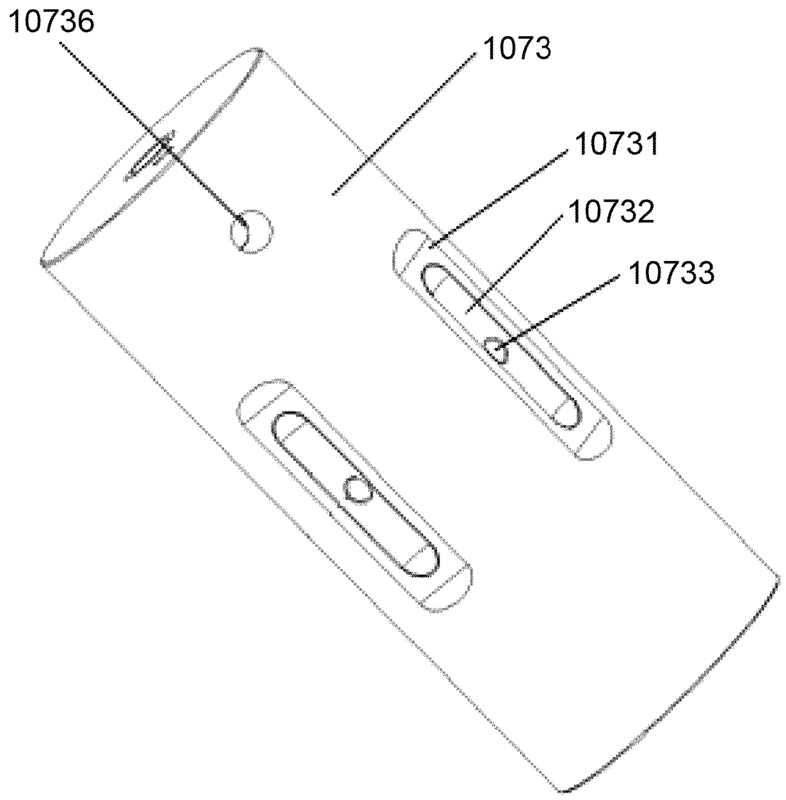

FIG.19

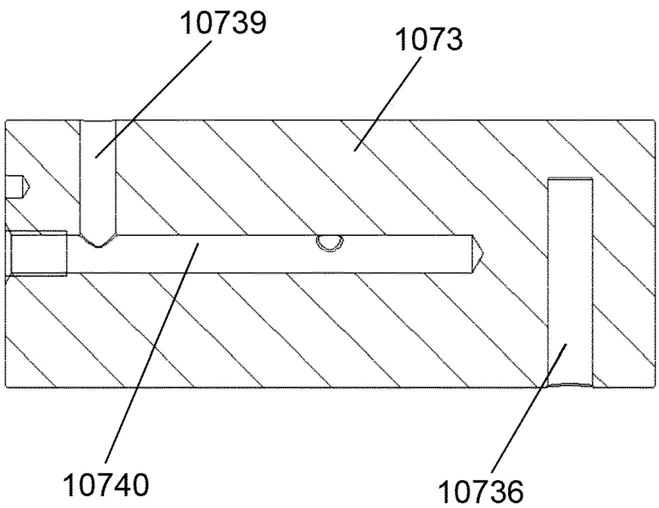

FIG.20

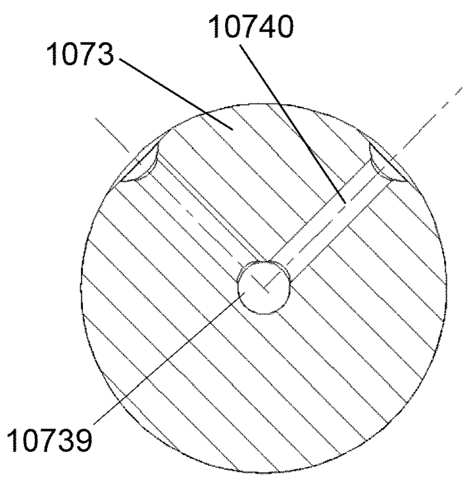

FIG.21

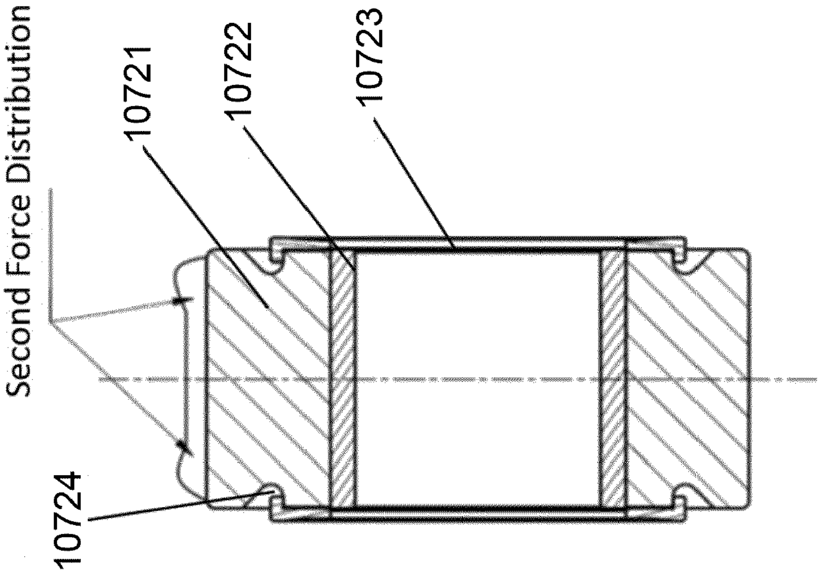
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FIG.23

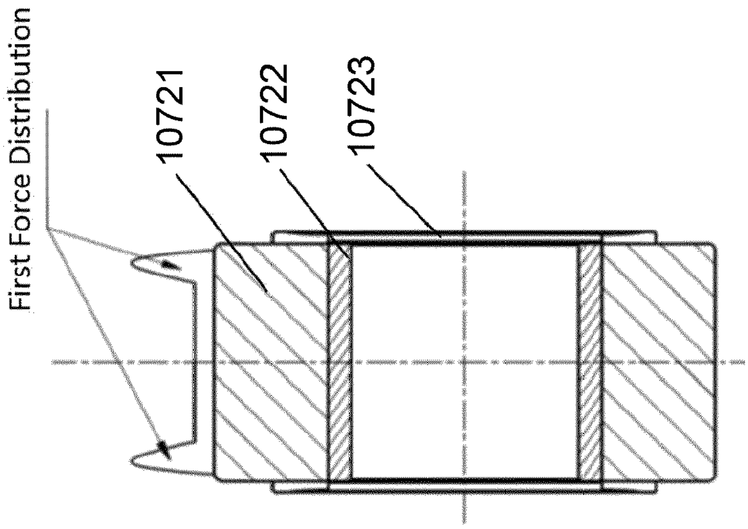
First Force Distribution
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FIG.22

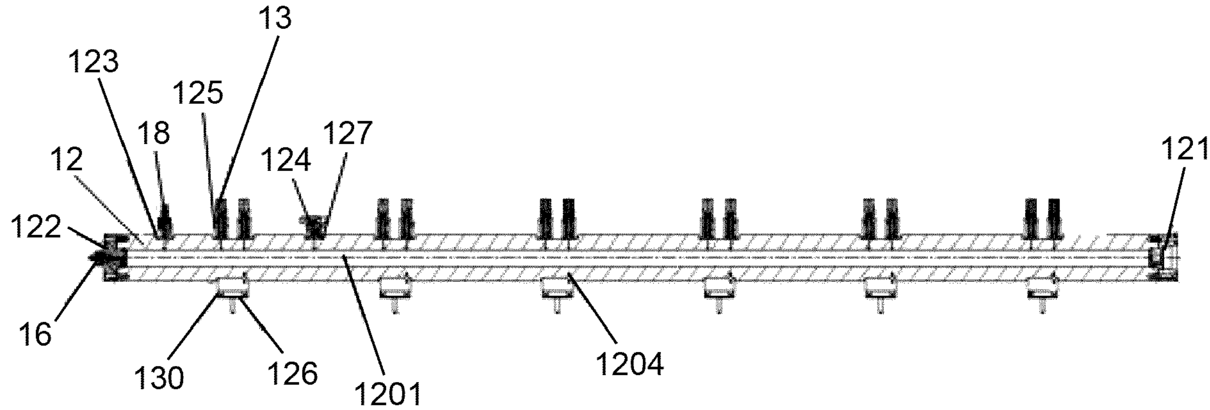

FIG.24

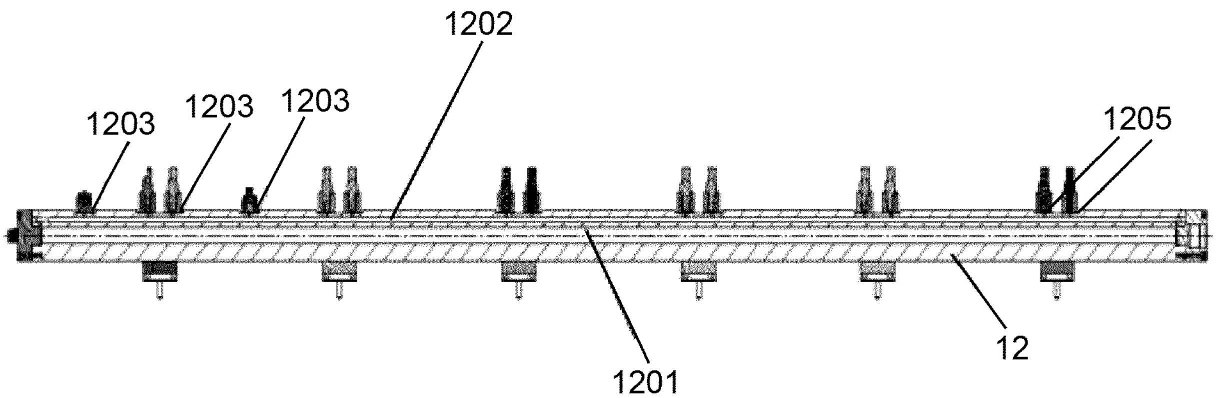

FIG.25

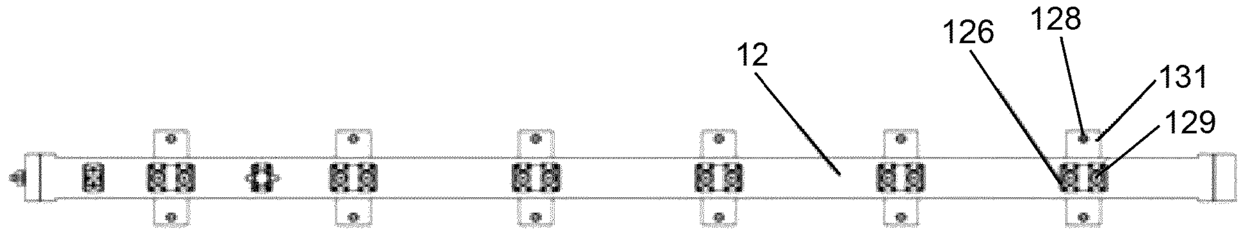

FIG.26

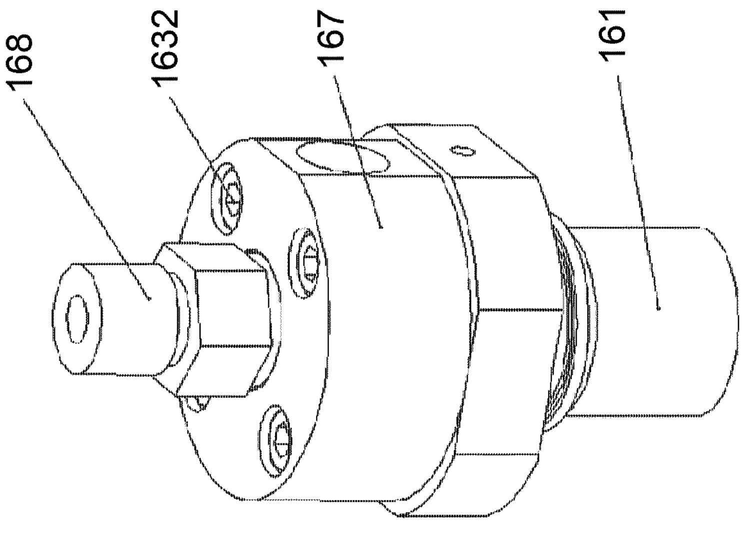
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FIG.27

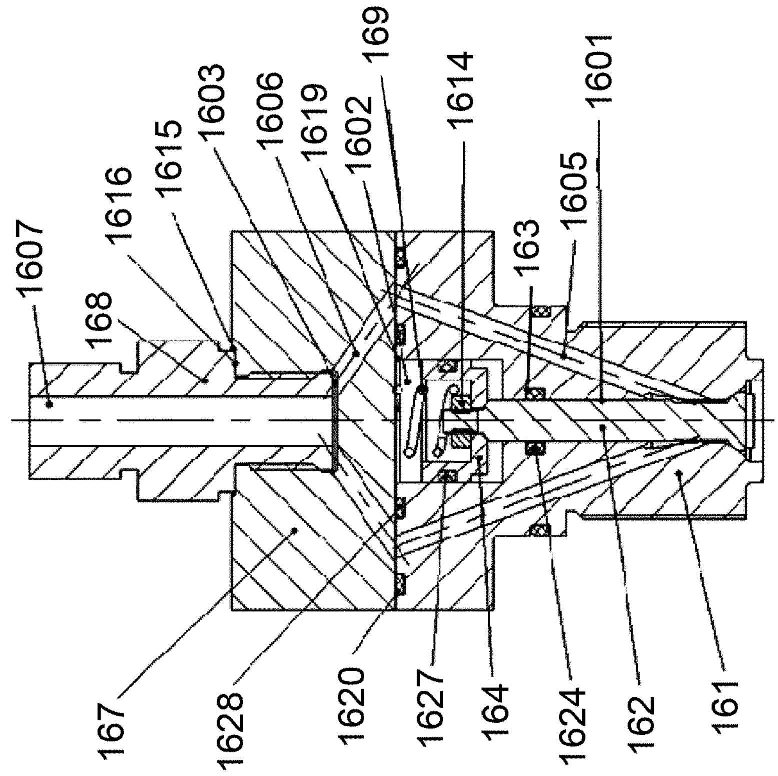
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FIG.28

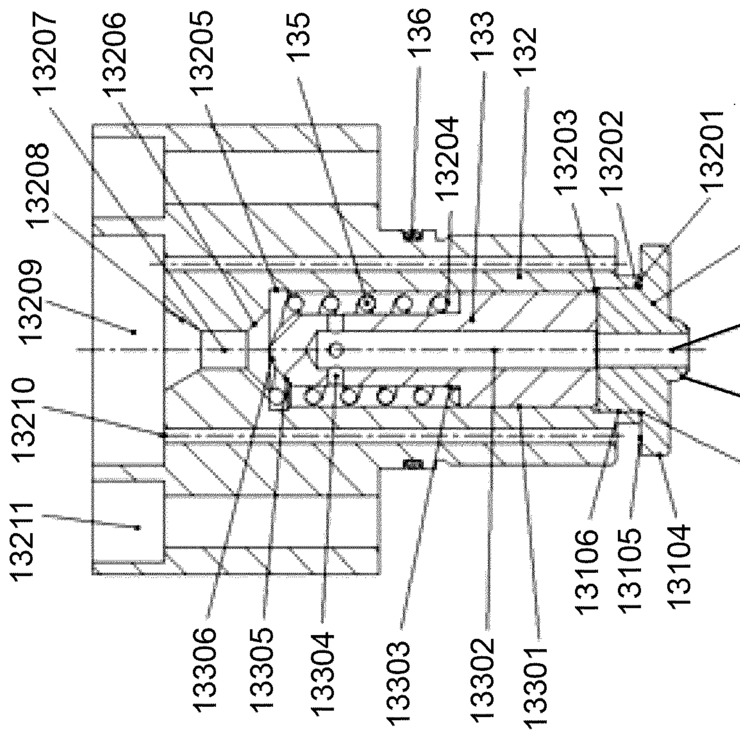
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FIG.30

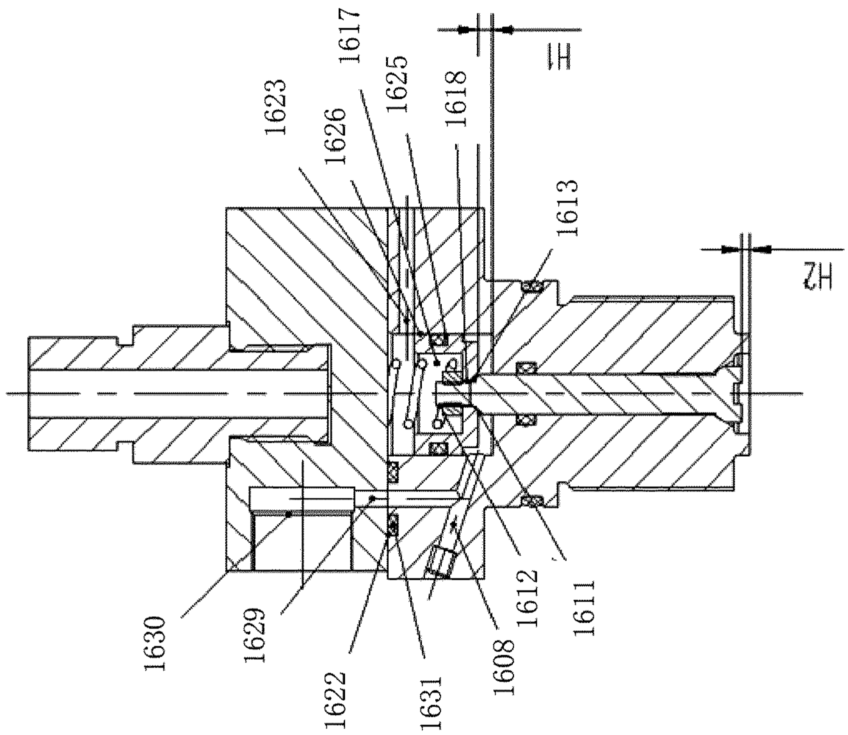
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FIG.29

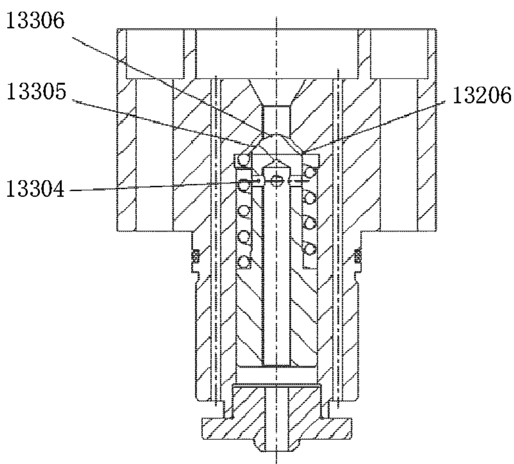

FIG.31

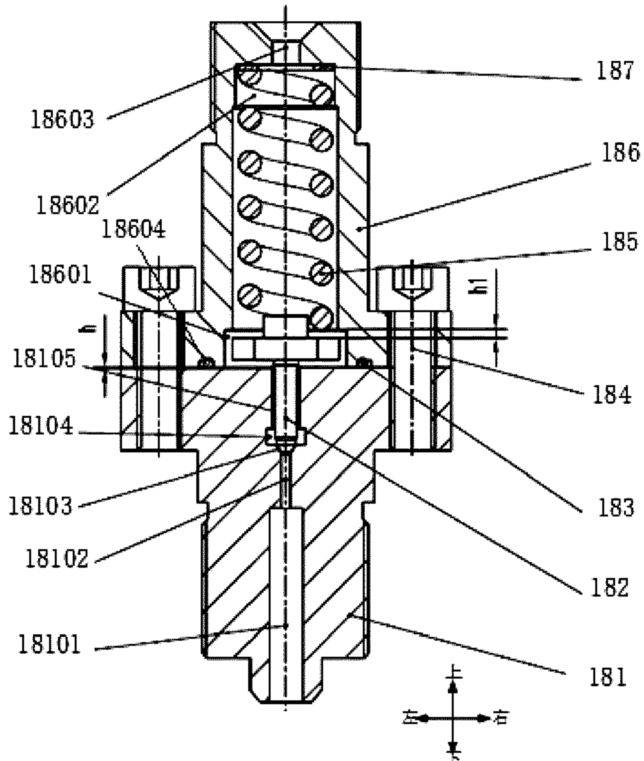

FIG.32

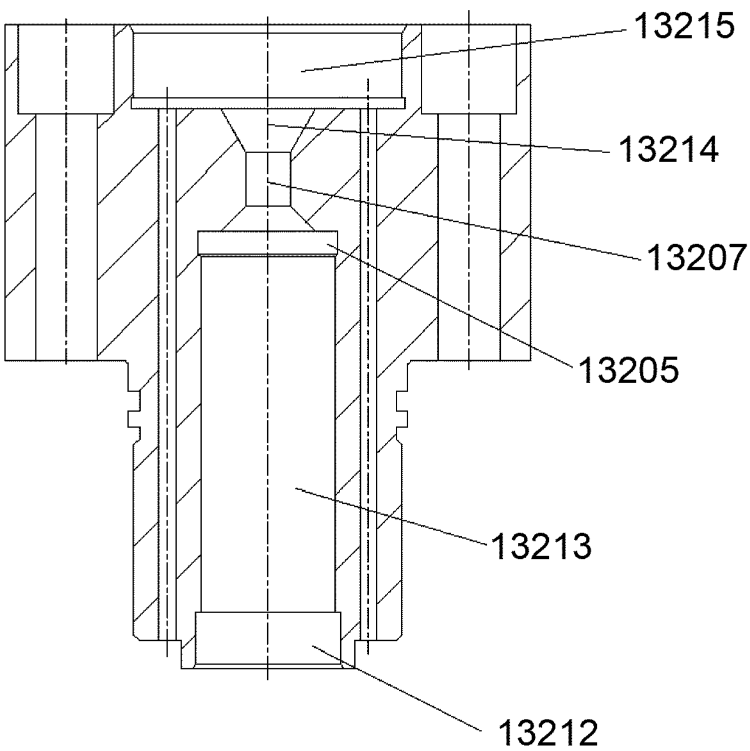

FIG.33

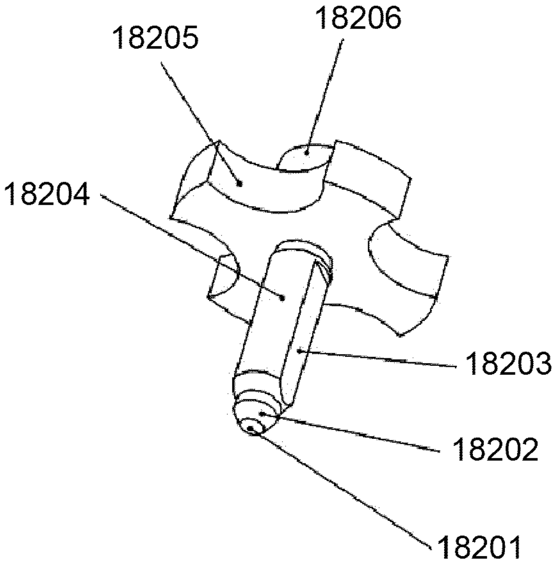
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FIG.34

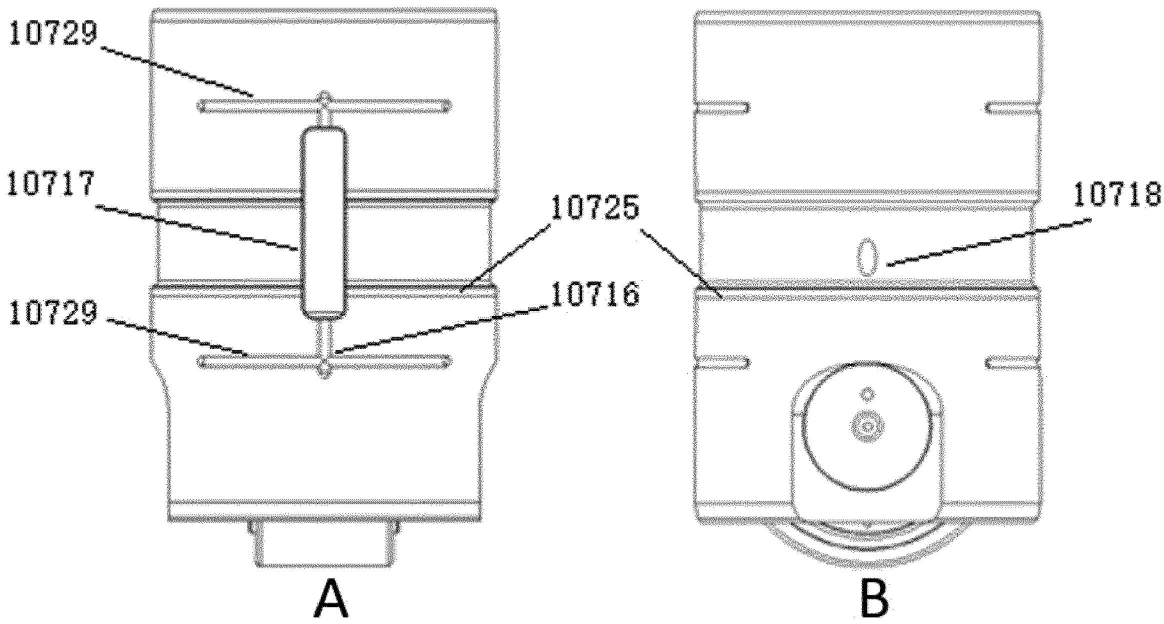
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FIG.35

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EUROPEAN SEARCH REPORT

Application Number
EP 20 20 5864

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	EP 1 314 883 A2 (MAN B & W DIESEL AG [DE]) 28 May 2003 (2003-05-28) * paragraphs [0002], [0006], [0007], [0019], [0025], [0029], [0042]; figures 1-4 *	1-15	INV. F02M63/02 F02D41/38 F02M55/02 F02M59/02 F02M63/00
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A	US 5 577 479 A (POPP HEINZ [AT]) 26 November 1996 (1996-11-26) * figure 1 *	1-15	
A	US 2016/153366 A1 (STEINERT ULF [DE]) 2 June 2016 (2016-06-02) * figure 1 *	1-15	
A	US 6 024 064 A (KATO MASAAKI [JP] ET AL) 15 February 2000 (2000-02-15) * figure 1 *	1-15	

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F02M
F02D

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	18 March 2021	Le Bihan, Thomas

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18-03-2021

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