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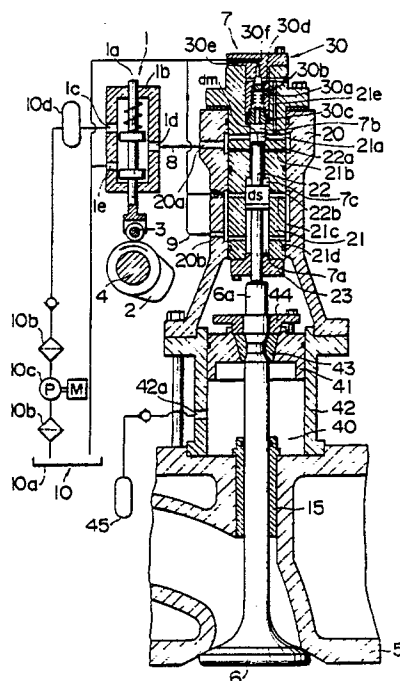
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54 **Valve system of internal combustion engine.**

57 The present invention relates to a hydraulic valve system to be used for driving an intake valve or an exhaust valve in a Diesel engine or the like. In the valve system there is employed as an actuator a piston (22) equipped with a large-diameter piston (22b) and a small-diameter piston (22a). In the valve-opening stroke, the hydraulic pressure is arranged to be applied to both of the large- and small-diameter pistons as a first stage, and the hydraulic pressure is applied only to the small-diameter piston as a second stage. Further, in the valve-closing stroke, the valve is operated by a spring (40), and the speed of the valve closing is closed by a two-stage cushioning action.

**FIG. 1**



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## VALVE SYSTEM OF INTERNAL COMBUSTION ENGINE

### 2. FIELD OF THE INVENTION AND RELATED ART STATEMENT:

The present invention relates to an accumulator type valve system for controlling opening and closing of an exhaust valve or an intake valve for a Diesel engine.

Figure 8 shows the prior art accumulator type valve system. The system comprises a five-port directional control valve 01 which controls a hydraulic fluid that is pressurized by a pump 010 and accumulated to a predetermined pressure in an accumulator 011, a cam 02, a tappet roller and a cam shaft 04 that drive the control valve, an actuator which moves an intake and exhaust valve 06 provided on a cylinder head 05 by means of the hydraulic fluid, and pipelines 08 and 09. Figure 8 shows the state in which the tappet roller 03 is on the base circle (namely, the valley) part of the cam 02, the control valve 01 is introducing the hydraulic fluid to a lower chamber 07a of the actuator 07 through the pipeline 09, and the intake and exhaust valve 06 is closed by being moved upward under the force of the hydraulic fluid. In this situation the hydraulic fluid in an upper chamber 07b of the actuator 07 is drained into a tank 010a of the pump 010 through the pipeline 08 and the control valve 01. When the tappet roller 03 is lifted as a result of rotation of the cam 02, the hydraulic fluid in the lower chamber 07a is drained into the tank 010a through the pipeline 09 and the control valve 01, and the high pressure hydraulic fluid in the accumulator 011 acts on the upper chamber 07b via the control valve 01 and the pipeline 08. The force of the hydraulic fluid moves the intake and exhaust valve 06 downward against the pressure in the cylinder that acts on the intake and exhaust valve 06, whereby causing the intake and exhaust valve to open. When the lift of the tappet roller 03 is decreased and reaches the base circle of the cam by a further rotation of the cam 02, the control valve 01 takes on the state in the figure and the intake and exhaust valve 06 is closed by the upward motion as described above.

As in the above, the prior art apparatus requires five ports for the control valve 01 which makes its structure complex and large-sized, moreover, there are needed two high pressure tubes 08 and 09 and the actuator 07 is complicated and necessarily large-sized. In addition, the prior art device makes use of a hydraulic fluid at high pressure for opening and closing the input and exhaust valve 06, and the high pressure hydraulic fluid is drained into the tank 010a and discarded after operating the actuator 07, whereby increasing

the consumption of the hydraulic fluid and the energy loss. Moreover, the prior art device has a problem that it is difficult to control the opening and closing and the seating of the intake and exhaust valve.

### 3. OBJECT AND SUMMARY OF THE INVENTION:

It is the object of the present invention to provide a valve system for an internal combustion engine which enables to eliminate problems occurring in the prior art system, to simplify the structures of the control valve and the actuator, to optimally control the valve closing speed, and to suppress the consumption of the hydraulic fluid to a necessary minimum and markedly reduce the power loss by performing the valve closing by means of the spring force.

In a valve system equipped with an accumulator-incorporated hydraulic power source which stores the hydraulic fluid by pressuring the fluid, an actuator having a hydraulic fluid operated piston which drives the intake valve or the exhaust valve, and a control valve which supplies the hydraulic fluid in the accumulator to the actuator in a controlled manner, the valve system for the internal combustion engine in accordance with the present invention is characterized in that; the piston consists of a small-diameter main piston and a large-diameter sub-piston; and a main communicating port which supplies the hydraulic fluid to the top surface of the main piston, a sub-communicating port which supplies the hydraulic fluid to the top surface of the sub-piston, and a drainage port which drains the hydraulic fluid that acted on the sub-piston are drilled in a cylinder which guides the piston in an oiltight manner, the main- and sub-communicating ports and the drainage port are arranged so as to be opened and closed by the sliding motion of the main piston and the sub-piston, respectively, and the intake and exhaust valve is equipped with a valve spring which energizes the valve in the direction of the valve closing.

Namely, in the present invention the actuator is given a structure in which the high pressure hydraulic fluid is used exclusively for the opening of the intake and exhaust valve, and the energy of the valve spring stored accompanying the lift of the intake and exhaust valve is used for opening the valve.

With the above arrangement, the structures of the control valve, the high pressure pipelines and the actuator are simplified and made small-sized,

and it becomes possible to reduce the consumption of the high pressure hydraulic fluid at the time of valve opening and to save the driving energy.

#### 4. BRIEF DESCRIPTION OF THE DRAWINGS:

Figures 1 through 7 show an embodiment of the valve system in accordance with the present invention, wherein FIG. 1 is an overall sectional diagram of the system and FIGS. 2 to 7 are sectional diagrams of the important parts for explaining the action of the actuator of the system and FIG. 8 is a sectional diagram showing the parts corresponding to FIG. 1 of the prior art system.

#### 5. DETAILED DESCRIPTION OF A PREFERRED EMBODIMENT:

Referring to FIGS. 1 through 7, an embodiment of the present invention will now be described next.

In FIG. 1 which shows the principal structure of the valve system of the exhaust valve in the Diesel engine, 1 is a three-port directional control valve, and a piston 1a is driven in the vertical direction via a tappet roller 3 by a cam 2 fixed to a cam shaft 4. The piston 1a slides freely by keeping oil tightness within a cylinder 1b equipped with a hydraulic fluid inlet port 1c, an outlet and inlet port 1d and a drainage port 1e. Reference numeral 8 is a high pressure pipe which connects the control valve 1 to the actuator 7. The actuator 7 consists of a main body 20, a cylinder 21, a piston 22, a cover 23 and a check valve unit 30. In the cylinder 21 there are formed an upper chamber 7b, an intermediate chamber 7c and a lower chamber 7a, and on the circumferential wall of the cylinder there are formed a main communicating port 21a, a sub-communicating port 21b, an intermediate port 21c, a lower port 21d and an oil passage 21e. The main communicating port 21a connects an inlet and outlet port 20a of the main body 20 and the upper chamber 7b of the cylinder 21. The sub-communicating port 21b connects the inlet and outlet port 20a of the main body 20 and the intermediate chamber 7c. The intermediate port 21c connects a return port 20b of the main body 20 and the intermediate chamber 7c. The lower port 21d connects the return port 20b of the main body 20 and the lower chamber 7a. The oil passage 21e connects the main communicating port 21a and the check valve unit 30. The piston 22 has a small-diameter main piston 22a which partitions the upper chamber 7b and the intermediate chamber 7c, and a large-diameter sub-piston 22b which partitions the intermediate chamber 7c and the lower chamber 7a. The main piston 22a and the sub-

piston 22b respectively slide on the inside of the cylinder 21 in oil-tight manner. The check valve unit 30 has a valve plug 30b which is oil-tightly guided on the inside of a guide 30f and is energized upward by a stopper 30c and a spring 30a. The valve plug 30b disconnects the oil passage 21e and the upper chamber 7b with its tip abutted on a seat 30d. The valve plug 30b blocks the flow of the hydraulic fluid from the upper chamber 7b to the oil passage 21e, but permits the flow of the hydraulic fluid from the oil passage 21e to the upper chamber 7b. Further, the check valve 30 has an orifice 30e at an end of the oil passage 21e. The orifice 30e is for discharging air that is mixed in the hydraulic fluid to the outside of the actuator 7 along with the fluid.

The exhaust valve 6 slides within a guide 15 fixed to a cylinder head 5 and is energized in the upward direction by a pneumatic valve spring 40. The upper end of a valve stem 6a of the exhaust valve 6 is abutted on the piston 22. A piston 41 is arranged via a cone sleeve 43 around the valve stem 6a of the exhaust valve 6, and the piston 41 is pressure-fitted to the cone sleeve 43 by means of a presser 44. The piston 41 is air-tightly guided in the piston 41. Pressurized air is supplied to the interior of the cylinder 42 from an air tank 45 through an air hole 42a, whereby forming a pneumatic valve spring 40 within the cylinder 42. The actuator 7 is fixed to the upper part of the cylinder 42.

A hydraulic power source 10 consists of a tank 10a, a filter 10b, a high-pressure pump 10c, an accumulator 10d, and the like, wherein a high-pressure hydraulic fluid is sent by the pump 10c to the accumulator 10d and the accumulator stores the high-pressure hydraulic fluid.

Next, the valve opening operation of the exhaust valve 6 will be described.

Referring to FIG. 2, the tappet roller 3 is lifted with the rotation of the cam shaft 4, the piston 1a is raised to disconnect the drainage port 1e from the outlet and inlet port 1d, and the outlet and inlet port 1d is communicated with the inlet port 1c. As the piston 1a is further raised later, the hydraulic fluid in the accumulator 10d is supplied to the inlet and outlet port 20a of the actuator 7 via the control valve 1, whereby a portion of the fluid is supplied from the inlet and outlet port 20a to the upper chamber 7b via the main communicating port 21a, the oil passage 21e and the check valve unit 30, while the remaining portion is supplied to the intermediate chamber 7c from the inlet and outlet port 20a via the sub-communicating port 21b. Under the action of these portions of the high-pressure hydraulic fluid the piston 22 is moved downward in the direction of the arrow → to be brought to the condition as illustrated in FIG. 3. Figure 3 depicts

the state of the system at the time the main communicating port 21a is just about to be communicated with the upper chamber 7b. The valve plug 30b of the check valve unit 30 which has been separated from the seat 30d up to this point is thereafter brought into contact with the seat 30d under the direct inflow of the hydraulic fluid through the main communicating port 21a into the upper chamber 7b, and the communication of the oil passage 21e with the upper chamber 7b is interrupted. The piston 22 is further moved downward and the state shown in FIG. 4 is reached. The state in FIG. 4 depicts the situation which the communication of the sub-communicating port 21b with the intermediate chamber 7c is about to be interrupted. With a further downward motion, from this condition, of the piston in the direction of the arrow  $\rightarrow$ , the hydraulic fluid acts only on the top surface of the main piston 22a without acting on the top surface of the sub-piston 22b. Then, the piston 22 is lowered and the state shown in FIG. 5 is reached. The state in FIG. 5 depicts the condition at the time when the intermediate chamber 7c is just about to be communicated with the intermediate port 21c. It should be noted that the system is designed such that the fluid in the intermediate chamber 7c is allowed to expand during the displacement  $\delta_1$  (see FIG. 4) of the piston 22 from the position in FIG. 4 to that in FIG. 5, but following the condition in FIG. 5 the fluid in the intermediate chamber 7c is discharged to the intermediate port 21c by the main piston 22a under the condition of positive pressure. The drain oil in the intermediate port 21c is drained to the tank 10a of the hydraulic power source 10 via the return port 20b. The hydraulic fluid that enters the upper chamber 7b from the inlet and outlet port 20a via the main communicating port 21a acts on the top surface of the main piston 22a, so that the piston 22 is moved downward in the direction of the arrow  $\rightarrow$  and reaches the condition shown in FIG. 6. The condition in FIG. 6 illustrates the timing at which the lower chamber 7a and the lower port 21d are just about to be disconnected. With a further downward motion of the piston 22, the pressure of the fluid within the lower chamber 7a is raised because of the sealing, and the fall of the piston 22 is brought to a gentle stop by the pressure of the hydraulic fluid. Namely, the lower chamber 7a forms a cushion chamber, and the valve-opening motion is completed by the achievement of a maximum lift position ( $l_{\max}$  in FIG. 7) by the exhaust valve 6. It is to be noted that although the hydraulic fluid is further kept acting on the top surface of the main piston 22a via the inlet and outlet port 20a, the main- and sub-communicating ports 21a and 21b and the upper chamber 7b, the exhaust valve 6 is not moved. However, the pressure within the

cylinder 42 of the pneumatic valve spring has been increased during the above period.

Next, the valve-closing operation of the exhaust valve 6 will be described.

During the operation the piston 22 moves in the direction of the arrow  $\rightarrow$  in FIG. 2 to FIG. 6. When the cam 2 starts rotating from the state in FIG. 6 and the tappet roller 3 reaches the base circle and the piston 1a of the control valve 1 achieve the state in FIG. 1, the high-pressure hydraulic fluid in the upper chamber 7b is drained into the tank 10a of the hydraulic power source 10 via the main- and sub-communicating ports 21a and 21b, the inlet and outlet port 20a and the high-pressure tube 8 through the control valve 1. The exhaust valve 6 is energized upward via the cone sleeve 43 and the presser 44 by the pneumatic pressure of the pneumatic valve spring 40 that acts on a piston 41. The valve closing motion is achieved by the pneumatic valve spring 40. When the state in FIG. 6 goes to that of FIG. 5, the intermediate chamber 7c is closed by the sub-piston 22b. Thereafter, the hydraulic pressure the intermediate chamber 7c goes up accompanying the rise of the piston 22, performs a primary cushioning action by the fluid pressure during the displacement  $\delta_1$  shown in FIG. 4, and gently decelerates the speed of upward motion of the piston 22. When the piston 22 reaches the condition in FIG. 3 by a further motion of the piston 22 in the direction of the arrow  $\rightarrow$ , the communication of the upper chamber 7b with the main communicating port 21a is interrupted. Since, however, the seat 30d of the check valve unit 30 is closed, the pressure of the fluid in the upper chamber 7b is raised thereafter and the upper chamber 7b forms a fluid pressure cushioning chamber. The speed of the upward motion of the piston 22 is further decreased accompanying the secondary cushioning action of the upper chamber 7b shown in FIG. 2. During the displacement  $\delta_2$  until the seating of the exhaust valve 6, the speed of the exhaust valve 6 is so controlled as to be optimum for the valve seating to terminate the upward movement, completing the valve-closing operation. The motion described in the above is represented by diagrams in which FIG. 7(a) shows the relationship between the lift  $l$  of the piston 22 and the crank angle  $\theta_k$ , and FIG. 7(b) shows the relationship between the exhaust valve resistance  $F_v$ , forces  $F_1$  and  $F_2$  acting on the piston 22 and the crank angle  $\theta_k$ . In FIG. 7(a),  $l_1$  indicates the distance during which the hydraulic fluid acts on the main- and sub-pistons as shown in FIG. 2. The pressure  $P$  of the hydraulic fluid, diameter  $d_s$  of the sub-piston and the lift  $l_1$  are related to the force  $F_1$  pressing the exhaust valve downward and the consumption of the hydraulic fluid by the following relations:

$$F_1 = \frac{\pi}{4} d_s^2 P,$$

$$Q_1 = \frac{\pi}{4} d_s^2 l_1,$$

where the force  $F_1$  and the exhaust valve resistance  $F_v$  satisfies the inequality  $F_1 > F_v$ .

The interval of the lift from  $l_1$  to  $l_{\max}$  corresponds to the period during which the hydraulic fluid acts only on the main piston 22a. The diameter  $d_m$  of the main piston is related to the force  $F_2$  pressing the exhaust valve downward and the consumption  $Q_2$  of the hydraulic fluid by the following relations:

$$F_2 = \frac{\pi}{4} d_m^2 P,$$

$$Q_2 = \frac{\pi}{4} d_m^2 (l_{\max} - l_1),$$

where the force  $F_2$  and the exhaust valve resistance  $F_v$  satisfies the inequality  $F_2 > F_v$ .

Since the consumption  $Q$  of the hydraulic fluid in the case of the prior art system which is not two-stage type is given by

$$Q = \frac{\pi}{4} d_s^2 l_{\max},$$

it follows that

$$Q_1 + Q_2 < Q$$

and it can be seen that it is possible to reduce the consumption of the hydraulic fluid to a large extent.

Moreover, during the valve-closing movement over a distance  $l_2$ , a primary cushioning is operative for the period corresponding to the displacement  $\delta_1$  which decelerates the upward motion of the piston 22, and a secondary cushioning is operative for the period corresponding to the displacement  $\delta_2$ , so that it is possible to optimally control the seating speed of the exhaust valve, whereby prolonging the service life of the exhaust valve.

It should be noted that in the above description the control valve 1 is operated by mechanically driving the piston 1a with the cam 2. However, it is also possible to drive the piston 1a electrically or to use an electromagnetic valve as the control valve.

With the aforementioned construction of the present invention, the structure of the control valve is simplified to involve only three ports, the number of the high-pressure pipes is reduced to one, and the construction of the actuator is correspondingly simplified. As a result, there are created cushioning actions due to the hydraulic pressure which eliminate the mechanically colliding parts, and it becomes possible to optimally control the seating of the exhaust valve by causing the valve to terminate its valve-opening cycles in a gentle manner.

Moreover, the hydraulic fluid is arranged to act on the piston in two stages in response to the resistance of the exhaust valve, the consumption of the hydraulic fluid is suppressed to a necessary minimum, and the driving energy of the hydraulic fluid is reduced, whereby contributing markedly to the reduction of the power loss.

Furthermore, although the aforementioned embodiments are described in conjunction with the valve system of an exhaust valve, the valve system

of the present invention can naturally be applied also to an intake valve.

## 5 Claims

In a valve system equipped with a hydraulic power source having an accumulator which stores a hydraulic fluid by pressurizing the fluid, an actuator having a piston activated by the hydraulic fluid which drives an intake valve or an exhaust valve, and a control valve which supplies the hydraulic fluid in said accumulator to said actuator in a controlled manner, the valve system for an internal combustion engine characterized in that said piston (22) comprises a main piston (22a) with small diameter and a sub-piston (22b) with large diameter, and a main communicating port (21a) for supplying the hydraulic fluid on the top surface of the main piston (22a), a sub-communicating port (21b) for supplying the hydraulic fluid on the top surface of the sub-piston (22b) and drainage ports (21c) and (21d) for draining the hydraulic fluid that acted on the sub-piston (22b) are drilled in a cylinder (21) that oil-tightly guides the piston (22), whereby the main- and sub-communicating ports (21a), (21b) and the drainage ports (21c), (21d) are arranged so as to be opened and closed by the sliding of the main piston (22a) and the sub-piston (22b), respectively, and a spring is disposed so as to energize an intake and exhaust valve (6) in the direction of closing the valve (6).

FIG. 1

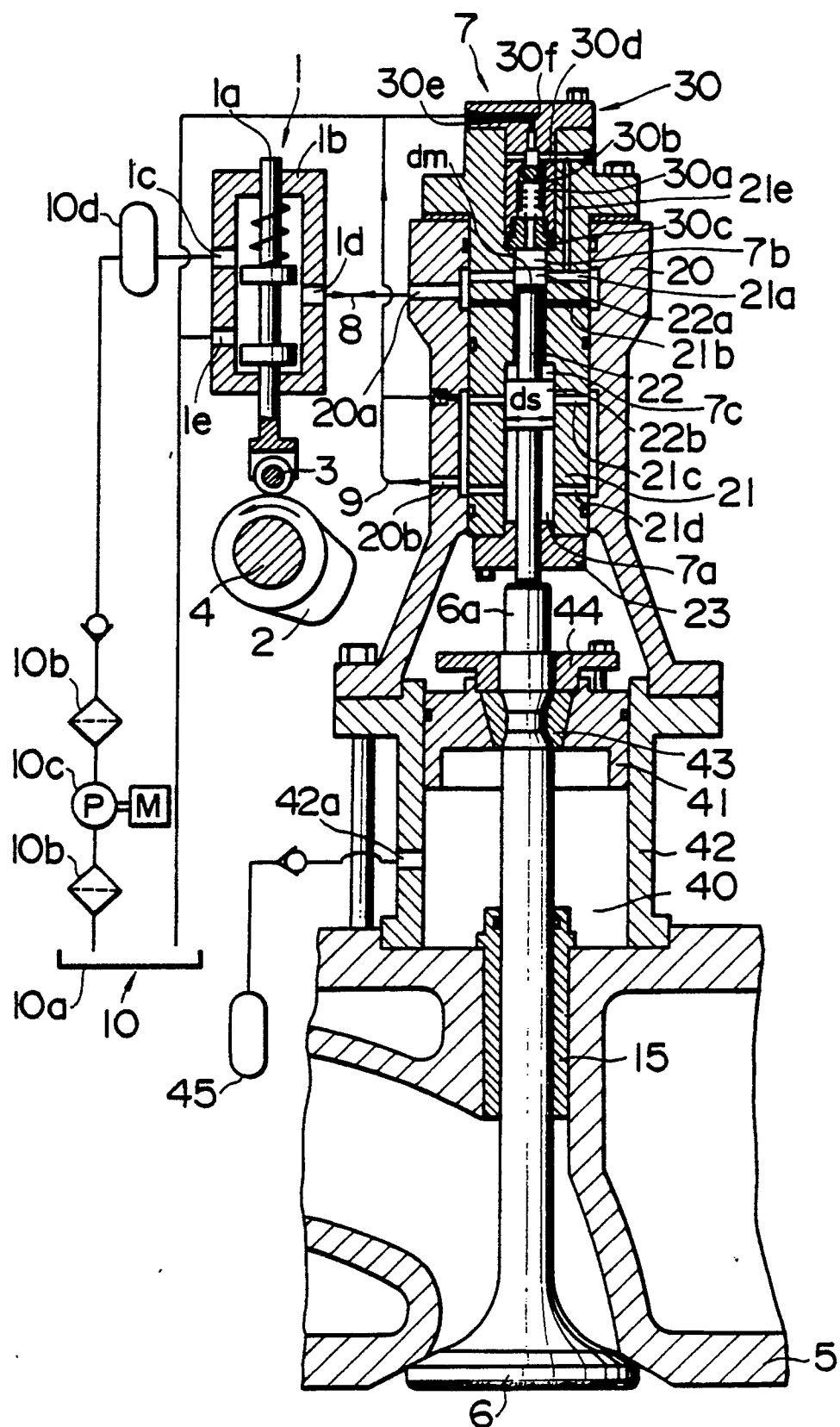




FIG. 3

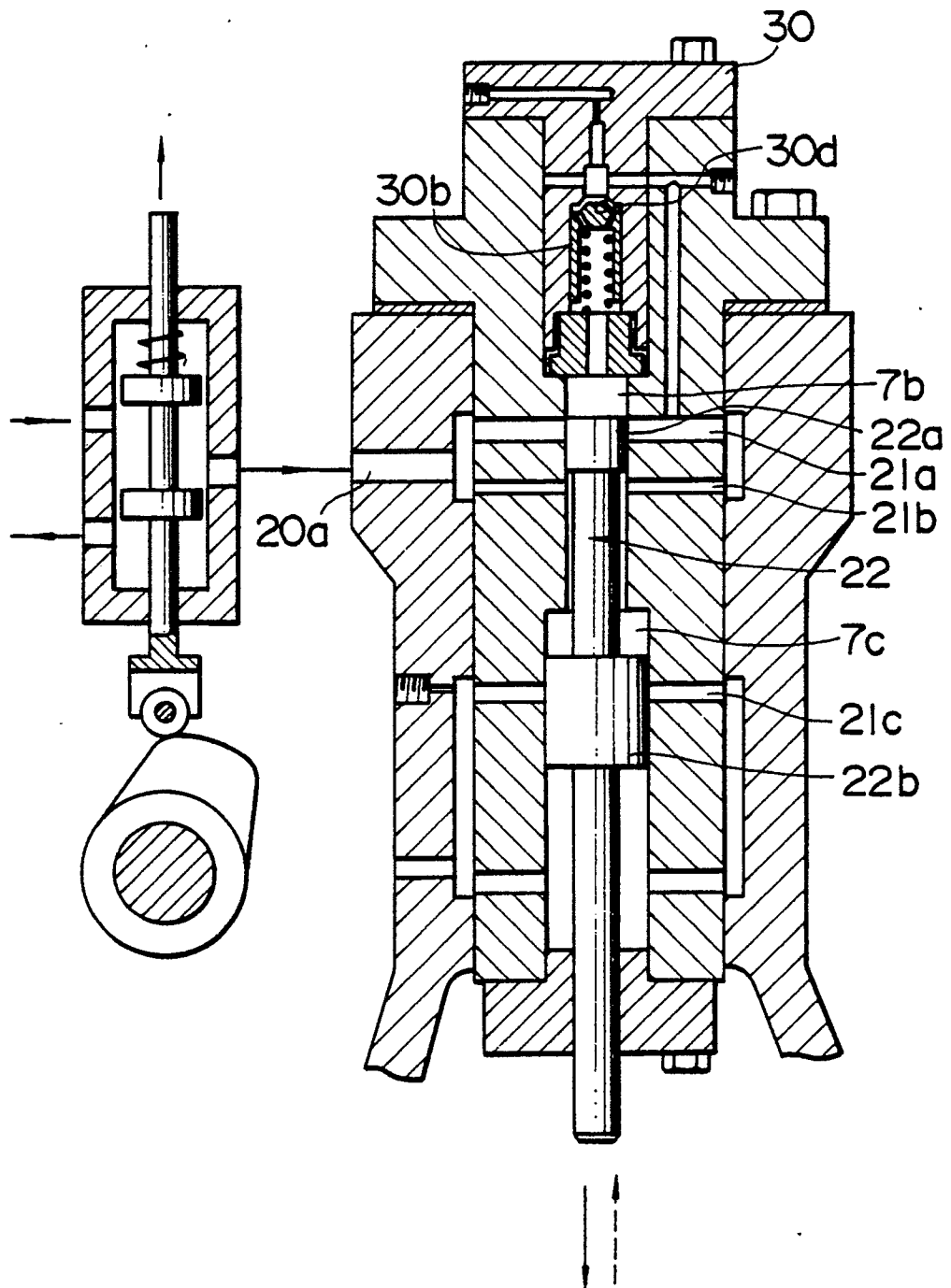






FIG. 5

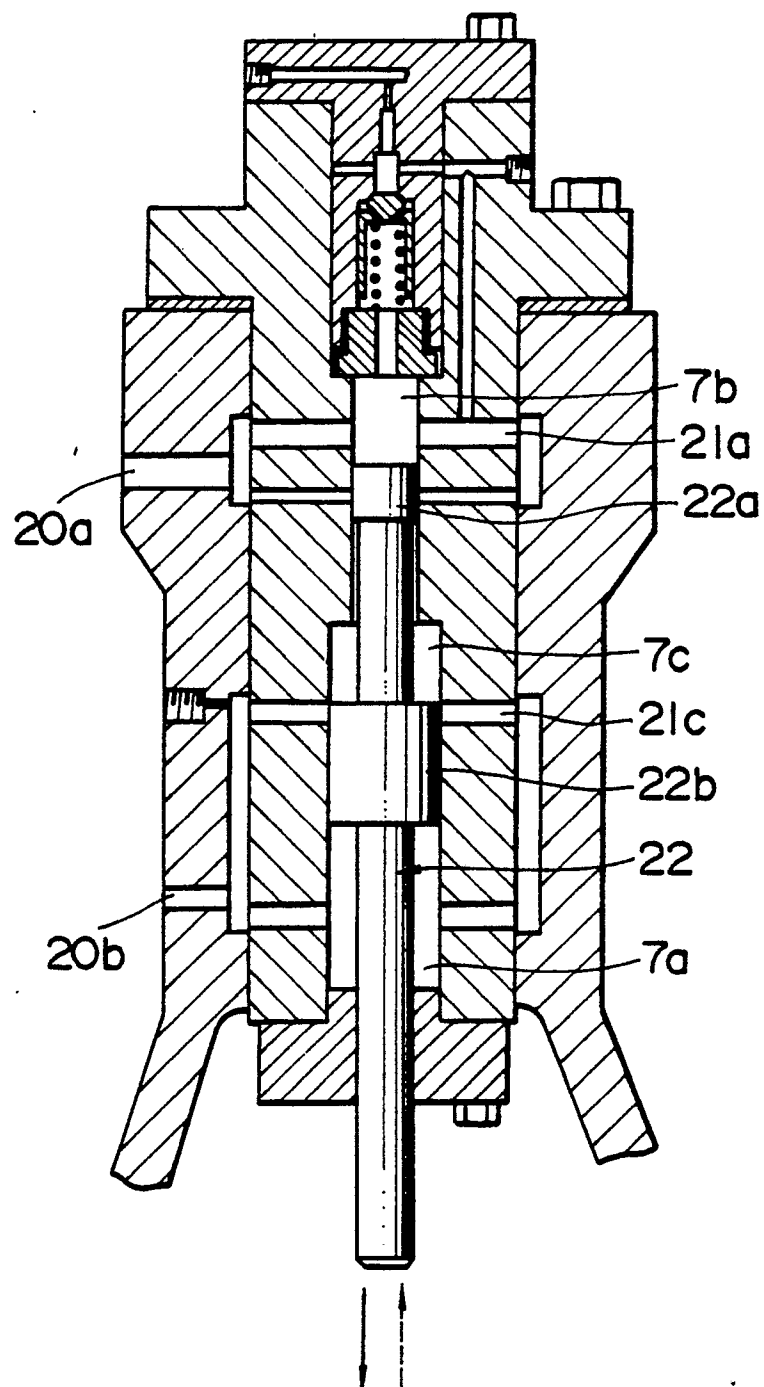


FIG. 6

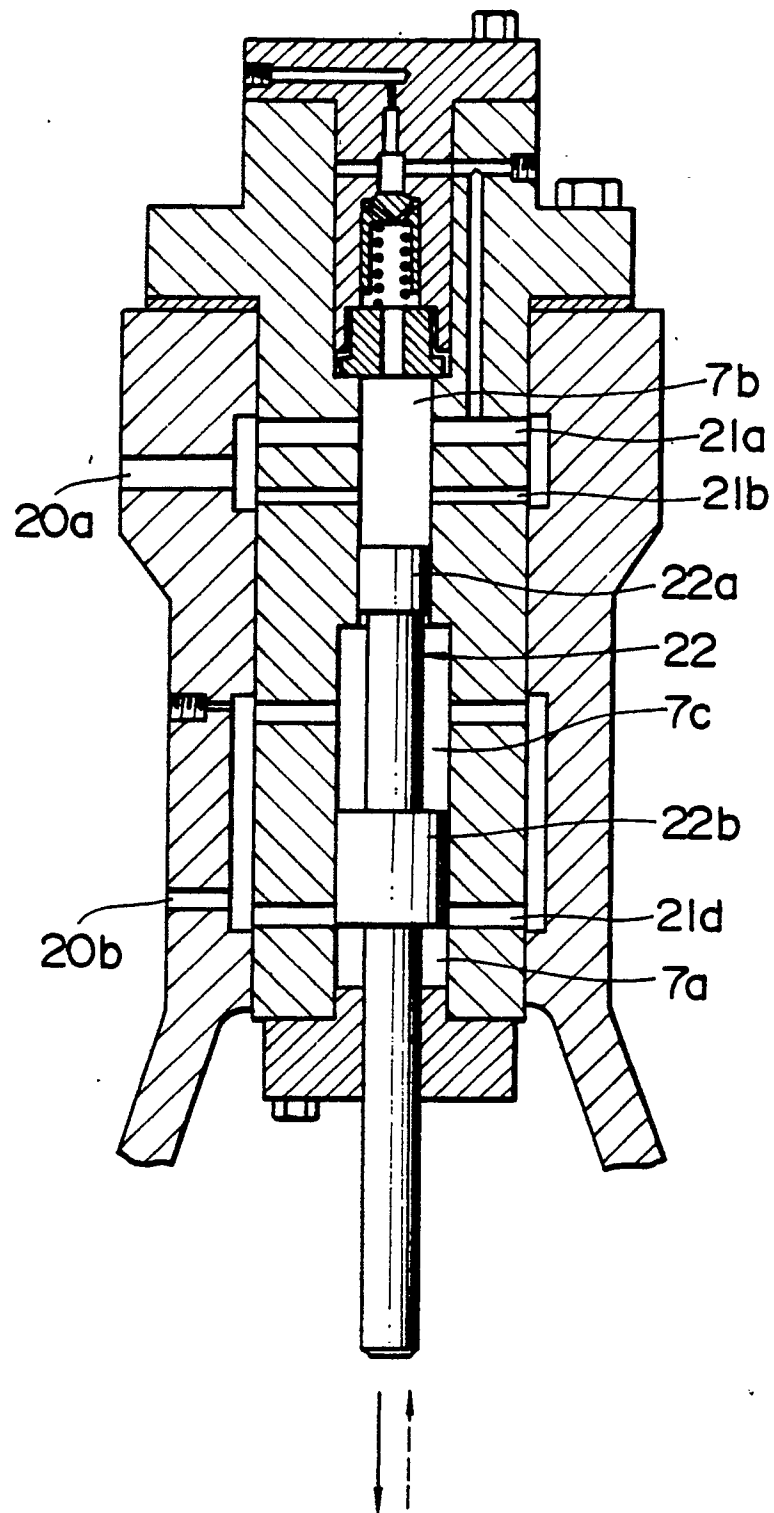


FIG. 7

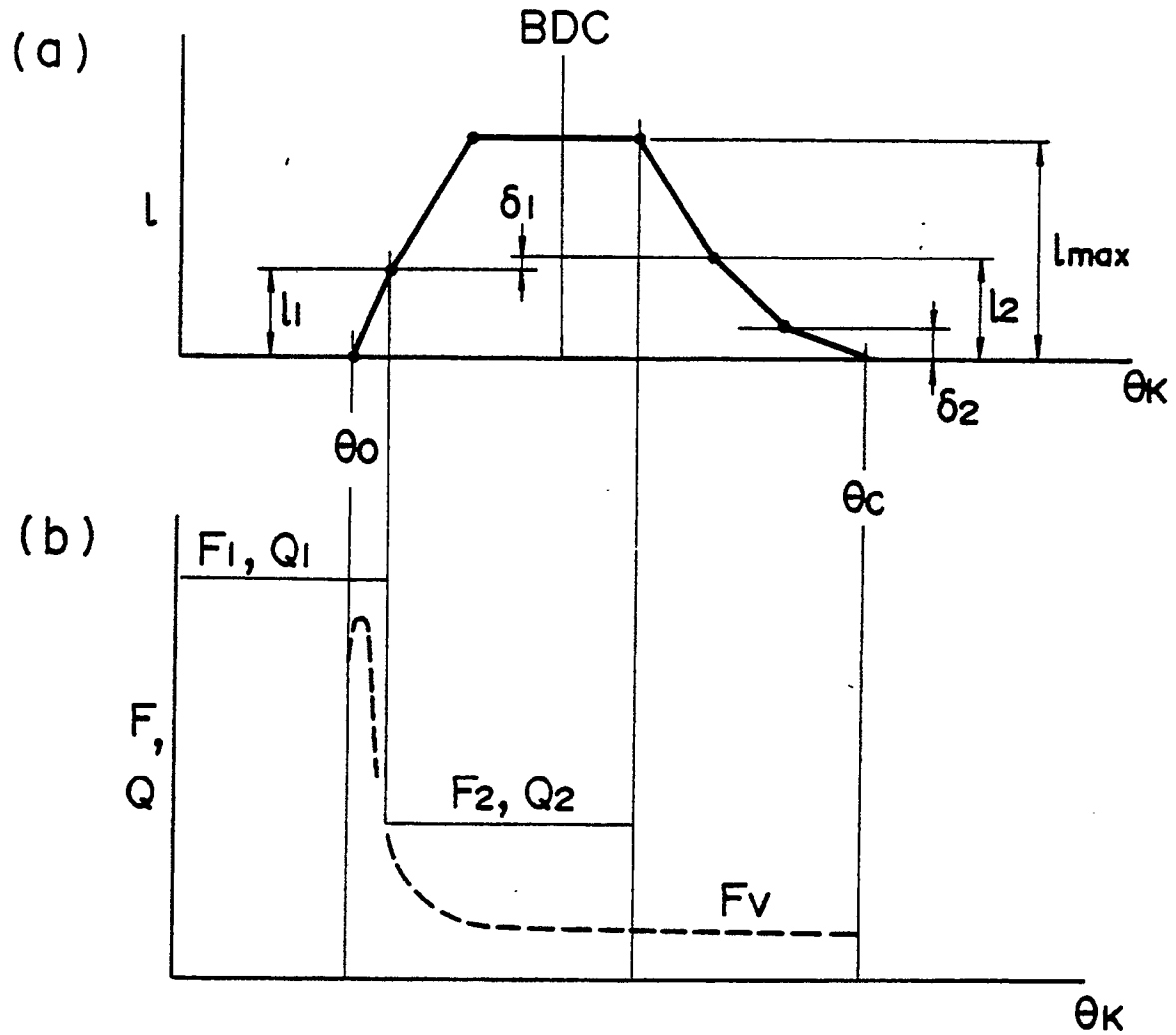
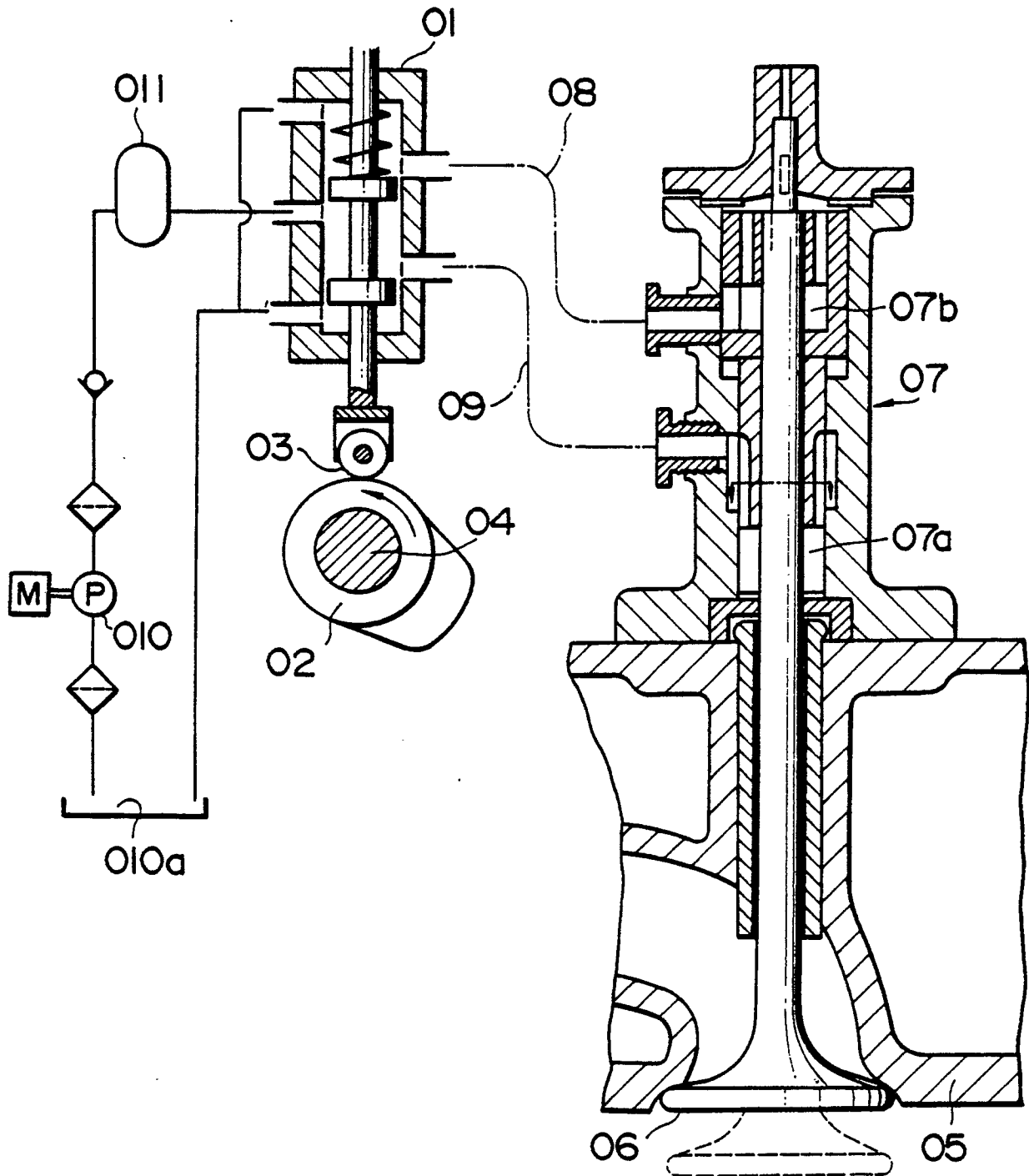
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FIG. 8





European Patent  
Office

## EUROPEAN SEARCH REPORT

Application Number

EP 90 25 0079

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
A	DE-B-1246315 (MITSUBISHI) * column 7, lines 1 - 53; figure 1 *	1	F01L9/02
A	PATENT ABSTRACTS OF JAPAN vol. 9, no. 168 (M-396)(1891) 13 July 1985, & JP-A-60 40711 (YANMAR) 04 March 1985, * the whole document *	1	
A	GB-A-2102065 (SULZER) * page 1, line 102 - page 2, line 14; figure 1 *	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl.5)
			F01L
The present search report has been drawn up for all claims			
Place of search THE HAGUE		Date of completion of the search 21 JUNE 1990	Examiner LEFEBVRE L.J.F.
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