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(56) Documents cited
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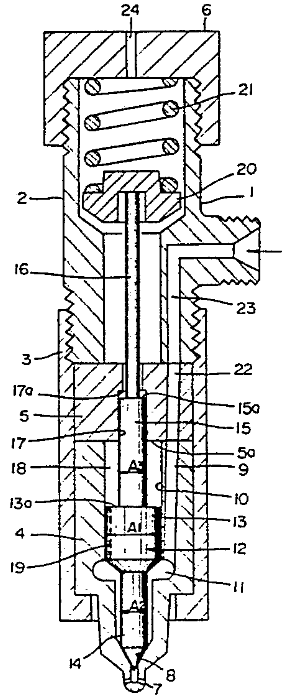
US 4635854 A

(58) Field of search
UK CL (Edition J) F1B
INT CL[4] F02M

(54) **Fuel injection valve**

(57) The surface 13a of the valve member 12 is subject to the fuel supply pressure through a throttle formed by a groove 19 in the valve member or a passage (27, Figs. 3 and 4) between the inlet passage 23 and the bore containing the push rod 16. The pressure on the surface 13a assists the spring 21 to provide valve closing. The assisting pressure may be regulated by a valve (30, Fig 4) controlling a leakage outlet (24').

Fig. I



At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

GB 2 230 559 A

Fig. 1/6

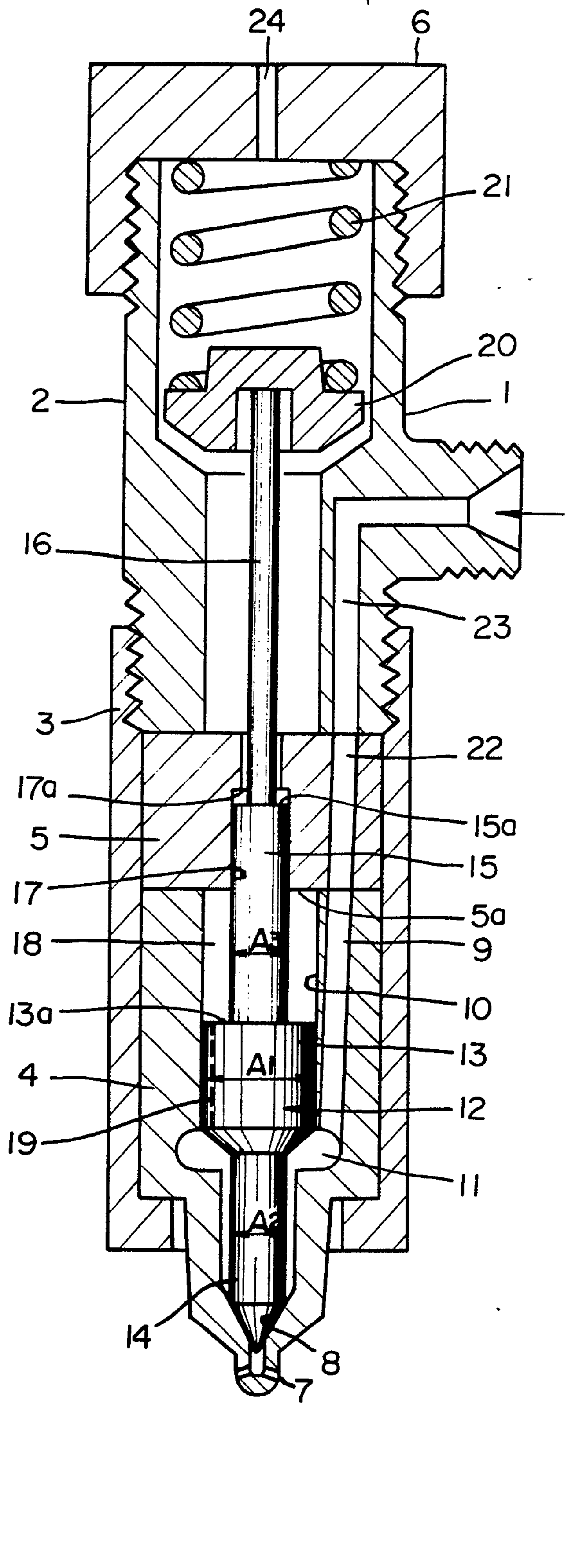
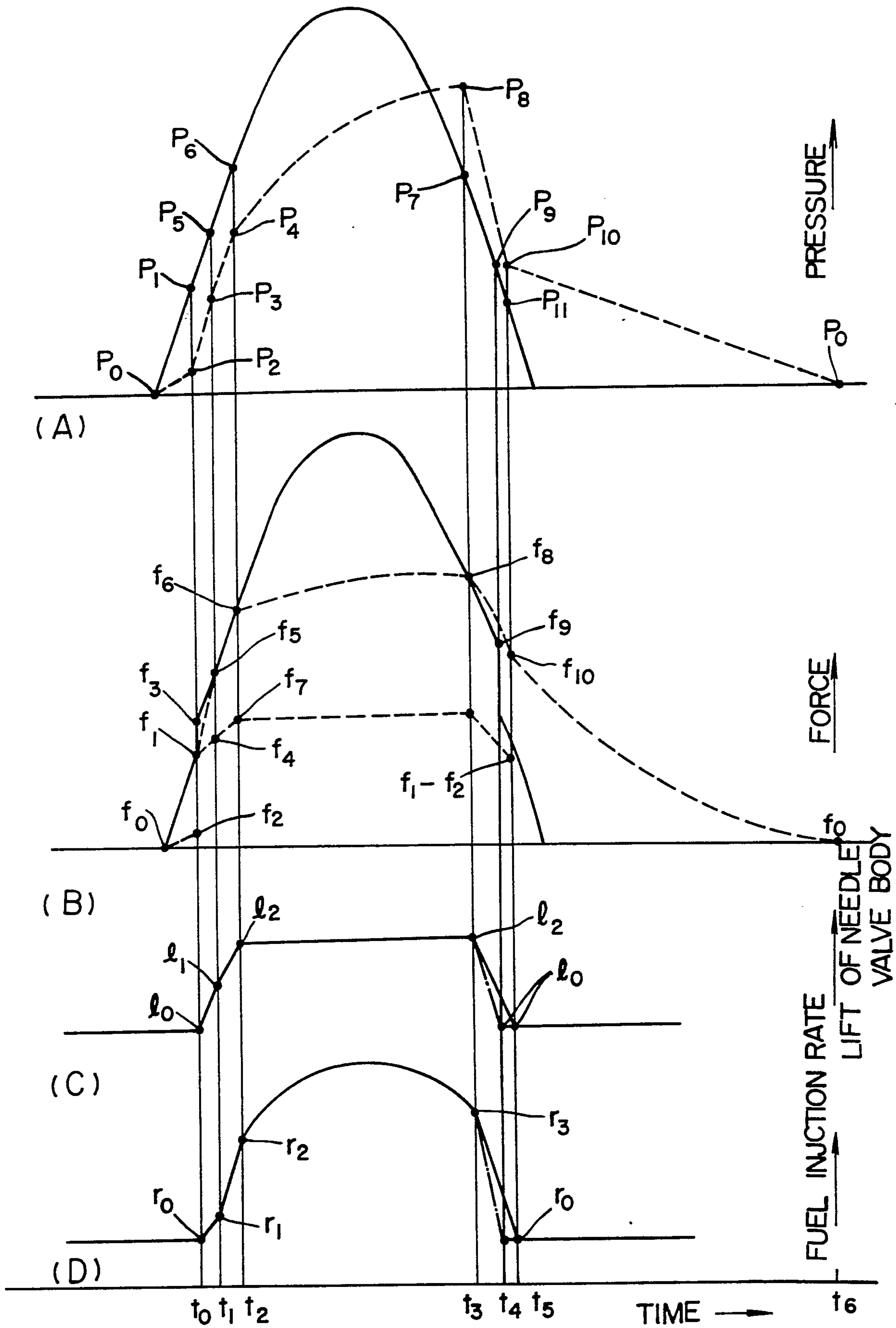

24
6
21
20
2
1
16
23
3
22
17a
15a
5
15
17
5a
18
A3
9
10
13a
13
A1
4
12
19
11
A2
14
8
7

Fig. 2/6



PRESSURE
P_0
P_1
P_2
P_3
P_4
P_5
P_6
P_7
P_8
P_9
P_{10}
P_{11}
P_0
(A)
FORCE
f_0
f_1
f_2
f_3
f_4
f_5
f_6
f_7
f_8
f_9
f_{10}
$f_1 - f_2$
f_0
(B)
LIFT OF NEEDLE VALVE BODY
l_0
l_1
l_2
l_2
l_0
(C)
FUEL INJECTION RATE
r_0
r_1
r_2
r_3
r_0
(D)
t_0 t_1 t_2
t_3 t_4 t_5
TIME
t_6

Fig. 3/6

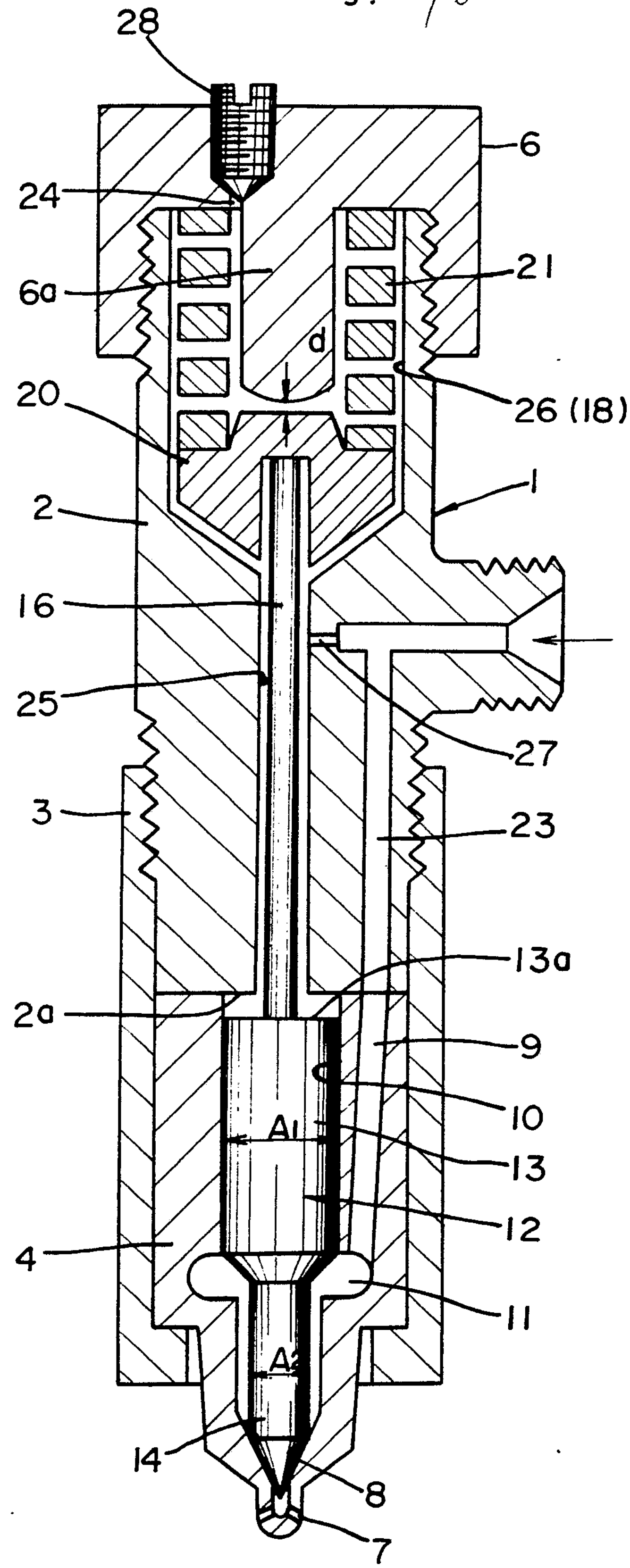

28
6
24
21
6a
d
26 (18)
20
1
2
16
25
27
3
23
13a
2a
9
10
A1
13
12
4
11
A2
14
8
7

Fig. 4/6

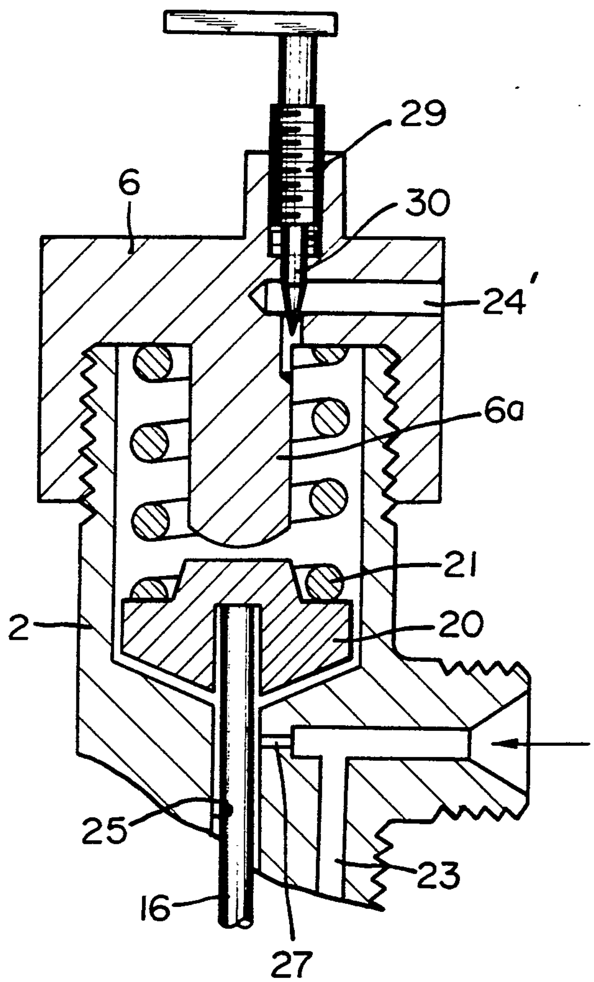
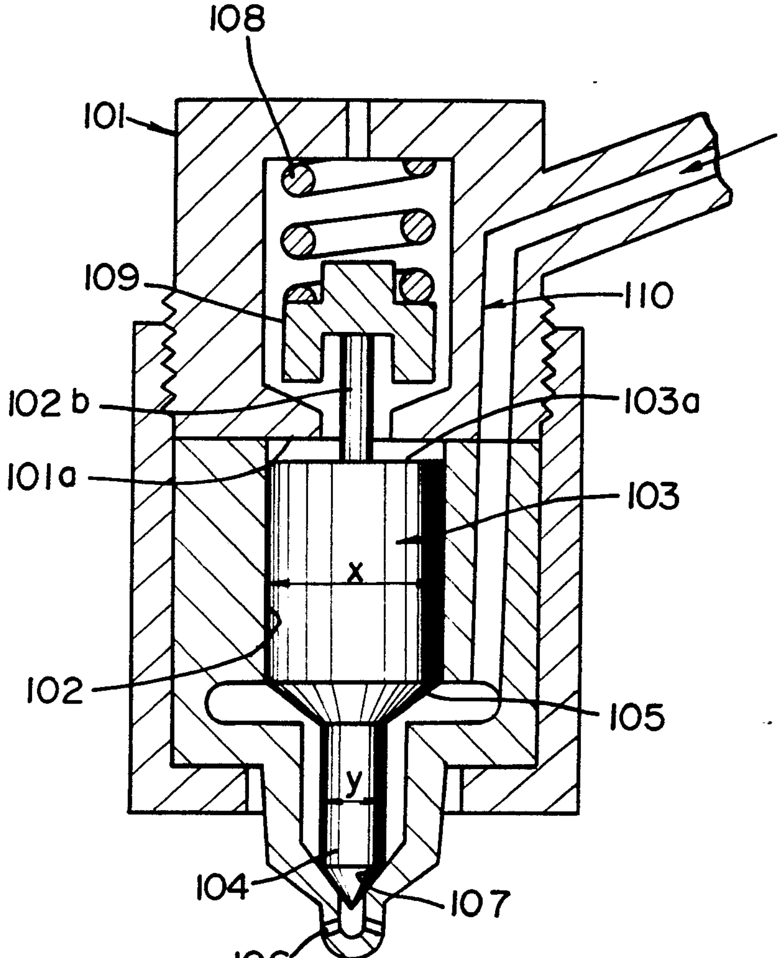
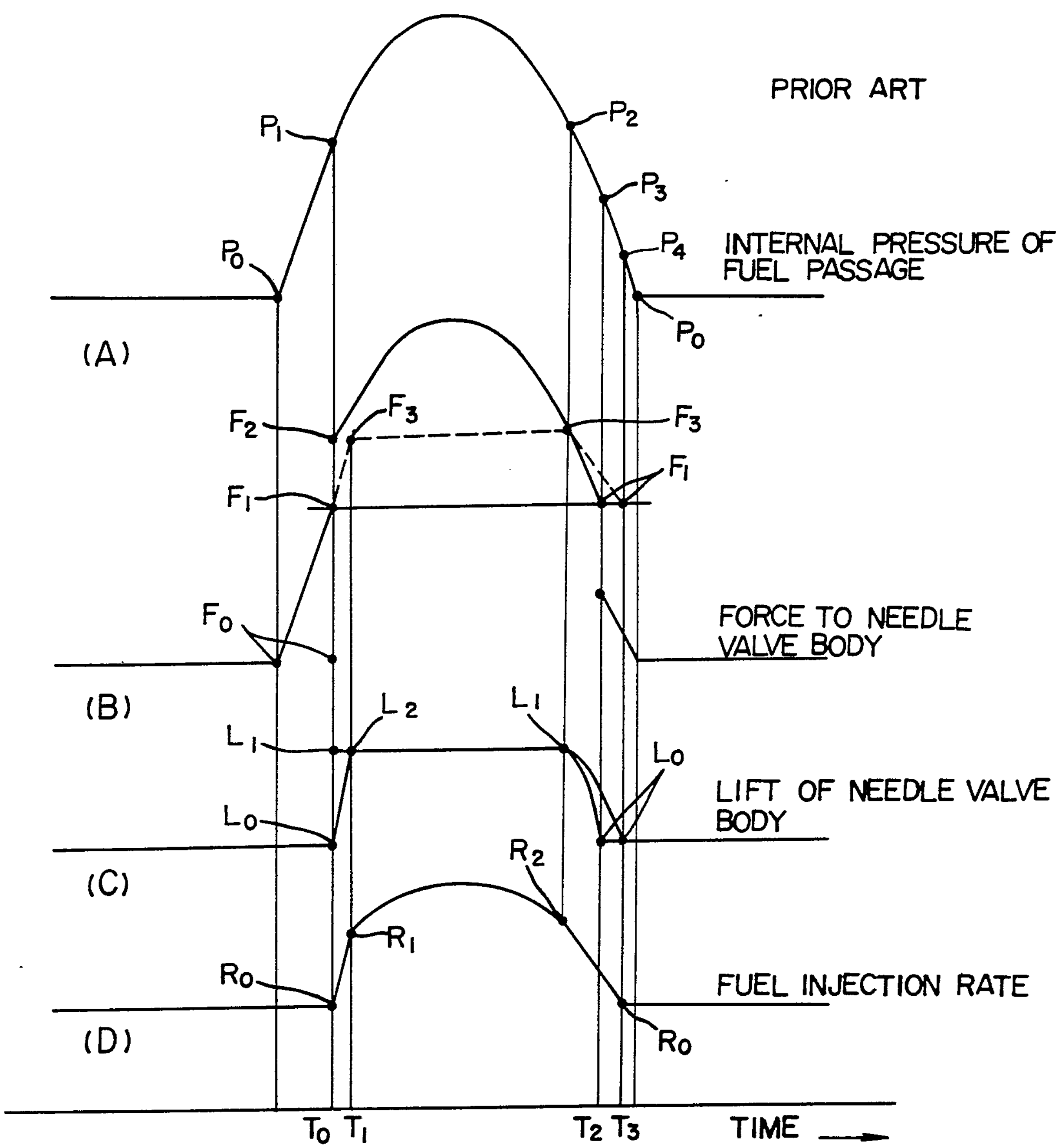


Fig. 5/6



PRIOR ART

Fig. 6 /6


PRIOR ART
P_1
P_2
P_3
P_4
P_0
INTERNAL PRESSURE OF FUEL PASSAGE
P_0
(A)
F_2
F_3
F_3
F_1
F_1
FORCE TO NEEDLE VALVE BODY
F_0
(B)
L_2
L_1
L_1
L_0
LIFT OF NEEDLE VALVE BODY
L_0
(C)
R_2
R_1
R_0
FUEL INJECTION RATE
R_0
(D)
T_0 T_1
T_2 T_3
TIME

FUEL INJECTION VALVE

The present invention relates to a fuel injection valve, more particularly, such a valve for use in a diesel engine and effective in enhancing the rate of fuel injection at the end of an injection cycle rather than at of the beginning of the cycle.

One known automatic fuel injection valve for diesel engines shown in Figure 5, includes a needle valve body 103 having a tip needle valve 104 cooperating with a valve seat 107 to control the flow of fuel to a nozzle 106. The valve body 103 is slidable in a bore 102 in the lower part of a valve body 101 and urged downwardly by a valve spring 108 through a seat 109.

Fuel under pressure P from a fuel injection pump (not shown) flows into a fuel passage 110 and acts upwardly on the needle valve body 103 over an effective area (X - Y) at the lower end 105 of the needle valve body 103. When the upward force (area (X-Y) x pressure p) exceeds the downward force of the spring 107, the needle valve body 103 moves upward lifting the needle valve 104 from its seat 107 to open the valve so that fuel oil is injected from the nozzle 106.

Once the valve is open, the effective area of the needle valve body 103 (on which pressure acts) increases from (X - Y) to (X), so that the force tending to lift the needle valve body 103, also increases, and the needle valve body 103 accelerates until an upper end 103a thereof engages an upper end 101a of the bore.

Figures 6A to 6D are graphs, Figure 6A indicating on the ordinate axis, the change of pressure a fuel oil is fed from from the fuel injection pump into the fuel passage 110, Figure 6B indicating the force tending to lift the needle valve body 103 and the force to depress the needle valve body 103, Figure 6C indicating the lift of the needle valve body 103, and Figure 6D indicating the change of the fuel injection rate, in each case plotted against time.

As the delivered fuel pressure rises from P_o toward P_1 the force tending to lift the needle valve body 103 rises from F_o to $F_1 = P_1 \times (x - y)$.

The opposing spring force is set to be F_1, so that when the pressure exceeds P_1, the needle valve body 103 ascends and the pressure P_1 is also applied to a lower surface of the needle valve 104, the force tending to lift the needle valve body 103 sharply increases to $F_2 = P_1 \times X$. This accelerates the needle valve body 103 which lifts quickly from L_o to L_1 until its upper end engages the upper end 101a of the bore 102. Here a delay between T_o and T_1, is due to the time taken to accelerate the mass of the needle valve body.

The force of the spring 108 increases from F_1 to F_3 as the needle valve body 103 lifts, however, the force tending to lift the needle valve body 103 remains larger than the spring (108) force as indicated by a full line of Figure 6B, and the needle valve body 103 retains maximum lift.

When approaching the end of fuel injection, the pressure in the passage 110 reduces to P_2, and the force tending to lift the needle valve body 103 is $F_3 = P_2 \times X$, which becomes equal to the force F_3 of the spring 108 tending to depress the needle valve body 103.

As the fuel pressure decreases, the needle valve body 103 is lifted on the force of the spring 108, and when the pressure becomes P_3 at a time T_2, the force tending to lift the needle valve body 103 becomes $F_1 = P_3 \times X$; lower than F_3. The needle valve body 103 lift becomes L_o and the needle valve closes when the fuel pressure P_3 (at the time of closing of the needle valve 104) becomes F_1/X, the pressure P_1 at the time of opening becomes lower than $F_1/X - Y$, and thus the fuel injection rate at the time of closing is reduced.

In practice, the time taken to accelerate the mass of the needle valve body 103 creates a delay so that the valve closes not at time T_2 but at T_3. During this delay the pressure in the fuel passage 110 further decreases to P_4. Accordingly, the fuel injection rate proportional to the fuel passage internal pressure is minimised inevitably at the end of fuel injection as shown in Figure 6D.

However, fuel injected at a high rate at the beginning of injection is burned quickly in the combustion chamber of the diesel engine to raise pressure, cause combustion noise or so-called diesel knock, raise the combustion maximum pressure raise the combustion temperature accordingly, thus often producing noxious NOx.

Further, deterioration of the fuel injection rate at the end of the injection period and the resultant increase in fuel injection time, and the coarse fuel spray caused by a decrease in injection pressure are capable of causing so-called after-burning, and thus the imperfect combustion. This may not only cause noxious black smoke (CO, HC) but also reduce the thermal efficiency.

The acceleration delay might be shortened by lessening the mass of the needle valve body 103 or increasing an apparent spring constant by using an upper portion of the needle valve body 103 as a pressure accumulating chamber. However, such measures have not yet been made effective in improving performance stability.

One object of the present invention is to ensure a low initial fuel injection rate and to enhance the fuel injection rate toward the end of an injection cycle to shorten the fuel injection period, thereby decreasing NOx, CO and HC and enhancing thermal efficiency.

In accordance with the present invention, we propose a fuel injection valve having a seated valve for controlling the flow of fuel under pressure from a fuel supply passage to a fuel injection nozzle wherein the valve is operable by a plunger biased toward a valve closed position, and urged in the valve closing direction by the pressure acting in a pressure accumulating chamber in communication via a throttle with the fuel supply passage.

A preferred embodiment of fuel injection valve comprises a needle valve body serving as the plunger, with a needle valve for co-operation with a valve seat near a nozzle hole. The needle valve body runs in a bore and the space within the bore on one side of the plunger (opposite to the needle valve), forms at least part of the pressure accumulating chamber. A fuel passage communicating with valve seat also communicates with the pressure accumulating chamber through a throttle which may be in the form of an axial groove extending axially of the plunger.

When the valve opens as the pressure in the fuel passage rises at the beginning of fuel injection, fuel in the pressure accumulating chamber is compressed as the needle valve body lifts. This reduces the opening velocity of the needle valve body and the fuel injection rate is lowered. Thus, the heat generated in an engine combustion chamber during the initial stage of combustion is retarded, and the combustion pressure is kept from rising.

When the valve closes at the end of a fuel injection cycle, pressure in the accumulating chamber is kept high by the fuel passage internal pressure through the throttles which is applied to an upper end of the needle valve body, the needle valve of the needle valve body contacts with valve seat in co-operation with the spring, thus closing the valve. The fuel passage internal pressure at the time the valve begins to close is higher than that at the end of opening, and due to the increase in the fuel passage internal pressure at the end of the valve closing, the particle size of the oil-mist injected from the nozzle at the end of fuel injection is minimised, and combustion is improved.

Further, high pressure injection at the time of valve closing enhances the fuel injection rate sufficiently to prevent after-burning, decrease the exhaust of black smoke, CO and HC, and further to enhance a constant volume of Sabathe cycle, thus improving the thermal efficiency of the diesel engine.

Preferred embodiments of the invention will now be described by way of example with reference to the accompanying drawings in which:

Figure 1 is a longitudinal sectional view of a fuel injection valve;

Figures 2A to D are performance curves for the valve of Figure 1;

Figure 3 is a longitudinal sectional view of another fuel injection valve;

Figure 4 is a fragmentary longitudinal sectional view of another fuel injection valve;

Figure 5 is a longitudinal sectional view of a conventional fuel injection valve; and

Figures 6A to D are performance curves for the valve of Figure 6.

With reference to Figure 1, a fuel injection valve body 1 comprises a valve body upper portion 2, and a valve body lower portion 4 held together with a pressure accumulating chamber cover 5 therebetween by a nut 3 screwed onto the upper body portion 2, the upper end of which has screwed thereon a stopper 6.

Within a nose on the lower body portion 4 protruding from the nut 3 is formed a valve seat 8 and downstream thereof a nozzle 7. A needle valve body 12 co-operating with the seat 8 is slidable within a bore 10 formed centrally of the lower body portion 4 and opening into an application chamber 11 upstream of the valve seat 8.

The needle valve body 12 carries a needle valve 14 having a sectional area A_2 for contact with the valve seat 8, the body or sliding part 13 having a sectional area A_1, and a smaller diameter part 15 to which a pushrod 16 is connected having a sectional area A_3.

A stop hole 17 formed in the pressure accumulating chamber cover 5, co-axial with and communicating with the bore 10 in the lower body portion 4, receives the smaller diameter body part 15. An annular pressure accumulating chamber 18 defined between the upper face 13a of the sliding part 13 of the needle valve body 12 and the lower face 5a of the cover 5, communicates via a throttle groove 19 formed on a side of the sliding part 13, with the application chamber 11.

Between the stopper 6 and a spring seat 20 carried by the push rod 16 is a spring 21 which biases the needle valve body 12 downward toward the valve closed position (with the needle valve 14 in contact with the valve seat 8).

A fuel passage 22 through the pressure accumulating chamber cover 5, communicates with the fuel passage 9 in the valve body upper portion 4 and a fuel inlet passage 23 in upper body portion 2, for connection to a fuel injection pump (not shown). A leak hole 24 is provided in the stopper 6.

When fuel from a fuel injection pump (not shown) flows into the fuel passages 23, 22 and 9, pressure in the passage begins to rise from P_o (Figure 2), the pressure in the pressure accumulating chamber 18 communicating with the application chamber 11 through the throttle groove 19 also being P_o, and the rate of pressure rise in the pressure accumulating chamber 18 is low (P_o - P_2; dotted line in Figure 2).

When the internal fuel pressure reaches P_1, the needle valve 14 is ready for opening. The force $f_1 = P_1 \times (A_1 - A_3)$ shown in Figure 2B acts against the spring force $f_1 - f_2$ and an internal pressure $f_2 = P_2 \times (A_1 - A_3)$ of the pressure accumulating chamber 18, and when it exceeds P_1, the needle valve 14 parts from the valve seat 8 to open the valve, and fuel injection (from the nozzle hole 7) commences.

At the instant the valve opens, the effective area of the needle valve body 12 on which pressure acts increases to A_1 from $(A_1 - A_2)$, and thus the force tending to lift the needle valve body 12 increases to $f_3 = P_1 \times A_1$ (Figure 2B), so that ascent of the needle valve body 12 is accelerated by the force f_3.

The valve lift increases from ℓ_o to ℓ_1 during the period t_o to t_1.

As the lift increases, fuel in the pressure accumulating chamber 18 is compressed, the pressure increasing successively along the curve P_2 to P_3 to P_4 (Figure 2A), $P_3 \times (A_1 - A_3)$ plus the force of the spring 21 becomes $f_4 = f_5 = P_5 \times A_1$ at the pressure P_3 midway of the lift (time t_1) , and further lift of the needle valve body 12 is limited by the pressure rise in the pressure accumulating chamber 18. The lift of the needle valve body 12 at time t_1 is ℓ_1 (Figure 2C), and since the pressure in the fuel passage and the lift of the needle valve body 12 are small, the initial fuel injection rate is low as indicated by $r_o - r_1$ in Figure 2D.

When the pressure increases, the fuel in the pressure accumulating chamber 18 and the spring 21 are further compressed until $P_6 \times A_1 = f_6 = P_4 \times (A_1 - A_3)$ + force f_7 of the spring 21 at time t_2, and when the pressure in the fuel passage exceeds P_6, an upper end 15a of the small diameter part 15 abuts the upper end 17a of the stop hole 17, to determine the maximum lift (ℓ_2 in Figure 2C).

As shown in Figure 2D, the fuel injection rate increases from r_1 to r_2 in the period $t_1 - t_2$, but remains low as compared with the prior art fuel injection valves.

As the pressure in the fuel passage increases further as shown in Figure 2A, the force tending to open the needle valve 14 also increases as shown in Figure 2B, and the needle valve 14 remains open. However, high-pressure fuel in the application chamber 11 of the fuel passage 9 is fed into the pressure accumulating chamber 18 through throttle groove 19, so that the pressure in the pressure accumulating chamber 18 continues to rise, even after the pressure in the fuel passage begins to decrease as the fuel injection end approaches.

The force tending to open the needle valve 14 is balanced by the force tending to close the valve at time t_3; the force tending to open the valve being given by $f_8 = P_7 \times A_1$, and the force tending to close the valve being $P_8 \times (A_1 - A_3)$ plus the force of the spring 21 at the time of full lift $f_7 = f_8$. Accordingly, as the pressure in the fuel passage decreases further so the lift of the needle valve body 12 decreases. The valve-closing start pressure P_7 is very high as compared with the known prior art for the internal pressure P_8 of the pressure accumulating chamber 18.

Since the pressure in the accumulating chamber 18 and the spring 21 depress the needle valve body 12 continuously the volume of the pressure accumulating chamber 18 increases, the pressure in the accumulating chamber 18 decreases, and the force tending to open the needle valve 14 also decreases.

The needle valve 14 closes at t_4, when the force tending to open the needle valve 14 $P_9 \times A_1 = f_9$ balances the force tending to close the valve $P_{10} \times (A_1 - A_3)$ plus the force of the spring 21 at the time of full closing $f_1 - f_2 = f_9$, however there is a delay ($t_4 - t_5$) due to the time taken to accelerate the mass of the needle valve 14, and thus the valve closes at a point t_5 in time. The pressure in the fuel passage then drops as low as P_{11}. However, the valve-closing start pressure P_7 and the valve-closing end pressure P_{11} are very high as compared with conventional injection valves, by virtue of providing the pressure accumulating chamber 18 and the throttle groove 19, the effect being not only minimisation of the particle size of the fuel mist during the fuel injection end period but also an enhancement of the fuel injection rate.

After the end of fuel injection, the pressure of fuel in the accumulating chamber 18 continues to a drop as shown in Figure 2A, to the pressure P_0 of fuel in the fuel passage - the pressure before the fuel injection starts at time t_6 - in readiness for the next fuel injection cycle.

Other embodiments shown in Figures 3 and 4 (wherein equivalent reference numerals are used) have no pressure accumulating chamber cover 5 and the needle valve body 12 does not have a small diameter part equivalent to part 15 in Figure 1.

The valve body upper portion 2 is in direct contact with the valve body lower portion 4 and is coupled thereto by the nut 3. The pressure accumulating chamber 18 is formed by the space between an upper surface of the sliding part 13 of the needle valve body 12 and a lower surface 2a of the valve body upper portion 2, the annular space, the pushrod 16 and a hole 25 through which it passes and of a chamber 26 containing a square section spring 25 and the spring seat 20.

Also, instead of the throttle groove 19 in Figure 1, a throttle 27 is formed between the fuel passage 23 and the insertion hole 25.

An air vent valve 28 is screwed into the leak hole 24 formed on the stopper 6 which has a central projection 6a forming an abutment determining the maximum lift, the clearance d being smaller than the distance between the upper surface 13a of the sliding part 13 of the needle valve body 12 and the lower surface 2a of the valve body upper portion 2 when the needle valve 14 is closed. Thus, the upper end 13a of the needle valve body 12 is always subject to the pressure of fuel in the pressure accumulating chamber 18, even at maximum needle valve 14 lift.

The embodiment of Figure 4 is basically similar to that of Figure 3 except that the spring 21 is circular in section and that a needle valve 30 movable by a screw 29 is provided for adjusting the effective flow area of the leak hole 24'.

In the embodiments of Figure 3 and Figure 4 when fuel oil is fed from the fuel injection pump and the pressure in the fuel passage rises from P_o to P_1, fuel flows through the throttle 27 and the pressure in the pressure accumulating chamber 18 also rises from P_o to P_2 as described above with reference to Figure 2.

The force tending to open the valve is $f_1 = P_1 \times (A_1 - A_2)$ as shown in Figure 2B, is opposed by the pressure in the pressure accumulating chamber 18 acting on the upper surface of the sliding part 13 ($P_2 \times A_1$) and a spring force $f_1 - f_2$. When the pressure in the fuel passage exceeds P_1, the needle valve 14 lifts from the valve seat 8, and fuel injection from the nozzle hole 7 begins.

Instantaneously, the effective (pressure receiving area) of the needle valve body 12 increases from ($A_1 - A_2$) to A_1, so that the force tending to lift the needle valve body 12 increases to $f_3 = P_1 \times A_1$ (Figure 2B) and as the needle valve body 12 lifts it is accelerated by the force f_3.

As lift of the needle valve body 12 increases, so fuel in the pressure accumulating chamber 18 is compressed to increase the pressure as shown in Figure 2A. When at half lift the pressure reaches P_3 (at t_1), $f_4 = f_5$ of $P_3 \times A_1$ + force of the spring 21, is balanced by the force tending to open the needle valve 14 ($P_5 \times A_1$) and an increase in the lift of the needle valve body 12 is limited.

When the pressure in the fuel passage rises to exceed P_5, the fuel in the pressure accumulating chamber 18 and the spring 21 are further compressed ($P_5 \times A_1 =$

$f_6 = P_4 \times A_1$ - force f_7 of the spring 21 at time t_2), until an upper end of the spring seat 20 comes in contact with the projection 6a at maximum lift (ℓ_2 of Figure 2C).

The pressure in the fuel passage then increases as shown in Figure 2A, so that force tending to open the needle valve 14 also increases as shown in Figure 2B, and the needle valve 14 remains opening. However, high-pressure fuel in the fuel passages 23, 22, 9 is fed into the pressure accumulating chamber 18 through the throttle 27, so maintaining a high pressure in the pressure accumulating chamber 18.

Even after the fuel passage internal pressure begins to drop as the end of fuel injection approaches the pressure in the pressure accumulating chamber 18 continues to rise. At time t_3 the force $f_8 = P_7 \times A_1$ tending to open the needle valve 14 is balanced by the force tending to close the valve ($P_8 \times A_1$ + force of the spring 21 at the ime of full lift $f_7 = f_8$), and as the pressure in the fuel passage drops thereafter so the lift of the needle valve body 12 reduces. Thus the spring 21 depresses the needle valve body 12, increasing the volume of the pressure accumulating chamber 18 and decreasing the pressure therein. The force tending to open the needle valve 14 also continues to reduce.

The needle valve 14 is to close at time t_4 when the force to open the needle valve 14 ($P_3 \times A_1 = f_9$) is balanced by the force to close ($P_{10} \times A_1$ plus force of the spring 21 at the time of full close $f_1 - f_2 = f_9$), however, a delay arises due to the time taken to accelerate the mass of the needle valve 14, and the valve in fact, closes at time t_5.

After the end of fuel injection, fuel in the pressure accumulating chamber 18 continues to flow out through the throttle 27 to equalise with the pressure in the fuel passage and as shown in Figure 2A, it drops to P_o at time t_6.

In the embodimentof Figure 4 pressure in the pressure accumulating chamber 18 can be regulated by the needle valve 30 thus enabling adjustment of the fuel injection valve according to the operating state of an engine.

As described above, the invention comprises disposing a needle valve body with a needle valve coming in contact with valve seat nearby a nozzle hole and its upper end depressed downward by a valve spring in a sliding hole perforated in a valve body lower portion, forming a pressure accumulating chamber over the needle valve body, leading fuel passage communicating with the valve seat into the pressure accumulating chamber through a throttle, therefore a fuel injection rate at a fuel injection beginning will be moderated, thus a heat generation is suppressed to combustion in a diesel engine combustion chamber, a combustion pressure is kept from rising and a noise is also suppressed, and NOx will be less produced.

Further, the invention has advantages in that the fuel injection rate at the end of an injection cycle is enhanced, the fuel injection period is shortened at the same time, after-burning in the combustion of diesel engine is avoided, there is a decrease in the black smoke, CO and HC exhausted and an improvement in thermal efficiency. Furthermore, the fuel injection valve of the invention is simple in structure and obtainable at a moderate cost.

CLAIMS

1. A fuel injection valve having a seated valve for controlling the flow of fuel under pressure from a fuel supply passage to a fuel injection nozzle, wherein the valve is operable by a plunger biased toward a valve closed position, and urged in the valve closing direction by the pressure acting in a pressure accumulating chamber in communication via a throttle with the fuel supply passage.

2. A valve according to claim 1 wherein the arrangement is such that in use, fuel supply pressure acts on one side of the plunger in the sense to open the seated valve and in opposition to bias, and that the pressure in the pressure accumulating chamber acts on the other side of the plunger.

3. A valve according to claim 1 or claim 2 wherein the throttle is in the form of a groove extending axially of the plunger to communicate between opposite sides thereof.

4. A valve according to any one of claims 1 to 3 comprising a needle valve cooperating with a valve seat and carried by a needle valve body that is slidable in a bore to form the said plunger, the needle valve being smaller in diameter than the body and extending into a chamber communicating with the fuel supply passage upstream of the valve seat.

5. A valve according to any one of the preceding claims wherein the space within the bore on a side of the plunger opposite to the valve forms at least a part of the pressure accumulating chamber.

6. A valve according to any one of the preceding claims comprising an abutment limiting maximum lift of the valve.

7. A fuel injection valve constructed and arranged substantially as hereinbefore described with reference to and as illustrated in Figures 1 and 2, Figure 3 and/or Figure 4 of the accompanying drawings.

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