

A fuel injector for an internal combustion engine

The present invention relates to a fuel injector for an internal combustion engine, particularly a large two-stroke diesel engine, having an external housing for mounting in a cylinder cover and a through-going fuel passage opening out into a nozzle, and having a valve slider which is longitudinally displaceable in a slider guide and is acted upon in a direction towards its valve seat by a preloaded compression spring and in the opposite direction by the pressure of the fuel in the fuel passage, a first piston, which is axially displaceable in the housing, being located at one end of the compression spring and having a first surface which faces away from the spring and, together with a stationary component, delimits a first pressure chamber which communicates with the fuel passage through a channel, said compression spring biasing the first piston in a direction towards an extreme position with a minimum fuel volume in the first pressure chamber.

In such an injector described in Danish patent No. 152 619, corresponding to Japanese patent No. 1851989, the channel from the fuel passage to the first pressure chamber is designed as a throttle channel so that the pressure in the chamber follows the current pressure in the fuel passage with a delay. When the fuel pressure increases during an injection period, the pressure also increases in the pressure chamber so that the first piston is pressed towards the closing spring and increases the force with which the spring acts on the valve slider in a direction towards its seat, which increases the closing pressure of the injector.

This remedies the problem that the closing pressure, i.e. the pressure at which the valve slider moves towards its seat at the end of the delivery period, is

usually smaller than the opening pressure, i.e. the pressure in the fuel passage at which the valve slider is lifted off its valve seat at the beginning of a delivery period. The lower closing pressure is due to
5 the fact that in the closed position of the injector, the fuel pressure acts on an effective area of the valve slider which is smaller than when the valve slider is in its open position, when the pressure also acts on the slider area below the seat surface.

10 In the injector described in the Danish patent, the first piston makes continuous adjustment movements during and immediately following the delivery period of the injector, which may result in a not inconsiderable wear on the guide surface of the piston with consequent
15 increased leakage from the pressure chamber. After the end of each delivery period, the compression spring presses the first piston back to its extreme position with a minimum fuel volume in the pressure chamber, so that the opening pressure of the injector is not
20 influenced by the hydraulic loading of the compression spring.

It is known that the compression pressure in the engine cylinder depends on the engine load so that at full loads the pressure is substantially higher than at
25 low loads. At full load, for example, the compression pressure may be at about 120 bar, while at idling load the compression pressure is at about 40 bar.

When the valve slider is in its closed position, the pressure in the engine cylinder spreads through the
30 nozzle holes and further along to the slider area below the seat surface, i.e. the slider section positioned at the nozzle side of the valve seat. Therefore, the current compression pressure acts on the valve slider with a force in the opening direction. The compression
35 pressure rising with the engine load thus leads to a

drop in the opening pressure of the known injectors at increased engine loads. In a typical fuel injector for a large two-stroke diesel engine, the opening pressure may, for example, drop from 400 bar at idling load to
5 325 bar at full engine load. The lower opening pressure at full load does not promote the atomization of the fuel at the start of the injection period.

At low engine loads, the fuel pressure in the injector is determined by its opening pressure, as the
10 fuel pumps supply such a small amount of fuel that the flow resistance in the nozzle has no influence on the fuel pressure. Contrary to this, the fuel amounts delivered from the pumps at higher engine loads are so large that the flow resistance in the nozzle becomes
15 decisive to the fuel pressure in the injector, i.e. the fuel pressure is in this case substantially higher than the opening pressure of the injector.

The opening pressure in the known injectors is determined by the preloading of the compression spring.
20 The manufacture of the springs is subject to certain production tolerances, resulting in the fact that the fuel injectors in an internal combustion engine are not necessarily all set to exactly the same opening pressure. The variations in the opening pressures of the
25 injectors often become more pronounced after a long period of operation of the engine, as the springs sag during operation, i.e. lose some of their spring force. Thus, there are time-determined changes in the opening pressures of the injectors, which necessitates control
30 and retightening of the compression springs at regular intervals to maintain satisfactory operation of the engine. This is labour-consuming and undesirable.

The object of the invention is to provide a fuel injector which has an increased opening pressure at

increasing engine loads, and which requires less maintenance.

With this object in view, the fuel injector according to the invention is characterized in that the spring is in force-transmitting connection with a second
5 piston having a second surface which faces away from the spring and constitutes the end wall in a second pressure chamber, that at movement in a direction away from said extreme position the first piston opens a flow connection between the first and the second pressure chambers,
10 that the second pressure chamber has a larger effective cross-sectional area than the first chamber, and that a restricted drain passage connects the second pressure chamber with a drain.

At increasing engine loads, as mentioned above, the
15 pressure grows in the fuel passage as a consequence of the flow resistance in the nozzle. Thus, also the pressure in the first pressure chamber increases, which causes displacement of the first piston so that the flow connection between the first and the second pressure
20 chambers is opened, and an amount of fuel flows into the second pressure chamber. As this pressure chamber has a larger effective cross-sectional area than the first chamber, the pressure build-up in the second chamber will lead to a return of the first piston to its extreme
25 position with a minimum fuel volume, and at the same time the amount of fuel in the second pressure chamber is confined, as the first piston interrupts the flow connection between the chambers. As the second piston is in force-transmitting connection with the spring, the
30 latter will be shortened in step with the filling of the chambers, whereby the spring force increases. Apart from a small amount of fuel drained away, the fuel is kept confined in the pressure chamber, until the fuel injector again has to be actuated for renewed injection
35 of fuel, and thus, the increased force of the spring is

maintained, which imparts to the fuel injector both a higher opening pressure and a higher closing pressure.

If the highest pressure in the fuel passage during the next injection period is higher because the engine load increases, the fuel pressure in the first pressure chamber will generate a force on the first piston which is larger than the opposite spring force, which leads to displacement of the first piston so that the flow connection between the chambers is opened, until the pressure build-up in the second pressure chamber has generated an increased spring force slightly larger than the fuel pressure on the effective cross-sectional area of the first pressure chamber. The increased spring force will then again return the first piston to its extreme position with a blocked flow connection between the chambers.

The restricted drain passage provides continuous drainage of a small amount of fuel from the second pressure chamber. This ensures that the loading of the spring is also reduced when the engine load and thus the highest pressure in the fuel passage decreases. If the load does not decrease, the amount of fuel drained away will be replaced by new fuel at the next injection period as the draining away of fuel produces a small lowering of pressure in the second pressure chamber so that the first piston can again open the flow connection.

The opening pressure of the fuel injector at a specific load in the upper load range of the engine depends on the effective cross-sectional area of the first chamber. As such an area can be manufactured at very fine tolerances, all the fuel injectors in the engine will adjust themselves with the same opening and closing pressures, as the various manufacturing-determined variations in the spring characteristics of the

compression springs and their different sagging during the operating period will be equalized by compression of the springs until they yield the same spring force which adjusts to the highest fuel pressure in the injectors. This adjustment takes place automatically during the operation of the engine, and this obviates a substantial part of the need for periodical manual adjustment of the injectors.

It is possible to design the flow connection with a side opening which can be covered or uncovered by the first piston at its displacement, but preferably, the first piston has a seat section which blocks the flow connection between the first and the second pressure chambers by contact with a corresponding seat section on the stationary component. Such seat sections have proved very reliable in fuel injectors over many years, and they produce a well-defined closure capable of resisting large pressure differences.

The drain passage may suitably be of such limited size that the volume drained away at full engine load during an engine cycle ranges from one half to one twentieth of the fuel volume in the second pressure chamber. If the amount drained away becomes greater than half, it will be difficult, particularly at low engine loads, to achieve the desired increase in the opening pressure, and in addition, the piston movements will become large and frequent, as the second pressure chamber will have to be refilled at each injection period, also when the engine load is constant. If the volume drained away is less than one twentieth, the opening pressure will drop unsuitably slowly at a sudden reduction of the engine load. The said drain ratios apply at full engine load.

In one embodiment, the effective cross-sectional area of the first pressure chamber may be smaller than

the opening area of the valve slider. The result of this is that the first piston will remain in the above extreme position at low engine loads when the fuel pressure is determined by the opening pressure of the injector produced solely by the mechanical preloading of the spring. Only when the fuel pressure rises at increasing engine load will the pressure on the effective cross-sectional area of the first chamber result in a force which can overcome the spring force and move the first piston away from its extreme position.

Preferably, the second pressure chamber has an effective cross-sectional area which is several times larger than that of the first chamber. The result of this is that the pressure in the second pressure chamber is a corresponding number of times smaller than the pressure in the first chamber, when the force of the second piston acting on the spring and thus on the first piston equalizes the oppositely directed force of the fuel acting on the first piston. The large effective cross-sectional area of the second chamber thus results in closure of the second chamber at an advantageously low chamber pressure, which results in a relatively small pressure drop over the drain passage with a consequent small draining away of fuel from the second pressure chamber. The large area of the second chamber also yields the advantage that the chamber is filled with a large fuel volume at a certain displacement of the second piston and a corresponding compression of the spring. These two circumstances both contribute to the fact that the power of the spring is only changed slightly while the injector is in its closed position between two injection periods.

It is possible to position both pistons at the end of the spring closest to the nozzle, as the component which is stationary in relation to the first piston is

then constituted by the valve slider. The result of such a design is that the pistons participate in the adjustment movements of the valve slider. In this case the pistons will act as an increase of the mass of the slider, which will give the injector slower adjustment movements. As this is normally considered a disadvantage, the first piston may alternatively be formed at the opposite end of the spring. This results in the disadvantage that the flow connection between the two pressure chambers becomes elongated and relatively difficult to manufacture. In a preferred embodiment which both avoids these disadvantages and is simple to manufacture, the fuel injector is formed so that in a manner known per se the compression spring is mounted between two spring guides which are longitudinally displaceable on a central thrust member which is stationary in the housing, that the second piston is formed in the upper spring guide which is located opposite the nozzle and has a lower tubular wall which pressure-sealingly encloses the thrust member, and an upper tubular wall which has a larger internal diameter than the lower wall and pressure-sealingly encloses the first piston, and an intermediate member interconnecting the walls, its upper surface constituting the second surface, that the first piston is annular and is enclosed between the thrust member and the upper wall of the second piston and has a lower inward collar the upper surface of which constitutes the first surface, which inwardly merges into the seat section, and that the corresponding seat section of the thrust member faces downwards and is positioned between a through-going channel to the fuel passage and a lower area of reduced diameter constituting the flow connection between the two chambers.

The drain passage may be formed as an independent member, for example in the shape of a small bore through the second piston into the second chamber, but preferably, the drain passage consists of the pressure-sealing 5 annular slits between the two walls of the second piston, and the first piston and the thrust member, respectively, these annular slits being difficult to make completely pressure-tight as it is. The amount of fuel drained away will at the same time lubricate the 10 surfaces sliding against each other.

An example of an embodiment of the invention will now be explained in further detail below with reference to the drawing, in which

Fig. 1 shows a partial, longitudinal section 15 through a fuel injector according to the invention,

Fig. 2, on a larger scale, shows a portion of Fig. 1 depicting the compression spring with associated members, and

Fig. 3 is a diagram of the correlation between the 20 engine load and the opening pressures of injectors of the known type and those of the invention, respectively.

Fig. 1 shows a fuel injector generally designated 1, having an external housing 2 for mounting in a cylinder cover. The housing is elongated and at its 25 upper end has a mounting member 3 which projects sideways and by means of bolts fastened in the cover presses a contact surface 4 at the lower end of the housing against a corresponding contact surface formed in the cover. A fuel pump, not shown, or a similar 30 source of periodic supply of highly pressurized fuel is connected through a pressure conduit to a fuel inlet 5 at the top of the injector, from where a fuel passage 6 passes centrally through the injector down to a nozzle 7 with a central cavity 8, from which nozzle holes, not

shown, radiate for injection of fuel into the engine cylinder.

The fuel passage may have a valve which opens for circulation of preheated fuel in the injector between 5 injection periods. The fuel passage passes through a stationary component in form of a thrust member 9 which is in upward contact with a member 10 which is stationary in the housing, and in downward contact with an intermediate member 11 which is pressed firmly against 10 a slider guide 12 stationary in the housing. A valve slider 13 is received so as to be longitudinally displaceable in a central guide bore 12' in the slider guide, and by its one end encloses a downwardly projecting cylindrical section 11' on the intermediate member. 15 The guide bore centres the slider so that an annular conical seat surface 14 located at the lower end of the valve slider and formed as a needle, is coaxial with a corresponding valve seat on the slider guide 12. When the valve slider is in its closed position with the seat 20 surface pressed against the valve seat on the slider guide, the tip of the needle projects into the central cavity of the nozzle and is here exposed to the engine cylinder pressure which spreads through the nozzle holes into the cavity and acts on the valve slider with a 25 force in the opening direction. The lower end of the valve slider and the slider guide 12 delimit a pressure chamber 15 which communicates with the fuel passage 6 via oblique bores 16. The downward, annular end surface of the valve slider, which is inwardly delimited by the 30 needle, is influenced by the fuel pressure in the chamber 15, as the fuel pressure acts on the valve slider with a force in the opening direction. The opening area of the valve slider is substantially determined by the difference in diameters between the

external diameter of the cylindrical section 11' and the internal diameter of the guide bore 12'.

The valve slider is also loaded in the closing direction, i.e. in a downward direction towards the valve seat, by means of a compression spring 17, the upper end of which is in contact with an upper spring guide 18 mounted displaceably on the thrust member 9, and the lower end of which is supported via a lower spring guide 19 also displaceably guided on the thrust member 9, by a slotted thrust bushing, the lower end surface of which is in contact with an upward collar on the valve slider 13. The spring force is thus transmitted via the spring guide 19 and the thrust bushing 20 to the valve slider 13.

An annular first piston 21 is mounted axially displaceably around the upper section of the thrust member 9. The internal diameter of the sliding surface 22 (Fig. 2) of the piston is adapted to the opposite guide surface 23 on the thrust member in such a manner that the annular slit between the surfaces is sufficiently narrow for the piston to enclose the thrust member in a sealing manner. The guide surface 23 terminates downwards in an cylindrical recess worked into the thrust member and communicating through a channel 24 with the central fuel passage 6 in the thrust member. The recess merges downwards into a cylindrical section 25 of a smaller external diameter than the guide surface 23. Below the section 25, the thrust member has an annular conical downward seat section 26.

The first piston 21 has a lower inward collar 27 having a conical upward seat section 26' which may pressure-sealingly abut the seat section 26. In the area abreast of the recess and the cylindrical section 25, the first piston and the thrust member delimit a first pressure chamber 28 with an effective cross-sectional

area determined by the difference in diameters between the section 25 and the guide surface 23. The effective cross-sectional area is positioned on the upper surface of the collar 27, i.e. on a first surface facing away from the spring so that the fuel pressure supplied through the channel 24 in the first pressure chamber acts on the first piston with a downward force.

A second piston 29 is formed integrally with the upper spring guide 18 and comprises an annular intermediate member 30 which supports a lower tubular wall 31 and an upper tubular wall 32. The inner side of the lower wall 31 is pressure-sealingly and longitudinally displaceably in contact with a cylindrical second guide surface 33 of the thrust member 9, and the inner side of the upper wall 32 is in pressure-sealing and axially displaceable contact with the outer side of the first piston 21. The first and the second pistons together with the thrust member 9 delimit a second pressure chamber 34 of an effective cross-sectional area determined by the difference in diameters between the cylindrical inner sides of the lower wall 31 and the upper wall 32. A cylindrical recess formed in the outer side of the thrust member and positioned immediately below the seat section 26 forms a flow connection 35 between the two pressure chambers, when the first piston has been moved away from the seat section 26.

The bottom side of the collar 27 may have one or more projections or an annular projection with cuts keeping the part of the second pressure chamber 34 positioned radially in alignment with the projection in flow communication with the flow connection 35, when the projection abuts the upper side of the intermediate member 30. In an alternative embodiment, not shown, the projection may be annular, and the inner side of the lower wall 31 may have a smaller diameter than the inner

side of the collar 27 so that the part of the second pressure chamber positioned closest to the flow connection 35 has an upward effective cross-sectional area which is open to the flow connection 35 when the
5 projection on the collar 27 abuts the upper side of the intermediate member 30 and blocks the connection to the remaining part of the pressure chamber 34. At a suitable pressure increase in the flow connection 35, this effective area will cause the second piston 29 to be
10 moved away from the first piston 21 with simultaneous uncovering of the full effective cross-sectional area of the second pressure chamber.

The second pressure chamber 34 is continuously in contact with a restricted drain passage consisting of
15 a pressure-sealing annular slit between the inner side of the upper wall 32 and the cylindrical outer side of the first piston, and of the pressure-sealing annular slit between the inner side of the lower wall 31 and the guide surface 33 on the thrust member.

20 Now follows a description of how the two pistons automatically generate the desired spring force in the compression spring 17. When the engine is stopped and the fuel passage 6 is not pressurized, the two pistons take the position shown in the drawing where the
25 compression spring 17 with its factory preloading presses the second piston 29 upwards for abutment against the first piston 21, which transmits the spring force to the thrust member 9 through the seat sections 26 and 26'.

30 When the engine starts and the load increases, the pressure in the fuel passage 6 rises during each injection period to a highest pressure which at low loads corresponds to the opening pressure of the injector and at higher loads is determined by the flow

resistance in the nozzle holes. The maximum pressure in the fuel passage thus rises at increasing engine loads.

The pressure in the fuel passage 6 spreads through the channel 24 to the first pressure chamber 28, and
5 when the pressure here reaches a level where the downward force on the first piston overcomes the preset spring force, the first piston is moved towards the spring, which is compressed between the spring guides 18 and 19, and at the same time the fuel flows via the
10 flow connection 35 into the second pressure chamber where the pressure is built up to a level which returns the first piston 21 into contact with the seat section 26, while the second piston 29 remains in the position in which the spring is given the extra loading.

15 If the pressure in the fuel channel 6 at subsequent injection periods increases to a higher level, the piston movements are repeated so that the spring 17 is given a loading which depends in a linear manner on the maximum pressure in the fuel passage 6.

20 Through the pressure-sealing annular slits, a small amount of fuel is continuously drained away from the second pressure chamber, the fuel being passed to a drain opening, not shown, via the internal cavity in the housing 2.

25 In this manner, the fuel injector according to the invention achieves a spring force and thus an opening pressure which increases with increasing engine loads, as shown in Fig. 3. This renders it possible to lower the opening pressure at low engine loads, as the
30 injector automatically generates the high opening pressure required at full load. Thus the compression spring can be pre-manufactured with a preloading yielding an opening pressure at low loads of about 200 bar, which promotes stable engine operation at partial
35 loads, and at the same time the opening pressure at full

loads is higher than in the known injectors, which promotes good atomization of the fuel at the start of the injection period.

Instead of the above central course, the fuel
5 passage 6 may in a well-known manner comprise a number of channels which extend in a stationary intermediate member along the outer side of the spring and which open out in the pressure chamber 15 via oblique channels in the slider guide. With this design, the opening area of
10 the valve slider is determined by the downward annular end surface surrounding the needle.



P A T E N T C L A I M S

1. A fuel injector (1) for an internal combustion engine, particularly a large two-stroke diesel engine, having an external housing (2) for mounting in a cylinder cover and a through-going fuel passage (6) opening out into a nozzle (7), and having a valve slider (13) longitudinally displaceable in a slider guide (12), which slider is acted upon in a direction towards its valve seat by a preloaded compression spring (17) and in the opposite direction by the pressure of the fuel in the fuel passage, a first piston (21), which is axially displaceable in the housing, being located at one end of the compression spring and having a first surface which faces away from the spring and, together with a stationary component (9), delimits a first pressure chamber (28) which communicates with the fuel passage through a channel (24), said compression spring biasing the first piston in a direction towards an extreme position with a minimum fuel volume in the first pressure chamber, c h a r a c t e r i z e d in that the spring is in force-transmitting connection with a second piston (29) having a second surface which faces away from the spring and constitutes the end wall in a second pressure chamber (34), that at movement in a direction away from said extreme position the first piston (21) opens a flow connection (35) between the first and the second pressure chambers, that the second pressure chamber (34) has a larger effective cross-sectional area than the first chamber (28), and that a restricted drain passage connects the second pressure chamber with a drain.

2. A fuel injector according to claim 1, c h a r a c t e r i z e d in that the first piston (21) has a seat section (26') which blocks the flow connection (35) between the first and the second pressure chambers by



contact with a corresponding seat section (26) on the stationary component.

3. A fuel injector according to claim 1 or 2, characterized in that the drain passage is of such limited size that the volume drained away at full engine load during an engine cycle ranges from one half to one twentieth of the fuel volume in the second pressure chamber (34).

4. A fuel injector according to any one of claims 1-3, characterized in that the effective cross-sectional area of the first pressure chamber (28) is smaller than the opening area of the valve slider.

5. A fuel injector according to any one of claims 1-4, characterized in that the second pressure chamber (34) has an effective cross-sectional area which is several times larger than that of the first chamber (28).

6. A fuel injector according to any one of claims 1-5, characterized in that in a manner known per se the compression spring (17) is mounted between two spring guides (18, 19) which are longitudinally displaceable on a central thrust member (9) which is stationary in the housing, that the second piston (29) is formed in the upper spring guide (18) which is located opposite the nozzle (7) and has a lower tubular wall (31) which pressure-sealingly encloses the thrust member, and an upper tubular wall (32) which has a larger internal diameter than the lower wall and pressure-sealingly encloses the first piston (21), and an intermediate member (30) interconnecting the walls, its upper surface constituting the second surface, that the first piston (21) is annular and is enclosed between the thrust member and the upper wall of the second piston and has a lower inward collar (27) the upper surface of which constitutes the first surface, which



inwardly merges into the seat section (26'), and that the corresponding seat section (26) of the thrust member faces downwards and is positioned between a through-going channel (24) to the fuel passage (6) and a lower
5 area of reduced diameter constituting the flow connection (35) between the two chambers.

7. A fuel injector according to claim 6, c h a r -
a c t e r i z e d in that the drain passage consists
of the pressure-sealing annular slits between the two
10 walls (31, 32) of the second piston, and the first
piston (21) and the thrust member (9), respectively.