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(54) Title: FUEL ELECTRO-INJECTOR ATOMIZER FOR A FUEL INJECTION SYSTEM FOR AN INTERNAL COMBUSTION ENGINE

(57) Abstract: An atomizer (10) of a fuel electro-injector (1) for a fuel injection system for an internal combustion engine having a passage (13) that extends along a longitudinal axis (5) and includes a guide hole (46); the atomizer (10) has a sealing seat (21) defined by a chamfer or a fillet, which joins an cylindrical surface (41) inside the passage (13) and an external front surface (40); the atomizer (10) is equipped with a needle (27) having a first slider portion (25) coupled in an axially sliding manner to the guide hole (46), so as to define a dynamic seal and be axially movable under the thrust of an electro-actuator (32) from a closed position, in which a head (20) of the needle (27) is coupled to the seating seat (21) to close the outlet and define a static seal; the diameter (D) of the guide hole (46) is between 2.5 and 3.3 mm, while the difference between the average diameter of the sealing seat (21) and the diameter of the guide hole (46) is such as to generate a maximum axial imbalance of 50 N, directed in the opening direction, due to the effect of the fuel pressure around the needle (27) when the latter is positioned in the closed position.

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"FUEL ELECTRO-INJECTOR ATOMIZER FOR A FUEL INJECTION SYSTEM FOR AN INTERNAL COMBUSTION ENGINE"

TECHNICAL FIELD OF INVENTION

The present invention relates to an atomizer of a fuel electro-injector, in particular of the piezoelectric or magnetostrictive actuation type, for a high-pressure fuel injection system for an internal combustion engine. In particular, the present invention refers to a fuel electro-injector for a fuel injection system of the common rail type for a diesel cycle engine.

STATE OF THE ART

In internal combustion engines, the fuel injectors are equipped with an atomizer having a nozzle and a needle, which moves under the action of an actuator for opening and closing a sealing seat provided on the nozzle.

In other solutions, the atomizer has a needle of the so-called pintle type, i.e. a nozzle that opens by an outward movement of the needle (outwardly opening nozzle type). In other words, in this type of atomizer the nozzle is opened by pushing the needle by a piezoelectric or magnetostrictive actuator.

Solutions of this type are described, for example, in EP1559904 and FR2941745.

In this type of solution, the electric command signal supplied to the actuator causes a lengthening of the actuator, proportional to the supplied electric command signal, and this lengthening, in turn, causes translation of the needle in a direction concordant with the aforesaid lengthening. When no electric command signal is present, the actuator automatically shortens to its initial length. It is evident that the axial position of the needle, and therefore the circular section of fuel discharge, vary continuously, and not discretely,
according to the electric command signal supplied to the actuator.

The solution described in EP1559904 operates directly. In other words, the lengthening of the actuator causes an identical axial displacement of the needle. Instead, in the solution described in FR2941745, the actuator is connected to the needle by a hydraulic connection, namely a chamber filled with fuel. This chamber tends to compensate for the play in the assembly phase and the dimensional variations of the needle during operation.

However, in FR2941745, the axial movement of the injector needle is affected by the entity of the fuel pressure surrounding the injection needle when the sealing seat is closed.

Furthermore, the sealing seat of the nozzle appears to be a sharp circular edge, which would cause high stress on the nozzle material and therefore the occurrence of nozzle defects or breakage.

A similar solution to FR2941745 is described in European patent application 13189601.1, in the name of the same applicant and still confidential at the priority date of this application. This document shows a needle that is axially balanced with respect to fuel pressure when the nozzle is closed by a terminal head of the needle. In particular, the needle comprises a portion opposite to this head and coupled to a guide seat in an axially sliding manner to define a so-called "dynamic seal" between fuel contained in the nozzle and fuel contained in the intermediate part of the injector. The guide seat, or rather the dynamic seal, has a diameter equal to that of the static sealing seat of the nozzle. Therefore, when the nozzle is closed, the axial pressure acting on the end portion of the needle, on one side, and on the head, from
the other side, are the same and therefore cancel each other. This stratagem significantly reduces the force acting on the static sealing seat of the nozzle.

Balancing of the needle is a very important aspect for the behavioural stability of the atomizer during the various injection operations, as it allows operation sensitivity to be reduced to the fact that the fuel supply pressure is variable depending on the engine's different speeds of rotation. In particular, this supply pressure can have values in excess of 2200 bar in modern injection systems.

However, on this injector, experimentation has revealed the occurrence of defects and breaking of the needle and/or nozzle, due to the high stresses to which they are subjected. To avoid these drawbacks, the most obvious solution would be to significantly increase the diameter of the needle, so as to increase rigidity and reduce contact stress between needle and nozzle.

However, an imprudent increase of this diameter could bring other problems.

In particular, a generalized increase in the diameter of the needle also results in increasing the diameter of the sealing zones of the needle. Due to this increase in axial area, when the nozzle is closed, the fuel pressure induces a greater tensile state on the needle, even if the latter is "balanced", and this tensile state tends to place the needle in traction and consequently cause undesired lengthening.

In addition, an increase in the diameter of the sealing zone at the dynamic seal causes an undesired increase in fuel leakage from the nozzle to the intermediate part of the injector: the coupling clearance at the dynamic seal could be reduced as a countermeasure, but this would entail an increase
in production costs and, secondly, would increase the risk of making the needle hyperstatic, thereby compromising its correct closure.

Furthermore, with an increase in the seal diameter at the sealing seat of the nozzle, it would be necessary to reduce the lift of the needle to obtain the same discharge section and therefore the same amount of injected fuel (for the same fuel supply pressure). This reduction in lift also tends to reduce the accuracy and precision of the amount of fuel injected: in fact, with a large seal diameter, even small errors in actuator travel, and consequently in needle lift, have a decidedly more significant effect on the amount of fuel effectively injected, especially for minimum or partial engine load operating conditions, i.e. for small volumes of fuel to be injected in the combustion chamber. In other words, these small errors cause greater imprecision in injection with respect to that specified by design and consequently make emissions significantly worse.

Finally, the end of the needle is subjected to the pressure in the combustion chamber, which can arrive to 200 bar and generate a closing force, the entity of which is proportional to the static seal diameter and which must be overcome in order to open the nozzle and consequently inject fuel: it follows that a larger static seal diameter causes a higher closing force, exerted by the pressure in the combustion chamber, and could require a higher performance actuator, with an inevitable increase in costs.

**SUBJECT AND ABSTRACT OF THE INVENTION**

The object of the present invention is that of providing an atomizer of a fuel electro-injector for a fuel injection system for an internal combustion engine, which enables the above-described drawbacks to be solved in a simple and inexpensive manner, and preferably has the right compromise
for the diameter of the needle and, more in general, for the geometry and configuration of the atomizer.

According to the present invention, an atomizer of a fuel electro-injector for a fuel injection system for an internal combustion engine is provided, as defined in claim 1.

**BRIEF DESCRIPTION OF DRAWINGS**

For a better understanding of the present invention some preferred embodiments will now be described, purely by way of non-limitative example and with reference to the attached drawings, where:

- Figure 1 shows, in cross-section along a meridian section, a first preferred embodiment of the atomizer of a fuel electro-injector for a fuel injection system for an internal combustion engine, according to the present invention;
- Figure 2 is an enlargement of the atomizer in Figure 1;
- Figure 3 shows a perspective cutaway of a detail of Figure 2;
- Figure 4 is similar to Figure 2 and shows a variant of the atomizer according to the present invention;
- Figure 5 shows, in perspective, a detail of Figure 4; and
- Figure 6 is similar to Figure 5 and shows a further variant of the atomizer according to the present invention.

**DETAILED DESCRIPTION OF THE INVENTION**

The present invention will now be described in detail with reference to the attached figures to enable a person skilled in the art to make and use it.

In Figure 1, reference numeral 1 indicates, as a whole, a (schematically shown) fuel electro-injector forming part of a high-pressure fuel injection system, for injecting fuel into a combustion chamber (not shown) of an internal combustion engine. In particular, the injection system is of the common
rail type, for a diesel-cycle internal combustion engine.

The electro-injector 1 comprises an injector body 4 that extends along a longitudinal axis 5, is preferably formed by a number of pieces fastened together, and has an inlet 6 to receive fuel supplied at high pressure, in particular at a pressure in the range between 600 and 2800 bar. In particular, the inlet 6 is connected (in a manner not shown) to a common rail, which in turn is connected to a high-pressure pump (not shown), also forming part of the injection system.

The electro-injector 1 terminates with a fuel atomizer 10 having a nozzle 11 fastened to the injector body 4. The electro-injector 1 comprises a valve needle 12, which extends along axis 5 and is axially movable in an axial seat or passage 13 for opening/closing the nozzle 11, by performing an opening stroke directed axially outwards from the seat 13 and a closing stroke directed axially inwards into the nozzle 11 of the injector body 4.

Given this movement configuration, this type of electro-injector 1 is generally referred to as an "outwardly opening nozzle type", or as "pintle".

In the example shown in Figure 1, the valve needle 12 is defined by two distinct parts arranged axially against each other. In particular, the valve needle 12 is constituted by a needle 27, which is part of the atomizer 10, and by a drive rod 28 arranged in an intermediate zone of the seat 13 of the injector body 4. Alternatively, the valve needle 12 is defined by a single part.

The nozzle 11 comprises a sealing seat 21, which, together with a head 20 of the needle 27, defines a discharge section 14 for the fuel. The discharge section 14 has a continuous circular crown-like shape, with a width that is constant along
the circumference, but continuously increases as the opening stroke of the valve needle 12 proceeds.

The fuel is thus injected into the combustion chamber with a spray that is homogeneous along the circumference of the discharge section 14, i.e. a conical or "umbrella" spray, and with a variable flow rate proportional to the axial stroke of the needle 27.

As will be explained below, and as can be seen in Figure 3, the sealing seat 21 is not defined by a sharp edge, but by a chamfer or a fillet radius, which joins together a front surface 40, external to the seat 13 and the sealing seat 21, and a cylindrical surface 41 inside the seat 13, so as to reduce the pressure or specific load of the head 20 on the nozzle 11 during closure and therefore reduce stress and risks of fatigue failure.

The head 20 has an external diameter greater than the maximum diameter of the sealing seat 21 and the remainder of the needle 27. Near the nozzle 11, the head 20 is delimited by a surface 42 suitable for shutting against the sealing seat 21 and defined by a truncated cone or a convex segment of a sphere symmetrical with respect to axis 5. These two components, when mated in contact, define a single "static seal", i.e. a seal that guarantees perfect closure of the outlet of the nozzle 11.

Advantageously, the sealing seat 21 is defined by a chamfer, which has taper angle equal to that of surface 42 and is therefore complementary to surface 42, so as to ensure that contact is defined by a surface, instead of by a circular line, such that the specific load is reduced. In addition, this configuration enables achieving the design-specified entity or extension of the sealing seat 21 in a relatively precise manner through the machining of the chamfer.
The angle at the vertex formed by the cone of surface 42 is preferably between 100° and 150°. This angle must be chosen during design according to the characteristics of the engine (for example, the bore and/or shape of the combustion chamber, the tumble index, etc.), other characteristics of the injector 1 (maximum stroke defined by the actuator 32, static seal diameter, etc.) and characteristics of the injection system (maximum injection pressure, type of injection strategies, etc.). By way of example, if it is wished to adopt the smallest possible static seal diameter, it is necessary to take the largest possible angle at the vertex of the surface 42 (between 140° and 150°), so as to ensure a more substantial discharge section 14.

According to a variant that is not shown, the sealing seat 21 is defined by a fillet radius, while surface 42 has a recess or notch, defined by the inverse of this fillet radius, i.e. by a concave surface complementary to that of the fillet radius. Contact is defined by an annular surface in this case as well.

From the manufacturing viewpoint, it is possible to obtain this recess on surface 42 by setting the nozzle 11 and the needle 27 in reciprocal rotation and applying a certain axial load between the two components for a predetermined machining time, in a temperature-controlled environment, and preferably adding a fluid containing abrasive micro-particles on the rubbing areas.

Alternatively, contact at the sealing seat 21 takes place along a circular line and not on a surface, for example, in the case where the sealing seat 21 is defined by a chamfer and surface 42 is defined by a convex segment of a sphere (with the advantage of automatically centring the needle 27 with respect to the nozzle in the closure phase); or in the case
where the sealing seat 21 is defined by a fillet radius and the surface of the sealing seat 21 has a truncated-cone shape.

These solutions with a circular contact line are still advantageous even if the specific load is not reduced. In fact, from predictive computer simulations and some experimental fatigue resistance tests on similar products, it has been found that stress on the material of the nozzle 11 at the sealing seat 21 is however less with respect to a condition with contact on a sharp edge of the nozzle 11.

As mentioned above, the sealing seat 21 and the needle 27 are sized for defining a discharge section 14 that varies continuously, and not in a step-wise discrete manner, as the axial position of the needle 27 varies. In particular, when starting from the closed position, in which the head 20 rests against the sealing seat 21 and the nozzle 11 is therefore closed, the outward opening stroke of the needle 27 causes an initial opening of the nozzle 11 and then a progressive increase in the discharge section 14 for the fuel.

Therefore, with a relatively small opening stroke, the discharge section 14 is also relatively small, and so the fuel is injected with high atomization and a spray characterized by moderate penetration. With a relatively long opening stroke, the discharge section 14 is also relatively long: thus, also considering the particular geometry of the head 20, the fuel is injected with a spray characterized by high penetration. This variability of the discharge section 14 can be advantageous in implementing an engine operating mode of the mixed type, namely an HCCI-type (Homogeneous-Charge Compression-Ignition) mode at low and medium loads, with high fuel atomization in the combustion chamber, and a traditional CI-type (Compressed Ignition) mode at high loads, with high fuel penetration in the combustion chamber.
With reference to Figure 2, the atomizer 10 has a passageway 16, which permanently communicates with the inlet 6 through at least one channel, made in the injector body 4 and in the nozzle 11, and is radially defined by a stem 43 of the needle 27 and by the nozzle 11.

The passageway 16 has a passage section that is sufficiently large to limit pressure drops in the nozzle 11 to a minimum during the injection phase. Thus, high-pressure fuel does not flow through any micro-holes and the amount of fuel injected depends exclusively on the size of the discharge section 14 and the pressure difference between the annular passageway 16 and the combustion chamber.

At one axial end, the passageway 16 comprises a portion 44 that is annular and defines the outlet of the nozzle 11 at the sealing seat 21, so that fuel can be injected in the combustion chamber.

In the particular example shown, at the opposite axial end with respect to portion 44, the passageway 16 comprises an annular chamber 18, usually called a "cardioid", which has a wider cross-section than the remaining part of the passageway 16 and receives pressurized fuel from a channel (not shown) that places chamber 18 in communication with the inlet 6. It is therefore evident that the passageway 16 constitutes part of a high-pressure environment, defined by the fuel supply pressure. As visible in Figure 1, the injector body 4 also has a low-pressure environment 22, which communicates with an outlet 23 connected, in use, to lines (not shown) that return fuel to a fuel tank and which are at a low pressure, for example, approximately 2 bar.

Returning to Figure 2, at the opposite axial end with respect to the sealing seat 21, the nozzle 11 comprises a rear guide portion 45 having a guide hole 46, defined by a section of the
seat 13 and engaged in an axially sliding manner by a slider portion 25 of the needle 27.

The coupling zone between portion 25 and the guide hole 46 defines a so-called "dynamic seal". In general, the term "dynamic seal" is to be intended as a sealing zone defined by a shaft/hole type of coupling, with sliding and/or a guide between the two components, where play in the radial direction is sufficiently small to render the amount of fuel that seeps through to be negligible. In particular, this radial coupling play is less than or equal to 2 \( \mu \text{m} \). Also due the small size of this radial play, a relatively small amount of fuel leaks from chamber 18: this fuel will then flow to the outlet 23 to return to the fuel tank. Preferably, but not exclusively, the above-mentioned "dynamic seal" axially separates chamber 18 of the passageway 16 from the low-pressure environment 22.

The diameter of the surface 41 at the outlet of the seat 13 is equal to that of the guide hole 46, while in the other zones of the passageway 16, the internal diameter of the nozzle 11 is greater or equal to this value. In this way, it is possible to insert portion 25 of the needle 27 through the outlet of the nozzle 11 during assembly of the injector 1, providing a needle 27 defined by a one-piece body.

With reference to Figure 3, due to the fillet or chamfer that defines the sealing seat 21, the latter has an average diameter that is slightly larger than the diameter of the guide hole 46 and of surface 41.

The difference \( R \) between the diameter of the dynamic seal at the guide hole 46 and the average diameter of the static seal at the sealing seat 21 causes an imbalance in the axial forces exerted by the fuel pressure on the needle 27 when the nozzle 11 is closed by the head 20 of the needle 27.
In other words, the slightly larger diameter of the "static seal" with respect to the "dynamic seal" causes the fuel pressure to exert an opening force on the needle 27 when the nozzle 11 is closed. According to one aspect of the present invention, the average diameter of the "static seal" is designed in such a way that this "unbalancing" opening force is less than 50 N, even when the injection pressure is equal to the maximum design value.

With reference to Figure 1, to cause translation of the valve needle 12, the electro-injector 1 comprises an actuator device 30, in turn comprising an electrically-controlled actuator 32, i.e. an actuator controlled by an electronic control unit (not shown) that, for each phase of injecting fuel and the associated combustion cycle in the combustion chamber, is programmed to supply the actuator 32 with one or more electric command signals to perform corresponding injections of fuel.

The type of actuator 32 is such as to define an axial displacement proportional to the electric command signal received: for example, the actuator 32 could be defined by a piezoelectric actuator or by a magnetostrictive actuator. The actuator device 30 further comprises a spring 35, which is preloaded to exert axial compression on the actuator 32 to increase efficiency.

The excitation given by the electric command signal causes a corresponding axial extension of the actuator 32 and consequently a corresponding axial translation of a piston 34, which is coaxial and fixed with respect to an axial end of the actuator 32. In the particular example shown in Figure 1, the same spring 35 holds the piston 34 in a fixed position with respect to the actuator 32.

The axial translation of the piston 34 pushes on the valve needle 12 and consequently causes the opening of the nozzle
11, against the action of a spring 31 that is preloaded to axially push the needle 27 inwards and consequently to close the nozzle 11.

In particular, as can be seen in Figure 2, the spring 31 is arranged axially between an axial end shoulder of the nozzle 11, indicated by reference numeral 48, and an end portion 49 of the needle 27. Preferably, on one side, the spring 31 rests axially against a half-ring 83 that, in turn, abuts against the end portion 49 and, on the other side, against a spacer 84, which in turn abuts axially against a half-ring 85 resting on the shoulder. Alternatively, the spacer 84 could be arranged between the spring 31 and the half-ring 83. The axial thickness of the spacer 84 can be opportune chosen to adjust the preloading of the spring 31. The half-ring 83 is simply slipped on the needle 27, or is fastened to the needle 27, for example by welding or interference fitting. According to a variant that is not shown, the half-ring 85 is not present, while the spacer 84 rests directly on the shoulder 48.

Preferably, the spring 31 is arranged in a section of the seat 13 that is part of the low-pressure environment 22.

The spring 31 forms part of the atomizer 10 and advantageously has a preloading of between 60 and 150 N so as to exert sufficient closing force to overcome the above-stated imbalance and immediately return the needle 27 to the closed position once the action of the actuator 32 ceases. In particular, the preload value of the spring 31 must be chosen in the design phase in a manner proportional to the static seal diameter, i.e. the average diameter of the sealing seat 21, and in a manner proportional to the maximum value of the fuel supply pressure.

With reference to Figure 1, preferably, but not exclusively, the actuator 32 is coupled to the valve needle 12 by a
hydraulic connection 36. The hydraulic connection 36 comprises a pressure chamber 37, which is coaxial with the valve needle 12 and the piston 34 and defines a control volume filled with fuel that, once compressed, transmits the axial thrust from the piston 34 to the valve needle 12. The amount of fuel in the control volume of the pressure chamber 37 varies automatically for compensating the axial play and dimensional variations of the valve needle 12 during operation, in a manner not described in detail.

As visible in Figures 2 and 5, the needle 27 is axially guided not only by the guide hole 46, but also by a zone 50 of surface 41. The needle 27 comprises a plurality of sectors 51, which project radially from the stem 43 so as to be coupled to zone 50 in an axially sliding manner, are axially set apart from the head 20 and portion 25 and are separated from each other by gaps that form part of the passageway 16 to allow the axial passage of fuel towards portion 44.

The section passage area for the fuel at such gaps is relatively high, namely it is higher than the area of the discharge section 14 even when the lift of the needle 27 reaches its highest value, so that it does not define any restriction in the flow and, therefore, any limitation in the fuel flowrate, which has to be defined only by the lift of the needle 27.

According to the variant in Figure 6, sectors 51 are replaced by projections 51a that do not have a rectilinear profile parallel to axis 5, but have a helical or twisted profile, and which also connect to zone 50 with the function of a slide. With this configuration, the fuel flowing in the passageway 16 towards the sealing seat 21 during each injection confers a rotary motion to the needle 27 with respect to axis 5. This stratagem causes a relative rotation between nozzle 11 and needle 27 to avoid the formation of craters, remove any
residue clogged in the passageway 16 and impedes the formation of coking between the two contact surfaces.

Furthermore, the projections 51a confer a component of angular motion about axis 5 to the fuel leaving the atomizer 10, so as to favour the combustion conditions in the combustion chamber.

In the example shown in Figure 2, the stem 43 extends from portion 25 with a diameter smaller than that of portion 25, so as to form a step or axial shoulder at chamber 18.

Instead, in the variant in Figures 4 and 5, the stem 43 comprises: a portion 53 that has the same diameter as portion 25 and defines an axial extension of portion 25; a portion 54 that has a smaller diameter and length with respect to portion 53 and axially joins the latter to sectors 51; and a portion 55 that (as in Figure 2) radially defines portion 44 of the passageway 16 and also has a smaller diameter than portion 25. At the same time, the nozzle 11 comprises a cylindrical zone 57, which radially faces all of portion 53 of the stem 43 and has a larger diameter than zone 50 so as to define a widening to ensure the correct passage section of the passageway 16.

This configuration enables strengthening the stem 43 of the needle 27 without having to excessively increase the diameter of the above-mentioned static and dynamic seals.

Preferably, the passageway 16 does not have chamber 18, as zone 57 extends axially with the same diameter up to guide portion 45. The fact of eliminating the cardioid 18 enables strengthening the structure of the nozzle 11 and making it less sensitive to fuel pressure, which tends to deform the nozzle 11 and consequently alter the stroke of the valve needle 12 with respect to that specified by design.

According to one aspect of the present invention, the diameter
of the guide hole 46 and of portion 25 is between 2.5 and 3.3 mm so as to obtain sufficient rigidity and resistance for the needle 27, but without having to excessively alter the rise of the valve needle 12 to obtain the desired fuel flow during each injection, and without having a significant increase in fuel leakage through the above-described dynamic seal.

Furthermore, from simulations carried out by the applicant, the above-indicated sizing for the sealing diameter of the needle 27 constitutes the right compromise to obtain structural resistance and, at the same time, the least possible sensitivity to variations in fuel supply pressure and the pressure present in the combustion chamber during injection.

To meet the maximum imbalance requirement of 50 N, the radial size R of the chamfer or fillet that defines the sealing seat must satisfy the following relation:

\[ 0 < (D + R)^2 - D^2) \frac{P_{max}}{4} \leq 50 \]

where:
- \( D [m] \) = diameter of the dynamic seal (i.e. of the guide hole 46, excluding slight differences due to radial coupling play with portion 25);
- \( R [m] \) = difference between the average diameter of the static seal and the diameter of the dynamic seal;
- \( P_{max} [Pa] \) = design-specified maximum fuel supply pressure for the injection system where the injector 1 will be installed.

By choosing, for example:

\( P_{max} = 2400 \text{ bar} \)

\( D = 3.3 \text{ mm} \)

it gives: \( 0 < R \leq 40 \mu \text{m} \).

In this way, the needle 27 is of an almost-balanced type with respect to axial pressure effects when the nozzle 11 is closed, and so the operation and axial stroke of the needle 27
are minimally sensitive to variations in fuel supply pressure.

Given the geometry of the nozzle 11, according to which the diameter of the guide hole 46 is equal to the diameter of the cylindrical surface 41 at the axial end of the seat 13, the value $R = 0$ (equivalent to the perfect balancing of the needle 27 to pressure effects when the sealing seat 21 is closed) cannot be reached in the embodiment in Figures 2 and 3.

In the case of a sealing seat 21 defined by a chamfer coupled to a truncated-cone shaped surface 42, contact takes place along a circular crown. The chamfer has a radial size substantially equal to the above-stated value $R$. As mentioned above, having a contact surface - and not a circular contact line - between the two components is important to reduce the specific contact load at the sealing seat 21 during closure.

In the case where the sealing seat 21 is defined by a fillet radius equal to $r$ - and not by a chamfer - a recess or notch is made on surface 42 of head 20, complementary to the fillet radius $r$, to obtain a surface contact and avoid a linear contact.

From the above, it is evident that the sizing of the above-described atomizer 10 enables finding the right compromise for increasing the life of the injector with respect to known solutions, without compromising the method of operation.

In fact, it is found that the above-established seal diameter and the maximum imbalance, set in the opening direction, enable achieving a relatively robust solution, without compromising other operating characteristics of the injector 1.

Other advantages have been explained above or are evident to a person skilled in the field from the characteristics of the
atomizer 10 that have been set forth herein.

In any case, various modifications can be made to the atomizer 10 that has been described and illustrated with reference to the attached figures, and the generic principles described can be applied to other embodiments and applications without departing from the scope of the present invention, as defined in the appended claims. Therefore, the present invention should not be considered as limited to the embodiments described and illustrated herein, but is to be accorded the widest scope consistent with principles and characteristics claimed herein.

In particular, the nozzle 11 could be defined by an end portion of the injector body 4, and/or the guide portion 45 could form part of a body separate from the nozzle 11, and/or the passageway 16 could be replaced by a passage that is shaped in a different way from that shown in the attached figures.

In addition, the pressure chamber 37 and not an intermediate zone of the passage 13 could be provided above the dynamic seal.

Furthermore, the valve needle 12 could be operated directly, i.e. the injector 1 could be devoid of the pressure chamber 37.

Finally, the shape of the projections 51a could be adopted in atomizers and injectors having different structural characteristics and/or dimensions from those indicated above (even if not falling within the scope of the appended claims).
CLAIMS

1) An atomizer (10) of a fuel electro-injector (1) for a fuel injection system for an internal combustion engine, the atomizer comprising a nozzle (11) having:
- a passage (13), which extends along a longitudinal axis (5) and comprises a guide hole (46);
- a front surface (40), which is external to said passage (13);
- a sealing seat (21) defined by a chamfer or a fillet, which joins a cylindrical surface (41) of said passage (13) to said front surface (40);
- said cylindrical surface (41) having a dimension equal to that of said guide hole (46);
- a stem (43), which has a diameter of less than said head (20), projects axially from said head (20) and engages said passage (13); said stem (43) and said nozzle (11) radially defining a passageway (16) between them through which a flow of high-pressure fuel can run; and
- a first slider portion (25) coupled in an axially sliding manner to said guide hole (46) so as to define a dynamic seal;
- said needle (27) being axially movable along an opening stroke axially directed outwards from said passage (13), starting from a closed position wherein said head (20) is coupled to said sealing seat (21) so as to define a static seal and close said nozzle (11); said sealing seat (21) and said head (20) defining a discharge section (14), which is annular and has a width that increases as the opening stroke of said needle (27) proceeds;
characterized in that
- the diameter of said guide hole (46) is between 2.5 and 3.3 mm;
- the difference between the average diameter of said
sealing seat (21) and the diameter of said guide hole (46) satisfies the following relation:
\[ 0 < (\frac{(D + R)^2 - 2}{D}) \leq \frac{\text{Pmax}}{4} < 50 \]

where:
- \( R \) [m] = difference between the average diameter of said sealing seat (21) and the diameter of said guide hole (46);
- \( D \) [m] = diameter of said guide hole (46);
- \( \text{Pmax} \) [Pa] = design-specified maximum fuel supply pressure in the passageway (16).

2) An atomizer according to claim 1, characterized in that said head (20) is delimited by a truncated-cone surface (42) axially facing said sealing seat (21); said sealing zone (21) being defined by a chamfer having a taper angle equal to that of said truncated-cone surface (42).

3) An atomizer according to claim 1, characterized in that said sealing seat (21) is defined by a fillet radius and in that said head (20) has a recess complementary to said sealing seat (21) and engaged by said sealing seat (21) in said closed position.

4) An atomizer according to claim 2, characterized in that said truncated-cone surface (42) forms an angle at the vertex of between 100° and 150°.

5) An atomizer according to any of the preceding claims, characterized by further comprising a spring (31) having a preloading that exerts thrust to move said needle (27) to said closed position; said preloading being between 60 and 150 N.

6) An atomizer according to any of the preceding claims, characterized in that said cylindrical surface (41) comprises a guide zone (50) distinct and axially spaced apart from said guide hole; said stem (43) comprising a second slider portion.
coupled in an axially sliding manner to said guide zone and defining at least one empty space forming part of said passageway for the passage of fuel towards the sealing seat.

7) An atomizer according to claim 6, characterized in that said second slider portion is defined by at least one projection having a substantially helical profile.

8) An atomizer according to any of the preceding claims, characterized in that said stem comprises:
   - a first portion, which defines an axial extension of said first slider portion; and
   - a second portion, which has a smaller diameter and length with respect to said first portion;

   said nozzle having a widened cylindrical zone, which radially faces all of said first portion and has a larger diameter than said cylindrical surface.

9) An atomizer according to claim 8, characterized in that said widened cylindrical zone has a constant diameter and axially terminates at said guide hole.

10) An atomizer according to claims 6 and 8, characterized in that said second portion axially joins said first portion to said second slider portion.

11) An atomizer according to any of the preceding claims, characterized in that said needle is defined by a one-piece body.
A. CLASSIFICATION OF SUBJECT MATTER

INV. F02M61/08 F02M61/12

ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F02M

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

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<td>US 7 942 349 BI (MEYER ANDREW E [US]) 17 May 2011 (2011-05-17) fig. ums 1-7 figures 1-5 -----</td>
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Further documents are listed in the continuation of Box C. See patent family annex.

* Special categories of cited documents:
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Date of the actual completion of the international search
28 January 2016

Date of mailing of the international search report
08/02/2016

Name and mailing address of the ISA
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk
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Authorized officer
Schwal Ier, Vincent

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