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(54) **INJECTION METHOD FOR INJECTING FUEL INTO A COMBUSTION CHAMBER OF AN INTERNAL-COMBUSTION ENGINE, ATOMIZER OF A FUEL ELECTRO-INJECTOR FOR CARRYING UT SUCH INJECTION METHOD, AND PROCESS FOR THE PRODUCING SUCH ATOMIZER**

(57) According to the method, fuel is injected into a combustion chamber of an internal-combustion engine via an atomizer (10) provided with a nozzle (11) having an internal seat (13) and a sealing seat (21); the atomizer is moreover provided with a valve needle (12) having a head (20) designed to couple to the sealing seat (21) and a stem (41), which engages the internal seat (13); the stem (41) and the nozzle (11) define a passage (16) having an annular chamber (43), which axially ends at the sealing seat (21); the stem (41) has an intermediate portion (45) coupled in an axially sliding way in the internal seat (13) and defining a plurality of channels (67), the outlets (72) of which are arranged at the annular chamber (43); the sealing seat (21) and the head (20) of the valve needle (12) define a outflow area (14), which is annular and has a width that increases as the opening stroke of the valve needle (12) progresses; the fuel is injected in such a way as to have a non-uniform velocity modulus at the outflow area (14), as the position in the circumferential direction varies, at least in a reference operating condition of the engine.

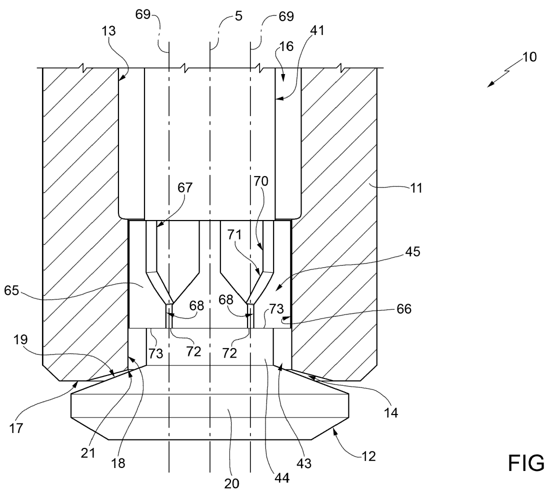

FIG. 2

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Description

TECHNICAL FIELD OF INVENTION

[0001] The present invention relates to an injection method for injecting fuel into a combustion chamber of an internal-combustion engine. Preferably, but not exclusively, the present invention refers to a fuel-injection system of the common-rail type for a diesel-cycle engine.

PRIOR ART

[0002] In internal-combustion engines, fuel injectors are provided with an atomizer having a nozzle and a needle, which translates under the action of an actuator for opening and closing a sealing seat provided on the nozzle.

[0003] In particular, in the most widely used diesel-cycle engines, the needle is actuated by means of a servo-actuation system, hence in an indirect way, basically on account of the strong forces of actuation that are necessary for getting the needle to translate, even though there is increasingly felt the need to design injectors with direct actuation of the needle, in particular to enable more complex actuation laws (for example, the so-called "boot shape" ones).

[0004] The atomizer is designed, in general, with the aim of obtaining a fuel spray such as to render as homogeneous as possible the distribution of the fuel with the air in the combustion chamber of the respective engine cylinder. In particular, a good homogenisation guarantees efficiency of combustion and hence reduces pollutant emissions.

[0005] Two important phenomena or factors that determine a higher or lower homogenisation of the fuel in the combustion chamber are the following:

- the so-called "penetration", i.e., the extension of the fuel spray inside the combustion chamber in a radial direction starting from the sealing seat provided at the end of the nozzle of the atomizer; and
- the atomization, i.e., the average size of the drops of fuel, and the tendency of the drops themselves to evaporate and/or to break up into smaller drops once they have entered the combustion chamber.

[0006] For a good homogenisation, it would be necessary to obtain fuel spray with a high penetration and with drops that have relatively small dimensions and/or have a high tendency to break up into smaller drops. In fact, high penetration is an essential condition, especially at high loads, i.e., at full power, to cause the fuel not to remain in the proximity of the outlet of the atomizer and to mix with the air. In fact, in diesel engines, at full power the pressure in the combustion chamber may assume values higher than 120 bar (according to the compression ratio and the pressure of supercharged air), and consequently the density of the air assumes values of approximately 40 kg/m^3. In these conditions, to enable the fuel spray to reach the proximity of the cylinder wall, it is necessary to overcome the high resistance offered by the density of the air, and hence a high penetration is necessary.

[0007] At the same time, a good atomization enables increase of the contact surface between fuel and air, given the same injected amount, to render combustion uniform.

[0008] In some solutions currently under production, distinguished by a fuel spray of a so-called "solid cone" type, the nozzle of the atomizer has a series of holes of preset dimensions (for example, 0.12 mm), arranged at equal distances apart about the axis of the injector. The needle axially translates under the control of the electric actuator so as to open/close an annular passage for supplying the fuel to such holes.

[0009] In this type of solutions, the needle lift determines a discrete variation of the fuel flow rate, basically of an on-off type. Consequently, the amount of injected fuel at each injection is determined by the times of opening of the nozzle and by the supply pressure of the fuel, and not by the needle lift. The sole exception is represented by the pilot injections, when volumes of fuel of less than 3-4 mm^3 are introduced. In this case, in fact, the times of actuation of the needle are extremely short and do not enable the needle to rise completely. In any case, the volume of fuel introduced once again depends upon the time of actuation of the electric actuator.

[0010] Furthermore, this type of solutions enables a spray with high penetration into the combustion chamber, but an atomization that is far from satisfactory at low and medium loads.

[0011] Finally, this type of atomizers does not enable a variable penetration and consequently is not compatible with injection strategies that envisage injection of the fuel markedly anticipated with respect to the top dead centre of the piston. In fact, the anticipated injections occur in a condition of low density of the air in the combustion chamber so that the solid-cone spray at high penetration would come to touch the wall of the cylinder, with consequent problems (dilution of the engine oil and greater formation of unburnt fuel and particulate).

[0012] In other solutions, the atomizer has a needle of so-called "pintle" type, i.e. the nozzle is opened via a displacement of the needle outwards ("outwardly opening nozzle type"), by pushing the needle via an actuator of a piezoelectric or magnetostrictive type.

[0013] Solutions of this type are illustrated, for example, in EP1559904.

[0014] In this type of solution, the electrical command supplied to the actuator causes a lengthening of the actuator itself, proportional to the electrical command supplied, and such lengthening in turn causes a translation of the needle in a direction concordant with the aforesaid lengthening. In the absence of the electrical command signal, the actuator shortens automatically and re-assumes its initial length. A spring thus brings the needle back into the closing position. The end of the needle is defined, in general, by a head delimited by a frustoconical surface that comes to bear upon a sealing seat defined by an annulus on the nozzle when the latter is closed. The spray deriving from this type of atomizer is shaped like a cone or an umbrella, commonly referred to as "hollow cone", in so far as it extends throughout the circumference of the sealing seat on the nozzle. It is evident that the axial position of the needle and, hence, the circular outflow area for the fuel vary in a continuous, and non-discrete, way as a function of the electrical command supplied to the actuator.

[0015] A particular solution of this type is proposed in US7942349, where the electric actuator is defined by an electromagnet, which in itself does not manage to lift the needle by an amount proportional to the actuation command. In particular, when the electromagnet is operated, the needle reaches its maximum lift and the flow rate is determined by a series of holes that are arranged upstream of the circular outflow area and have a passage section area that is calibrated, fixed in the spray. In the specific case, consequently, as in the solution with solid-cone spray, the amount of injected fuel is determined by the times of actuation of the electromagnet and by the supply pressure of the fuel.

[0016] As regards atomization of the fuel in the combustion chamber, the solution with hollow-cone spray enables drops to be obtained having a relatively small diameter at the atomizer outlet (approximately 1/10 of the exiting drops of a solid-cone spray).

[0017] As regards other aspects, however, this type of solution, in itself, is far from satisfactory.

[0018] First of all, by comparing a solution with solid-cone spray against a solution with hollow-cone spray, given the same supply pressure of the fuel and given the same pressure in the combustion chamber, it may be noted how the needle lift of the solution with hollow-cone spray must be relatively low to obtain the same fuel flow rate (if the solution according to US7942349 with calibrated holes upstream of the circular outflow area is not adopted). For example, with a hollow-cone spray it is necessary to raise the needle by less than 10 μm, if we consider a seal diameter of the needle of 3 mm, a supply pressure of diesel fuel of 1000 bar and a pressure of 40 bar in the combustion chamber, to obtain a flow rate of 0.025 kg/s (equal to that of an atomizer with solid-cone spray that operates in the same conditions).

[0019] As regards formation of the first drops, for the hollow-cone spray we have that, at the sealing seat (i.e., at the outflow area of the atomizer) there is hence formed a plane and continuous fuel film, which generates the drops of the spray; the thickness of this film is hence very close to the needle lift. The modality of formation of the first drops (the so-called primary "break-up"), for this type of atomizer, is illustrated in Figure 10A, and is referred to as "Linearized Instability Sheet Atomization" (LISA). It emerges that the diameter of the first drops generated is very close to the needle lift. Consequently, in the case considered, the first drops will have diameter of approximately 10 μm.

[0020] As regards formation of the first drops in the case of the solid-cone spray, as may be seen in Figure 10B, it is universally recognized that the first drops that are formed have a diameter equal to the diameter of the hole of the atomizer.

[0021] Consequently, from the standpoint of penetration of the spray into the combustion chamber, given the same pressure difference and hence given the same rate of the fuel exiting from the atomizer, the drops of fuel generated by the circular outflow area for the hollow-cone spray have a kinetic energy decidedly lower (approximately 1/1000) than that of the drops exiting in the case of the solid-cone spray, in so far as they have a mean radius that is smaller (approximately 1/10), which is determined basically by the needle lift, as explained above. This substantial difference of kinetic energy obviously leads the drops of fuel to follow a shorter path in the combustion chamber (approximately one half or one third, on the basis of experimental photographic findings) as compared to the solution with solid-cone spray.

[0022] In addition, even though the drops of fuel of the solution with hollow-cone spray are already relatively small at outlet from the atomizer, they present a low tendency to break up further into even smaller drops. This tendency is generally evaluated on the basis of a parameter referred to as "Weber number", which is the ratio of the inertial forces of the drops in the air to the surface-tension forces of the drops themselves. The higher the Weber number, the greater the tendency of the fuel to break up into smaller drops. Such parameter is directly proportional to the diameter of the drop so that it indicates clearly that the smaller drops of the hollow-cone spray have a lower tendency to break up than the larger ones of the solid-cone spray.

[0023] To limit the drawbacks described above, in the solutions with hollow-cone spray it would be expedient to increase the amount of the needle lift, i.e., the diameter of the first drops generated and hence the thickness of the liquid film that is formed at the sealing seat, keeping, however, the amount of injected fuel constant.

[0024] To obtain this result, assuming that the values of fuel flow rate and hence the width of the outflow area remain the same, in theory it would be possible to reduce the diameter of the sealing seat between the needle and the nozzle so as to reduce the diameter of the circular outflow area and proportionally increase the needle lift and hence the thickness of the liquid film. This reduction of the seal diameter, however, would cause also a reduction of the diameter of the needle. In practice, this diameter cannot be reduced beyond certain constructional limits (i.e., below 2 mm), in so far as the sturdiness and structural strength of the needle itself would be jeopardized.

[0025] Alternatively, the times of injection could be reduced and the instantaneous fuel flow rate could be increased, for example by doubling the needle lift. However, with this strategy, it would become problematical to control precisely the electric actuator in order to obtain the necessary precision, especially in the case of injection of small amounts of fuel. In fact, an error in the times of injection that is small in percentage terms would have significant repercussions on the amount of injected fuel. Moreover, the electric actuators have a minimum command time, below which it is not possible to obtain actuation. These drawbacks would be particularly evident during the pilot injections.

[0026] Consequently, instead of increasing the needle lift, it is expedient to find a compromise between the known solutions, in particular modifying the solutions of atomizer with hollow-cone spray according to the known art so as to obtain an operation that approaches the one obtained by atomizers with solid-cone spray, at least as regards penetration of the fuel spray into the combustion chamber in operating conditions at high loads (in particular at full power).

[0027] In particular, there is felt the need to provide an engine operating mode of a mixed type, namely, an HCCI (Homogeneous-Charge Compression-Ignition) mode at low and medium loads, with high atomization of the fuel in the combustion chamber and with contained penetration to prevent the phenomenon of impact or impingement of the fuel on the cylinder wall, and a traditional CI (Compressed Ignition) mode at high loads, with high penetration of the fuel into the combustion chamber.

SUBJECT AND SUMMARY OF THE INVENTION

[0028] The aim of the present invention is to provide an injection method for injecting fuel into a combustion chamber of an internal-combustion engine that will enable the drawbacks set forth above to be overcome in a simple and inexpensive way.

[0029] According to the present invention an injection method for injecting fuel into a combustion chamber of an internal-combustion engine is provided as defined in claim 1.

[0030] In addition, according to the present invention an atomizer is provided as defined in claim 4, as well as a process for producing such atomizer, as defined in claim 14.

BRIEF DESCRIPTION OF THE DRAWINGS

[0031] For a better understanding of the present invention some preferred embodiments thereof are now described, purely by way of non-limiting example, with reference to the attached drawings, wherein:

- Figure 1 is a cross-sectional view along a meridian section plane of a preferred embodiment of the atomizer of a fuel electro-injector for carrying out the injection method according to the present invention;
- Figure 2 is an enlargement of one end of the atomizer of Figure 1, with a nozzle illustrated in cross section and a valve needle illustrated with parts in view and parts in cross section;
- Figure 3 shows a diagram of hydraulic operation of the atomizer of Figure 2;
- Figure 4 is a schematic perspective view, with parts removed for reasons of clarity, of a velocity profile of the fuel within the atomizer of Figure 2;
- Figure 5 is a different perspective view that shows diagrams regarding the velocity of the fuel in the spray supplied by the atomizer according to the injection method of the present invention, in three different positions;
- Figure 6 regards a sequence of photographs taken in succession to show a fuel injection according to the injection method of the present invention;
- Figure 7 is similar to Figure 5 and shows a different diagram;
- Figures 8A, 8B, and 8C show patterns of the fuel spray supplied by the atomizer according to the injection method of the present invention, obtained via computer simulation abd different from the ones appearing in Figure 6;
- Figure 9 is similar to Figure 2 and shows part of a variant of the atomizer of Figure 2; and
- Figures 10A and 10B show the different ways of formation of the primary drops for the atomizers with hollow-cone spray and with solid-cone spray, respectively.

DETAILED DESCRIPTION OF THE INVENTION

[0032] The present invention will now be described in detail with reference to the attached figures to enable a person skilled in the art to implement it and use it.

[0033] In Figure 1, the reference number 1 designates a fuel electro-injector (illustrated in a simplified way) forming part of a high-pressure fuel-injection system, for injecting fuel into a combustion chamber 2 (schematically illustrated in Figure 3) of an internal-combustion engine. In particular, the injection system is of the common-rail type, for a diesel-cycle internal-combustion engine.

[0034] The electro-injector 1 comprises an injector body 4, which extends along a longitudinal axis 5, is preferably constituted by a number of pieces fixed together, and has an inlet 6 for receiving fuel supplied at high pressure, in particular at a pressure of between 600 and 2800 bar. In particular, the inlet 6 is connected in a way not illustrated to a common rail, which in turn is connected to a high-pressure pump (not illustrated), which also forms part of the injection system.

[0035] The electro-injector 1 ends with a fuel atomizer 10 comprising a nozzle 11, which is fixed to the injector body 4 and has a seat 13 obtained in a through way along the axis 5. The atomizer 10 further comprises a valve needle 12, which extends along the axis 5 and is axially mobile in the seat 13 so as to open/close the nozzle 11, performing an opening stroke, or lift, that is directed axially towards the outside of the seat 13, and a closing stroke that is directed axially towards the inside of the nozzle 11 and of the injector body 4.

[0036] Given this configuration of movement, this electro-injector 1 is of the type generally referred to as "outwardly opening nozzle", or else as "hollow-cone spray".

[0037] In the example illustrated in Figure 1, the valve needle 12 has a rear end portion 15 resting axially against a transmission rod 28, defined by a distinct piece set in an area in intermediate the injector body 4. According to an alternative not illustrated, the valve needle 12 and the rod 28 form a single piece.

[0038] With reference to Figure 2, the nozzle 11 has a sealing seat 21, which, together with a head 20 of the valve needle 12, defines a outflow area 14 for the fuel. The outflow area 14 is shaped like a continuous annulus, with a width that is constant along the circumference, but increases axially in a continuous way as the opening stroke of the valve needle 12 progresses.

[0039] Consequently, the fuel is injected into the combustion chamber 2 with a spray that is continuous along the circumference of the outflow area 14, i.e., with a spray that, immediately downstream of the outflow area 14, is conical or else umbrella-shaped, as may be seen also in Figure 5. The flow rate of the injected fuel through the outflow area 14 is variable, in a way proportional to the axial stroke of the valve needle 12.

[0040] Even though it is not clearly visible in Figure 2, the sealing seat 21 is not defined by a sharp edge, but by an annulus corresponding to a chamfer or to a radiusing that joins together a front surface 17, external to the seat 13 and to the sealing seat 21, and a cylindrical surface 18 of the seat 13. The chamfer or radiusing of the sealing seat 21 reduces the specific pressure or load of the head 20 on the nozzle 11 during closing and hence reduces the stresses and the risks of failure due to fatigue.

[0041] The head 20 has an external diameter greater than the maximum diameter of the sealing seat 21 and of the remaining part of the valve needle 12. Towards the nozzle 11, the head 20 is delimited by a surface 19 that is designed to come to bear upon the sealing seat 21 and is defined by a truncated cone or else by a convex cap symmetrical with respect to the axis 5. When these two components are coupled together in contact, they define a single static seal, i.e., a seal that guarantees a perfect closing of the outlet of the nozzle 11.

[0042] As mentioned above, the sealing seat 21 and the valve needle 12 are sized in such a way as to define a outflow area 14 that varies in a continuous way, and not in a discrete stepwise way, as the axial position of the valve needle 12 varies. In particular, starting from the closing position, in which the surface 19 of the head 20 rests against the sealing seat 21 and hence the nozzle 11 is closed, the opening stroke towards the outside of the valve needle 12 causes an initial opening of the nozzle 11 and then a progressive increase of the outflow area 14 for the fuel.

[0043] With a relatively small opening stroke, the outflow area 14 is also relatively small so that the fuel is injected with a marked atomization and a spray characterized by contained penetration and by a substantial homogeneity in the circumferential direction, if examined with a system of cylindrical co-ordinates.

[0044] With a relatively extensive opening stroke, the outflow area 14 is also relatively wide. As will be described more fully in what follows, the fuel is injected with a spray characterized by high penetration. This variability in the configuration of the spray as a function of the lift of the valve needle 12 proves advantageous, for example, for providing an engine operating mode of a mixed type, i.e., an HCCI mode at low and medium loads, with high atomization of the fuel in the combustion chamber 2, and a traditional CI mode at high loads, with high penetration of the fuel into the combustion chamber 2.

[0045] The same mode of actuation of the electro-injector 1 proves extremely advantageous also in the case where the combustion occurs with high percentage of EGR (Exhaust-Gas Recirculation), i.e., burnt gases made to recirculate in the combustion chamber 2, preferably in positions far from the outlet of the atomizer 10. In these conditions, the oxygen present in the combustion chamber 2 is concentrated in the proximity of the outlet of the atomizer 10 so that a spray with low penetration (stratified charge) is consequently necessary.

[0046] With reference to Figure 1, the atomizer 10 has a passage 16, which is defined radially by a stem 41 of the valve needle 12 and by the seat 13 of the nozzle 11. The passage 16 comprises an end zone 42 that communicates permanently with the inlet 6 by means of at least one channel (not illustrated), provided in the injector body 4 and in the nozzle 11, so that it defines a high-pressure environment. The end zone 42, in particular, is defined by an annular chamber, which is generally referred to as "cardioid" and has a cross section that is wider than the remaining part of the passage 16.

[0047] At the end zone 42 of the passage 16 there is substantially the same supply pressure (prail) as the one set by the fuel-injection system. The injector body 4 also has a low-pressure environment 22, which communicates with an outlet 23 connected, in use, with ducts (not illustrated), which get the fuel to recirculate towards a reservoir and are at low pressure, for example at approximately 2 bar.

[0048] As may be seen in Figure 2, at the opposite axial end, the passage 16 comprises an annular chamber 43, which is delimited radially by the surface 18 and by an axial end 44 of the stem 41. The axial ends of the annular chamber 43 are defined by the surface 19 of the head 20 and by an intermediate portion 45 of the stem 41, which will be described in detail in what follows. In other words, the annular chamber 43 axially ends at the sealing seat 21, so that the fuel is injected into the combustion chamber 2 through the outflow area 14.

[0049] As may be seen in Figure 1, at the opposite axial end with respect to the sealing seat 21, the nozzle 11 comprises a rear guide portion 46 having a guide hole

47, defined by a zone of the seat 13 and engaged in an axially slidable way by a slide portion 25 of the valve needle 12.

[0050] The coupling zone between the portion 25 and the guide hole 47 defines a so-called "dynamic seal". In general, by "dynamic seal" is meant a sealing area defined by a coupling of the shaft/hole type, with sliding and/or guiding between the two components, where the play in a radial direction is sufficiently small as to render the amount of leaking fuel negligible. In particular, such radial coupling play is smaller than or equal to 2 μm. Also thanks to the small extent of this radial play, a relatively small amount of fuel leaks from the end zone 42 of the passage 16. This fuel will then flow towards the outlet 23 and recirculate towards the reservoir. Preferably, the aforesaid dynamic seal separates axially the passage 16 directly from the low-pressure environment 22.

[0051] Preferably, at the chamber 43, the diameter of the surface 18 is equal to that of the guide hole 47, whereas in the other areas of the passage 16 the internal diameter of the seat 13 is greater than or equal to this value. At the same time, the mean diameter of the sealing seat 21 is slightly greater than the diameter of the guide hole 47 and of the surface 18. Consequently, the difference between the diameter of the dynamic seal at the guide hole 47 and the mean diameter of the static seal at the sealing seat 21 causes an unbalancing in opening of the axial forces exerted by the pressure of the fuel on the valve needle 12 when the nozzle 11 is closed by the head 20 of the valve needle 12. It is in any case a controlled unbalancing pre-defined in the design stage, which must not exceed the force exerted by the spring 54 (described in what follows). Alternatively, it is possible to replace the chamfer at the sealing seat 21 with a sharp edge, where the operating pressures so allow, or in the case where it is possible to use, for the nozzle 11, an extremely hard material (for example, tungsten carbide), or again to resort to surface-hardening treatments such as DLC (Diamond-Like Carbon) or nitridation: in this case, the diameter of the dynamic seal becomes exactly the same as the diameter of the sealing seat 21.

[0052] According to variants not illustrated, the relation between the mean diameter of the sealing seat 21 and the diameter of the guide hole 46 is different from the one indicated above for the preferred embodiments illustrated and described.

[0053] In order to cause translation of the valve needle 12, the electro-injector 1 comprises an actuator device 50, which in turn comprises an electrical-command actuator 51, i.e., an actuator governed by an electronic control unit (not illustrated), which is programmed for supplying to the actuator 51, for each step of fuel injection and corresponding combustion cycle in the combustion chamber 2, one or more electrical commands to provide corresponding fuel injections.

[0054] The type of the actuator 51 is such as to define an axial displacement proportional to the electrical command received: for example, the actuator 51 is defined by a piezoelectric actuator or by a magnetostrictive actuator. The actuator device 50 further comprises a spring 52, which is pre-loaded so as to exert an axial compression of the actuator 51 in order to increase the efficiency thereof.

[0055] The excitation provided by the electrical command causes a corresponding axial extension of the actuator 51 and hence a corresponding axial translation of a piston 53, which is coaxial and fixed with respect to an axial end of the actuator 51. In the particular example illustrated in Figure 1, it is the spring 52 itself that keeps the piston 53 in a fixed position with respect to the actuator 51.

[0056] Axial translation of the piston 53 causes a thrust to be exerted on the valve needle 12, via the rod 28, and hence opening of the nozzle 11, against the action of a spring 54, which is pre-loaded so as to push the valve needle 12 axially inwards and thus close the nozzle 11.

[0057] In particular, the spring 54 is set axially between an axial end shoulder of the nozzle 11, designated by the reference number 55, and the end portion 15 of the valve needle 12.

[0058] Preferably, the spring 54 rests axially, on one side, against a half-ring 57, which in turn axially bears upon the end portion 15, and, on the other side, against a spacer 58, which in turn axially bears upon a half-ring 59 resting on the shoulder 55. Alternatively, the spacer 58 may be set between the spring 54 and the half-ring 57. The axial thickness of the spacer 58 may be chosen in such a way it is appropriate for adjusting pre-loading of the spring 54. The half-ring 57 is simply fitted on the valve needle 12, or else is fixed to the valve needle 12 itself, for example via welding or via interference fit. According to a variant not illustrated, the half-ring 59 is not present, whereas the spacer 58 rests directly on the shoulder 55.

[0059] Preferably, the spring 54 is arranged in a cavity forming part of the low-pressure environment 22. Furthermore, the spring 54 advantageously has a pre-loading comprised between 60 N and 150 N so as to exert a closing force sufficient to overcome the aforesaid unbalancing and to bring the valve needle 12 promptly back into the closing position once the action of the actuator 51 has ceased. In particular, the value of pre-loading of the spring 54 must be chosen in the design stage in a way proportional to the static-seal diameter, i.e., to the mean diameter of the sealing seat 21, and in a way proportional to the maximum value of the supply pressure of the fuel.

[0060] Preferably, but not exclusively, the actuator 51 is coupled to the valve needle 12 via a hydraulic connection 61. The hydraulic connection 61 comprises a pressure chamber 62, which is coaxial to the valve needle 12 and to the piston 53, and defines a control volume filled with fuel, which, once compressed, transmits the axial thrust from the piston 53 to the valve needle 12. The amount of fuel in the control volume of the pressure chamber 62 varies automatically so as to compensate the axial play and the dimensional variations of the valve needle 12 and of the rod 28 during operation, in a way not described in detail.

[0061] According to variants (not illustrated), the hydraulic connection 61 is sealed with respect to the external hydraulic circuit for the fuel and is filled with a fluid without dissolved air (which would increase compressibility) and/or with a bulk modulus higher than that of the fuel.

[0062] As may be seen in Figure 2, according to one aspect of the present invention, the intermediate portion 45 is set at an axial distance from the portion 25 and is constituted by a plurality of sectors 65, which project radially outwards so as to couple in an axially sliding way to a surface 66 of the seat 13. The sectors 65 are separated from one another in the circumferential direction by channels 67, which allow passage of the fuel towards the annular chamber 43. The channels 67 are in a number greater than or equal to three and are set at equal distances apart about the axis 5. Preferably, the channels 67 are obtained on the outer surface of the portion 45 so that they are delimited radially on the outside by the surface 66.

[0063] The channels 67 may be obtained in the stem 41 by removal of stock, for example by electro-erosion or by laser etching. Alternatively, the channels 67 and the sectors 65, i.e., the portion 45, may be defined by a bushing that defines a piece separate from the rest of the valve needle 12 and is fixed, for example interference fitted, on the stem 41 during the production stages.

[0064] According to one aspect of the present invention, the channels 67 comprise respective end portions 68, which exit directly into the annular chamber 43 and extend along respective axes 69 parallel to the axis 5 with passage section areas that are constant along such axes 69. In this way, the portions 68 channel or guide the respective fuel flows, which then exit into the annular chamber 43, and do not bestow any swirling motion on such fuel flows in the annular chamber 43.

[0065] Preferably, the channels 67 further comprise respective initial portions 70, which define a passage section area greater than that of the portions 68 and are connected to the portions 68 via respective flared intermediate portions 71. The latter define a lead-in portion, with a passage section area that decreases progressively, without any steps, as far as the inlet of the portions 68 to limit the pressure losses at said inlet. Preferably, each pair of portions 70, 71 is aligned to the respective portion 68 along the axis 69.

[0066] In the particular and preferred example illustrated, the minimum passage section area is defined by the portions 68.

[0067] The presence of the initial portions 70, which are widened, has the function of limiting the axial length of the portions 68. In fact, the sectors 65 also have a function of guide for the valve needle 12 with respect to the nozzle 11 so that, practically, they cannot have an axial length smaller than 2 mm to carry out this function. On account of the relatively small passage section areas along the portions 68, there would be considerable losses by fluid friction if the portions 68 were as long as the sectors 65.

[0068] The minimum passage section area totally available for the fuel at the channels 67 is in any case relatively large. In fact, on the one hand, the passage restriction defined by the channels 67 introduces a drop in pressure not higher than 35% of the pressure present at the inlet of the channels 67 themselves. In this way, the fuel exiting from the portions 68 in the annular chamber 43 has a pressure that is at least 65% of the pressure at the inlet, with a velocity proportional substantially to the pressure difference (according to Bernoulli's principle, assuming the fuel to a first approximation as being incompressible and neglecting the losses by fluid friction).

[0069] On the other hand, the channels 67 do not have the function of determining the fuel flow rate delivered. In fact, their function is rather that of converting part of the pressure into velocity of the fuel within the annular chamber 43, without a substantial decrease in the total pressure of the fuel itself (the conservation of the total pressure depending, as explained hereinafter, upon the fluid friction).

[0070] For instance, if the maximum lift of the valve needle 12 is 0.02 mm, the mean diameter of the sealing seat 21 is 3 mm, and the half-angle at the vertex of the conical surface 19 with respect to the axis 5 is 55°, the passage area at the outflow area 14 is approximately 0.15 mm^2: applying the law of conservation of the flow rate in the portions 68 and in the outflow area 14 and resorting to Bernoulli's theorem applied between the inlet of the channels 67 and the outlets in the annular chamber 43, and moreover to Bernoulli's theorem applied between the inlet of the channels 67 and the outflow area 14, imposing that the pressure at the outlet of the portions 68 is at least 65% of the pressure in said inlet, neglecting moreover losses by fluid friction and/or by thermal dissipation, and considering the fluid as being incompressible, we can write a system of three equations in three unknowns (velocity of the fluid through the channels 67, velocity of the fluid through the outflow area 14, and total passage section area of the portions 68). From this system, we obtain that the minimum total passage section area of the channels 67 must be at least 0.28 mm^2.

[0071] In these conditions, if the total pressure, i.e., the pressure of the fuel in the common rail of the injection system, is for example 1000 bar, and the pressure in the combustion chamber 2 is for example 40 bar, the pressure at the outlet of the portions 68 will be approximately 650 bar and the velocity through the outflow area 14 will be approximately 365 m/s, while the velocity at the outlets into the annular chamber 43 will be approximately 210 m/s.

[0072] It is evident that the area or passage section area available for the fuel at the channels 67 is smaller than the one available in the passage 16 upstream and downstream of the intermediate portion 45 so that the channels 67 define a hydraulic resistance and determine a total pressure drop between the end zone 42 and the annular chamber 43 when the fuel flows. The outflow area 14, in turn, defines another hydraulic resistance, which can be adjusted by varying the lift of the valve needle 12. Wishing then to take into account these energy losses, it is necessary to increase the maximum value admissible for the pressure difference across the portions 68 by approximately 10% so that the maximum value admissible of the pressure difference through the channels 67 is 45%, noting that the preponderant part consists in conversion of the pressure into kinetic energy of the fuel.

[0073] Figure 3 shows a block diagram regarding this hydraulic configuration of the atomizer 10 during an injection. As mentioned above, in the end zone 42 there is substantially the supply pressure (prail) set by the injection system, whereas in the combustion chamber 2 there is the pressure (pcyl) of the air in the cylinder during injection. The mean pressure (p) within the annular chamber 43 assumes a value that is intermediate between prail and pcyl during delivery of the fuel and, once the geometry of the channels 67 and of the atomizer 10 as a whole is fixed, and the operating conditions of the electro-injector 1 (prail, pcyl, fuel flow rate) are fixed, can be calculated via the system of equations referred to above or else determined via appropriate fluid-dynamic simulations on a computer for evaluating more exactly the amount of the losses by fluid friction and by turbulence.

[0074] The outlets of the portions 68 of the channels 67 are designated in Figures 2 and 4 by the reference numbers 72. When the nozzle 11 is open, at the outlets 72 the fuel exiting from the channels 67 locally has a velocity greater than that of the fuel present in the annular chamber 43 in the points 73 that are intermediate between the outlets 72 along the same circumference (as may be noted from the flow lines that are schematically represented in Figure 4 and that derive from computer simulations).

[0075] According to one aspect of the present invention, the annular chamber 43 has dimensions sufficiently small as not to manage to render the velocity of the fuel uniform before the fluid fillets exiting from the channels 67 reach the outflow area 14, at least in a reference operating condition, for example the one in which the supply pressure (prail) assumes the maximum value allowed by the injection system, and the lift of the valve needle 12 assumes also the maximum value allowed (i.e., in operating conditions of maximum power or load).

[0076] Also Figure 5 is obtained from fluid-dynamic simulations made on a computer, and shows schematically the distribution of the velocity on three cylindrical surfaces within a wedge of the umbrella spray exiting from the nozzle 11, centred on the axis 5. In particular, the innermost cylindrical surface lies at the outflow area 14, whereas the other two lie on two different circumferences downstream of the outflow area 14. Figure 7 is similar to Figure 5 and shows some flow lines that, qualitatively, represent the paths of respective fluid fillets through the annular chamber 43 and downstream of the outflow area 14 in the combustion chamber 2.

[0077] At the outflow area 14, it may be noted how the velocity of the fuel film of the spray is not uniform along the circumference, but how it presents peaks of the velocity modulus in areas that are equal in number to the channels 67 and are substantially aligned with the outlets 72 along the respective axes 69. In other words, at the outflow area 14 the exiting fuel film is constituted by spray portions 75 that correspond to these areas at higher velocity, and by spray portions 76 that correspond to areas at lower velocity and are in an angular position intermediate between the channels 67. The difference in the velocity modulus between the maximum value and the minimum value must be appreciable, i.e., at least 10% of the maximum value.

[0078] The fuel film that exits from the outflow area 14 is not hence homogeneous in terms of velocity modulus, but has faster portions, the ones in the radial planes in which the axes 69 of the channels 67 lie, and slower portions, in the intermediate angular positions between the channels 67. By observing the flow lines L1 and L2 in Figure 7, both of them exit from the outlet 72 of the portion 68 at the same rate. In the first stretch of their own path, i.e., the one within the annular chamber 43, the particles of fuel along the flow lines L1 traverse a greater distance to reach the outflow area 14 as compared to the particles of fuel along the flow lines L2, which instead present a more direct path: this leads to a deceleration along the flow lines L1 with respect to the flow lines L2.

[0079] Immediately after the outflow area 14, the fuel film exiting therefrom is still intact and, given what has been said above, is not homogeneous in terms of velocity of the fluid fillets. As the fuel moves away from the outflow area 14, it encounters, however, the air present in the combustion chamber 2, which, as is known, exerts a deceleration force on the fuel film itself. This force is proportional to the square of the relative velocity between air and fluid fuel. Consequently, the fluid fillets along the flow lines L2 are subject to a deceleration force greater than that the fluid fillets along the flow lines L1.

[0080] The result is that, since the fluid fillets along the flow lines L2 are more hindered by the air, they tend to move away from the initial radial direction and to thicken laterally, i.e., towards radial planes that are intermediate between the axes 69 of the channels 67, so that in practice they thicken towards the fluid fillets that follow the flow lines L1. This phenomenon leads also to a delay in formation of the first drops, which, thanks to the thickening of the fluid fillets, will have a diameter greater than the thickness of the fluid film exiting from the outflow area 14. Furthermore, since the fluid fillets along the flow lines L1 are surrounded by the fluid fillets along the flow lines L2, they present favourable phenomena of mutual flow, which enable a greater penetration into the combustion chamber.

[0081] This consequence is visible in the sequence of experimental photographs of Figure 6, which have been taken along the axis 5, and which show the spray emitted by the atomizer 10 in successive instants during an injection of a prototype. At the moment of opening of the nozzle 11 and immediately after, the spray is substantially uniform along the circumference, whereas at the subsequent instants the spray appears progressively as being formed by a central umbrella-shaped part 77 and by a plurality of cusps or tentacles 78, which are in a number equal to that of the channels 67 and project from the perimetral edge of the central part 77. It is hence evident that the fluid fillets that formed the spray portions 75 concur with the fluid fillets of the spray portions 76 to form the cusps 78, with a greater penetration into the combustion chamber 2.

[0082] It should be pointed out that the photographic images appearing in Figure 6 refer to a test on a first prototype of atomizer, performed in quiescent chamber with gaseous means equivalent to air at a pressure of 30 bar and a temperature of 700 K, with a spray obtained by governing a lift of a few microns on a mean seal diameter of approximately 3 mm, with a relatively low injection pressure (700 bar). It may be hence understood that Figure 6 merely provided by way of illustration of the phenomenon.

[0083] In fact, by using higher injection pressures and a greater lift for the valve needle 12, there is an accentuation of the penetration, i.e., the cusps 78 are more defined and marked: the spray is very similar to the one obtained by an atomizer with solid-cone spray. Appropriate fluid-dynamic simulations made on a computer have, for example, highlighted spray patterns such as the ones illustrated in Figures 8A, 8B, and 8C. In particular, Figure 8A shows the evolution in time, for a series of successive instants, of the spray generated by an atomizer according to the method of the present invention, provided with five channels 67 each with a passage section area of 0.4 mm^2, with mean seal diameter of 3 mm, an injection pressure of 800 bar, and a lift of the valve needle 12 of 0.015 mm. As compared to the photographic sequence illustrated in Figure 6, in addition to the further cusp, there may be noted the smaller diameter of the central part 77. The diameter of the central part 77, as mentioned above, can also be modulated by means of variations of the lift and/or of the supply pressure, once the geometry of the atomizer 10 has been defined.

[0084] Figures 8B and 8C illustrate the comparison of two spray patterns, which also have different diameters of the central part 77 (the diameter of the central part 77 in Figure 8C is greater), obtained in this case in the same operating conditions of supply pressure and lift of the valve needle, but characterized by two different passage areas in the channels 67. The spray of the solution of Figure 8B is obtained with a minimum passage area of a single channel of 0.04 mm^2, as against a minimum passage area of 0.055 mm^2 for the channel of the solution of Figure 8C.

[0085] As mentioned above and as may be seen in Figure 5, starting from the outflow area 14, each of the spray portions 75 tends to divide into in two sub-portions 75a and 75b, basically as a result of the resistance opposed by the air in the combustion chamber 2. The sub-portions 75a and 75b generated by a given channel 67 move progressively away from one another in the circumferential direction, within the portions 76, as the distance of the fuel from the outflow area 14 increases. In other words, it is as if the flow lines followed by the fuel at a higher velocity were sucked back in the circumferential direction towards the areas where the fuel has a lower velocity. As this lateral drift of the path of the fuel proceeds faster, with respect to the original direction imposed by the channels 67 along the axes 69, the sub-portions 75a and 75b join up, in a way not illustrated, to the sub-portions 75b, 75a that have been generated by the adjacent channels 67. From this phenomenon it follows that the cusps 78 visible in Figure 6 are not aligned with the axes 69 of the channels 67, but are arranged, with respect to the axis 5, in angular positions that are intermediate between the channels 67, as already explained above in detail.

[0086] As mentioned above, to obtain this configuration of the fuel spray it is essential for the annular chamber 43 to have sufficiently small dimensions, also in relation to the type of fuel used, to the value of supply pressure (prail) and to the value of the lift of the valve needle 12 when the nozzle 11 is open. In particular, the further away the outflow area 14 is from the outlets 72, the more uniform is the velocity modulus of the fuel along the circumference at the outflow area 14, in so far as the velocity of the fuel exiting from the channels 67 has sufficient time and space to become uniform in the annular chamber 43 so that there is the risk that no cusp 78 will form. In the particular example of diesel as fuel, to obtain the cusps 78 in the reference operating condition, for example the one of maximum load or power (supply pressure prail and lift of the valve needle 12 equal to the maximum values allowed by the technologies normally used), it is preferable for the distance along the axes 69 between the outlets 72 and the outflow area 14 not to be greater than 1/3 of the mean diameter of the sealing seat 21. For instance, if this diameter is approximately 3 mm, the distance between the outlets 72 and the outflow area 14 is preferably smaller than or equal to 1 mm.

[0087] Also the shape and/or the volume of the annular chamber 43 can affect to a certain extent the profile of velocity of the fuel in the outflow area 14. In particular, a non-uniform profile of velocity is obtained that is increasingly evident as the volume of the annular chamber 43 decreases. For instance, in the case of diesel, to obtain the cusps 78 in a way sufficiently pronounced in operating conditions of high load, the maximum volume can be assumed as being equal to the volume of a hollow cylinder having an external diameter equal to the mean diameter of the sealing seat 21, a height equal to 1/3 of said mean diameter, and an internal diameter equal to 80% of the external diameter. As the volume of the intermediate chamber 43 increases, tendentially there is the effect of rendering the flow lines exiting from the portions 68 uniform, with the consequence of having a greater uniformity of the velocity at the outflow area 14 and hence a less pronounced effect of formation of the cusps 78.

[0088] A further factor that may affect the uniform or non-uniform profile of velocity of the fuel along the outflow area 14 is given by the minimum passage section area of each channel 67, as mentioned above. In fact, as said minimum passage section area decreases, it is possible to obtain a higher velocity of the fuel at the outlet 72 and hence a channelling and a more marked differentiation of the flow lines (L1 and L2) in the annular chamber 43, in the path of the fuel from the outlet 72 to the outflow area 14. Preferably, in the case of diesel fuel, for an injection system that operates with a maximum pressure of 2000 bar and that must deliver a maximum flow rate of approximately 70 g/s, to obtain the cusps 78 in a sufficiently marked way, for example in an operating condition of maximum load or maximum power, the passage section area of a single channel 67 at the outlet 72 is smaller than 0.05 mm^2.

[0089] In the design stage, once the engine and the injection system are known, the pressure of supercharged air (pcyl) and the supply pressure (prail) of the fuel are known and/or controllable. In particular, the atomizer 10 can be obtained via the following design steps:

- determining the amount of fuel to be injected into the combustion chamber 2 in a reference operating condition (for example, at full power or full load) in each single injection, possibly on the basis of the size and the application of the engine;
- determining the maximum value pmax, for example 1800 bar, that the supply pressure (prail) of the fuel supplied to the electro-injectors can reach;
- setting the number of cusps 78 (for example five cusps) that it is desired to obtain, also as a function of other parameters of the combustion chamber 2 (swirl index, dimensions, etc.); said number hence corresponds to the number of channels 67 to be provided by design and to be practically implemented;
- on the basis of the conformation of the combustion chamber 2, determining the angle of aperture of the cone of the surface 19 of the valve needle 12, for example 140°: this angle will define also the supply angle of the central part 77 of the hollow-cone spray;
- determining the maximum value possible for the lift of the valve needle 12 (for example, 20 μm), in particular on the basis of the actuator device 50 that has been chosen;
- determining the minimum value of the actuation time that can be managed with satisfactory precision (for example, 50 ms), on the basis of the accuracy of the control unit and of the actuator device 50 chosen;
- determining the minimum admissible value of the injection time into combustion chamber 2 for guaranteeing optimal combustion in the reference condition (for example, 600 μs) : in this way, having defined the maximum volume to be introduced, the admissible minimum instantaneous flow rate for the electro-injector 1 is defined;
- determining the mean diameter of the sealing seat 21 (for example, 2.5 mm) in such a way as to be able to respect the minimum injection time and guarantee the maximum fuel flow rate to be injected, and preferably in such a way that this mean diameter of seal is the smallest possible compatibly with the necessary structural strength to be assigned on the valve needle 12; and
- verifying that the lift of the valve needle 12 to obtain a pilot injection (equal, for example, to 1 mm^3 with an injection pressure of 1500 bar), with the minimum achievable actuation time, is preferably greater than 4 μm to guarantee a certain operating robustness during the pilot injections.

[0090] According to one aspect of the present invention, the process for producing the atomizer 10 envisages a design step in which the channels 67 are sized so as to respect the requirements of pressure and flow rate referred to above. In particular, using the system of equations indicated previously, and taking into account the energy losses (due to dissipation owing to fluid friction, turbulence, and yielding of heat to the atomizer 10), the total minimum passage section area of the channels 67 is determined in such a way that the mean pressure (p) in the annular chamber 43 is equal to (0.55 · pmax), when prail = pmax and when the lift of the valve needle 12 assumes the maximum value. In the conditions of the example we find that the cross section total is equal to 0.25 mm^2. The passage section area of each channel 67 will be obtained by dividing the total passage section area by the number of the channels established above (hence it will be equal to 0.05 mm^2).

[0091] Furthermore, according to a further aspect of the present invention, the atomizer 10 is obtained via one or more design steps that envisage appropriate sizing the annular chamber 43 to obtain as result the formation of the cusps 78 in the fuel spray, at least in a reference operating condition, for example the full-load condition. In particular, these design steps envisage appropriate positioning the outlets 72 of the channels 67 with respect to the sealing seat 21. Wishing to simplify this design step, as indicated above, the outlets 72 are positioned in such a way as to be located at an axial distance from the sealing seat 21 less than one third of the value of the mean diameter of seal previously calculated. In the example considered above, this distance will be less than 0.8 mm. Preferably, the inner diameter of the annular chamber 43 (i.e., the minimum diameter of the end 44) is greater than 80% of an external diameter so that in the example considered it will be greater than 2 mm.

[0092] From the above description, it is evident that the atomizer 10 enables an operation to be obtained that constitutes a compromise between the ones of the known art with solid-cone spray and the ones of the known art with hollow-cone spray.

[0093] In fact, by reducing the dimensions of the annular chamber 43 and, in particular, by approaching the outlets 72 to the outflow area 14, as compared to the solutions with hollow-cone spray of the known art, there is obtained a particular shape of the spray in the combustion chamber 2, i.e., a shape constituted by a central umbrella-shaped portion 77 and by a plurality of cusps 78, at least in a reference operating condition, for example in the one that corresponds to full power or full load of the engine. This particular shape of the spray enables a traditional CI mode to be obtained at high loads, i.e., a high penetration of the fuel into the combustion chamber 2, in a way similar to what happens with atomizers of the known art with solid-cone spray.

[0094] At the same time, at low and medium loads an operating mode of the HCCI type may possibly be maintained, with high atomization of the fuel and without cusps 78. Purely by way of example, in order not to cause the cusps 78 to appear in the injected fuel spray, it is possible to reduce the supply pressure (prail) in such a way as to reduce the velocity of the fuel at the outlets 72 and/or it is possible to set a relatively low lift of the valve needle 12 to have a greater backpressure in the annular chamber 43. With these operating modes (which evidently correspond to a fuel flow rate lower than of the one at full load), the annular chamber 43 can render the velocity of the fuel uniform notwithstanding its small dimensions to obtain thus a velocity modulus that is substantially uniform in the circular direction along the outflow area 14 in the operating conditions at low and medium engine loads.

[0095] Alternatively, it is possible to decide to size the passage areas of the portions 68 and/or the dimensions and/or shape of the annular chamber 43 in such a way as to have a spray pattern distinguished by very accentuated cusps 78, and then an extremely low central portion 77, also at low engine loads. This need may arise, for example, for particularly large combustion chambers 2, in which it is desired to prevent any concentration of fuel around the axis 5 of the electro-injector 1.

[0096] As regards the atomization of the drops of fuel in the spray delivered by the nozzle 11, the lateral drift of the flow lines L2 downstream of the outflow area 14 (Figure 7) also causes a partial re-thickening or coalescence of the drops of fuel at higher velocity, initially detached from the continuous film. These drops hence tend to increase in volume in the first part of their path. Thanks to this partial coalescence, the drops that will come to form the cusps 78 are larger and, hence, are distinguished by a higher kinetic energy and by a higher Weber number than the drops present in a spray with velocity modulus substantially constant along the circumference. It follows that the drops of fuel that will come to form the cusps 78 are more readily subject to a fragmentation into smaller drops in the second part of their path, namely, at the cusps 78. In other words, the behaviour of the drops of fuel that form the cusps 78 approaches decidedly what happens for the drops of fuel delivered by atomizers of the known art with solid-cone spray.

[0097] As mentioned above, in addition to the dimensions of the annular chamber 43, also the shape and dimensions of the channels 67 contribute to optimising the phenomenon of formation of the cusps 78 in the fuel spray, in so far as they determine a greater or smaller effect of guiding and channelling on the flow lines of the fuel exiting from the outlets 72 at the outflow area 14 and a greater or smaller conversion of the pressure into velocity, across the portion 45.

[0098] Furthermore, the surface roughness of the channels 67 and the conformation of the initial portions 70 can reduce the energy losses of the flow during traversal of the channels 67.

[0099] Moreover, as an alternative to or in combination with the reduction of the distance between the outlets 72 and the sealing seat 21, the geometry of the annular chamber 43 could be sized so as to have a non-homogeneous shape, i.e., a cross section that is not constant along the circumference so as to favour channelling and hence non-uniformity of the flow lines in the annular chamber 43, as may be seen by way of example in the variant of Figure 9.

[0100] In particular, by appropriately optimising the geometry of the annular chamber 43, it is possible to obtain the same effects of a non-uniform profile of velocity of the fuel along the outflow area 14 so as to be able to reduce the pressure difference and hence the conversion into velocity of the fluid in the channels 67, with the advantage of having lower energy losses.

[0101] For instance, the annular chamber 43 could be provided so as to create portions of divergent shape 80, i.e., with progressively increasing width, starting from the outlets 72 in the two opposite directions in the circumferential direction, as illustrated in the example of Figure 9. This divergence slows down further the fluid fillets L1, thus favouring formation of the cusps 78, as described above in detail.

[0102] Various modifications may in any case be made to the injection method, to the atomizer 10, and to the process of production that have been described with reference to the attached figures, and the generic principles described may be applied to other embodiments and applications, without thereby departing from the sphere of protection of the present invention, as defined in the annexed claims. Consequently, the present invention must not be considered as being limited to the embodiments described and illustrated, but it must be granted the widest sphere of protection in accordance with the principles and characteristics claimed herein.

[0103] In particular, the nozzle 11 could be defined by a end portion of the injector body 4, without being a distinct piece from the latter, and/or the guide portion 45 could form part of a body distinct from the nozzle 11, and/or the valve needle 12 could be operated directly, i.e., the injector 1 could be without the pressure chamber 62.

[0104] The shape of the annular chamber 43 could be different from the one illustrated by way of example in cross section in the attached figures.

[0105] The channels 67 could have a shape different from the preferred one that has been illustrated by way of example, and/or could be provided completely within the intermediate portion 45 of the stem 41, and/or could be in a number different from the one illustrated.

[0106] The channels 67 could be constituted only by the portions 68, i.e., have a constant passage section area along the axes 69 and hence be without the portions 70, 71.

[0107] As an alternative to an actuator of a piezoelectric or magnetostrictive type, an actuator of a solenoid type could be used, which, albeit operating basically only in two or three discrete positions, would be in any case able to generate a spray of the type illustrated in Figure 6, for example by regulating the injection pressure and/or the time of actuation of the electromagnet.

[0108] In addition, the atomizer 10 could be applied to fuels different from diesel, so that it might be necessary to set different dimensions of the annular chamber 43 and/or of the channels 67 to obtain an absence of uniformity of the profile of velocity of the fuel along the outflow area 14 and hence the same resulting effect of the cusps 78 illustrated in the attached figures.

[0109] Furthermore, particular shapes of the annular chamber 43, for example like the one of a convergent/divergent type illustrated in Figure 9, could be obtained by shaping the inner surface of the seat 13 of the nozzle 11, as an alternative or in combination to shaping of the stem 41 of the valve needle 12.

[0110] Finally, the channels 67 may be arranged in non-uniform positions about the axis 5, and closer to one another in the area of the combustion chamber 2 where a greater penetration of the spray is required. Especially in this case, it is also possible to obtain an asymmetry in the width or penetration of the cusps 78 in the spray itself.

Claims

1. An injection method for injecting fuel into a combustion chamber (2) of an internal-combustion engine; the method being carried out via an electro-injector (1) provided with an atomizer comprising a nozzle (11), which has:

    • a seat (13), which extends in a through way along a longitudinal axis (5);
    • a front surface (17), which is external to said seat (13); and
    • a sealing seat (21), which joins a first surface (18) of said seat (13) to said front surface (17);

    said atomizer further comprising a valve needle (12), which comprises:

    • a head (20) designed to couple to said sealing seat (21); and
    • a stem (41), which has a diameter smaller than said head (20), projects axially from said head (20) and engages said seat (13); said stem (41) and said nozzle (11) defining radially between them a passage (16), which is designed to get fuel to flow and comprises an annular chamber (43), which axially ends at said sealing seat (21); said stem (41) comprising an intermediate portion (45) coupled in an axially sliding way to a second surface (66) of said seat (13) and defining a plurality of channels (67), which have respective outlets (72) at said annular chamber (43);

    the method comprising the steps of:

    - supplying pressurized fuel into said passage (16); and
    - axially displacing said valve needle (12) along an opening stroke directed axially towards the outside of said seat (13), starting from a closing position in which said head (20) is coupled to said sealing seat (21), so as to obtain an injection of fuel into said combustion chamber (2); said sealing seat (21) and said head (20) defining an outflow area (14), which is annular and has a width that increases as the opening stroke of said valve needle (12) progresses;

    characterized in that the fuel is injected into said combustion chamber (2) in such a way as to have a non-uniform velocity modulus at said outflow area (14), as the position in the circumferential direction varies, at least in a reference operating condition of said engine.

2. The method according to claim 1, **characterized in that**, at said outflow area (14), the difference between the maximum value of the velocity modulus and the minimum value of the velocity modulus is at least equal to 10% of said maximum value.

3. The method according to claim 1 or claim 2, **characterized in that**, at said outflow area (14), the velocity modulus has a plurality of peaks, which occur in spaced apart positions, are in a number equal to that of said channels (67) and are substantially aligned with said outlets (72).

4. An atomizer (10) of a fuel electro-injector (1) for carrying out the method of any one of the preceding claims, the atomizer comprising a nozzle (11), which has:

- a seat (13), which extends in a through way along a longitudinal axis (5);
- a front surface (17), which is external to said seat (13); and
- a sealing seat (21), which joins a first surface (18) of said seat (13) to said front surface (17);

said atomizer further comprising a valve needle (12), which comprises:

- a head (20) designed to couple to said sealing seat (21); and
- a stem (41), which has a diameter smaller than said head (20), projects axially from said head (20) and engages said seat (13); said stem (41) and said nozzle (11) defining radially between them a passage (16), which is designed to get fuel to flow and comprises an annular chamber (43), which axially ends at said sealing seat (21); said stem (41) comprising an intermediate portion (45) coupled in an axially sliding way to a second surface (66) of said seat (13) and defining a plurality of channels (67), which have respective outlets (72) at said annular chamber (43);

said valve needle (12) being axially mobile along an opening stroke directed axially towards the outside of said seat (13), starting from a closing position in which said head (20) is coupled to said sealing seat (21); said sealing seat (21) and said head (20) defining a outflow area (14), which is annular and has a width that increases as the opening stroke of said valve needle (12) progresses;
characterized in that said annular chamber (43) has dimensions and/or a shape such as to inject fuel with non-uniform velocity modulus at said outflow area (14), as the position in the circumferential direction varies, at least in a reference operating condition of said engine.

5. The atomizer according to claim 4, **characterized in that** the axial distance between said outlets (72) and said sealing seat (21) is less than or equal to one third of the mean diameter of said sealing seat (21).

6. The atomizer according to claim 4 or claim 5, **characterized in that** the volume of said annular chamber (44) is smaller than or equal to a maximum volume equal to the volume of a cylinder, which has an external diameter equal to the mean diameter of said sealing seat (21), a height equal to one third of said mean diameter, and an internal diameter equal to 80% of said mean diameter.

7. The atomizer according to any one of claims 4 to 6, **characterized in that** said channels (67) are in a number greater than or equal to three and are set at equal distances apart about said longitudinal axis (5).

8. The atomizer according to any one of claims 4 to 6, **characterized in that** said channels (67) are arranged in non-uniform angular positions about said longitudinal axis (5).

9. The atomizer according to any one of claims 4 to 8, **characterized in that** said channels (67) comprise respective channelling portions (68), which define a minimum passage section area of said channels (67), extend along respective channelling axes (69) parallel to said longitudinal axis (5), and have constant passage section areas along the respective channelling axes (69).

10. The atomizer according to claim 9, **characterized in that** said channelling portions (68) extend axially throughout the entire axial length of said intermediate portion (45).

11. The atomizer according to claim 9, **characterized in that** said channels (67) comprise respective lead-in portions (71), which are arranged upstream of said channelling portions (68), when considering the direction of the fuel towards said sealing seat (21), and define a passage section area that decreases progressively as far as said channelling portions (68).

12. The atomizer according to any one of claims 4 to 11, **characterized in that** the opening stroke of said valve needle (12) has a maximum lift, and **in that** said channels (67), as a whole, define a minimum passage section area that is greater than the width of said outflow area (14) even when the opening stroke reaches said maximum lift.

13. The atomizer according to any one of claims 4 to 12, **characterized in that** said annular chamber (43) comprises, at each said outlet (72), pairs of portions (80), which have a diverging shape starting from said outlet (72) in the two opposite directions in the circumferential direction.

14. A process for producing an atomizer (10) according to any one of claims 4 to 13, **characterized by** comprising the steps of determining a reference pressure for the supply pressure (prail) of the fuel supplied to said atomizer (10), and establishing a reference lift for the opening stroke of said valve needle (12), and **characterized in that** said annular chamber (43) and/or the passage area for the fuel in said channels (67) is/are sized in such a way as to inject fuel with non-uniform velocity modulus at said outflow area (14), as the position in the circumferential direction varies, at least in an operating condition in which the

supply pressure (prail) is equal to said reference pressure and the opening stroke is equal to said reference lift.

15. The process according to claim 14, **characterized in that** the passage area for the fuel in said channels (67) is sized in such a way that said intermediate portion (45) generates a pressure difference smaller than or equal to 45% of the reference pressure.

16. The process according to claim 14 or claim 15, **characterized in that** the passage area for the fuel in said channels (67) is sized in such a way that the difference between the maximum value of the velocity modulus and the minimum value of the velocity modulus is at least equal to 10% of said maximum value.

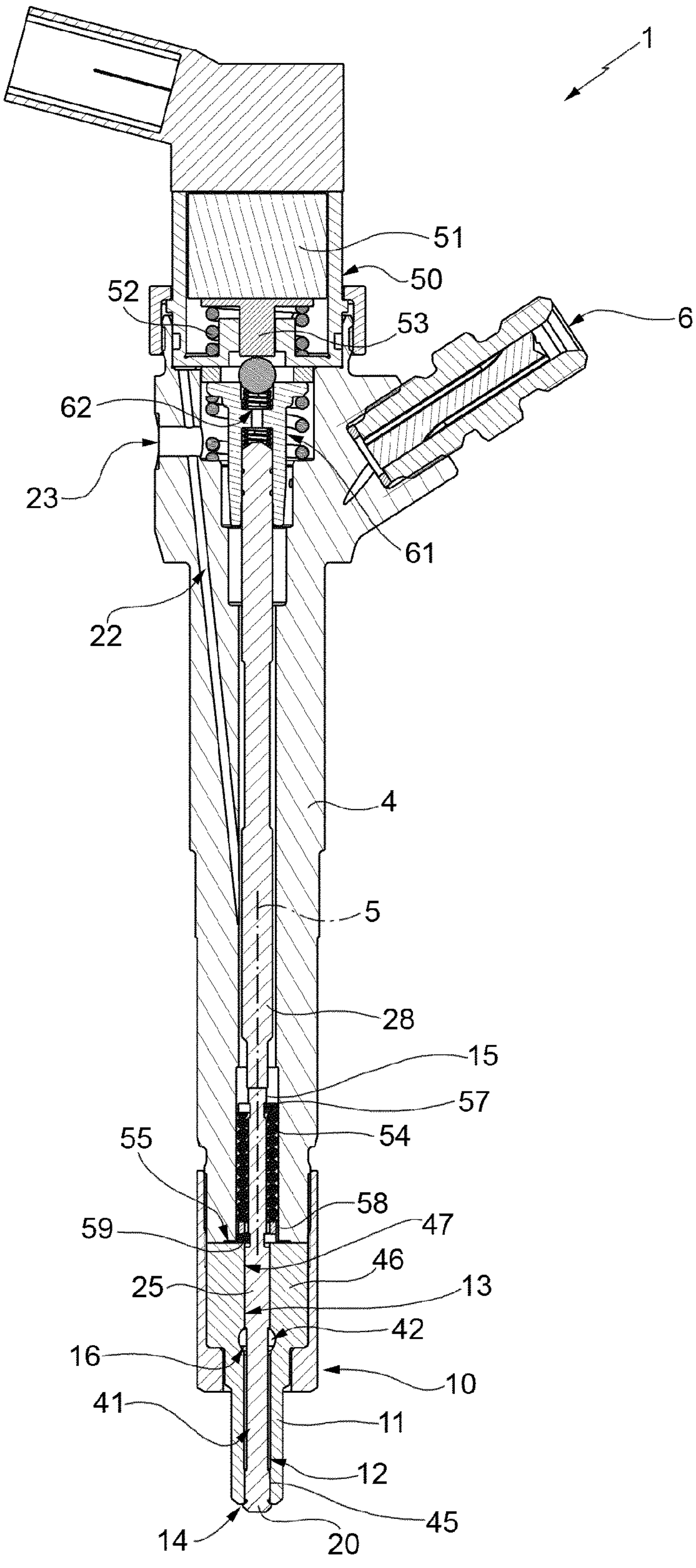
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FIG. 1

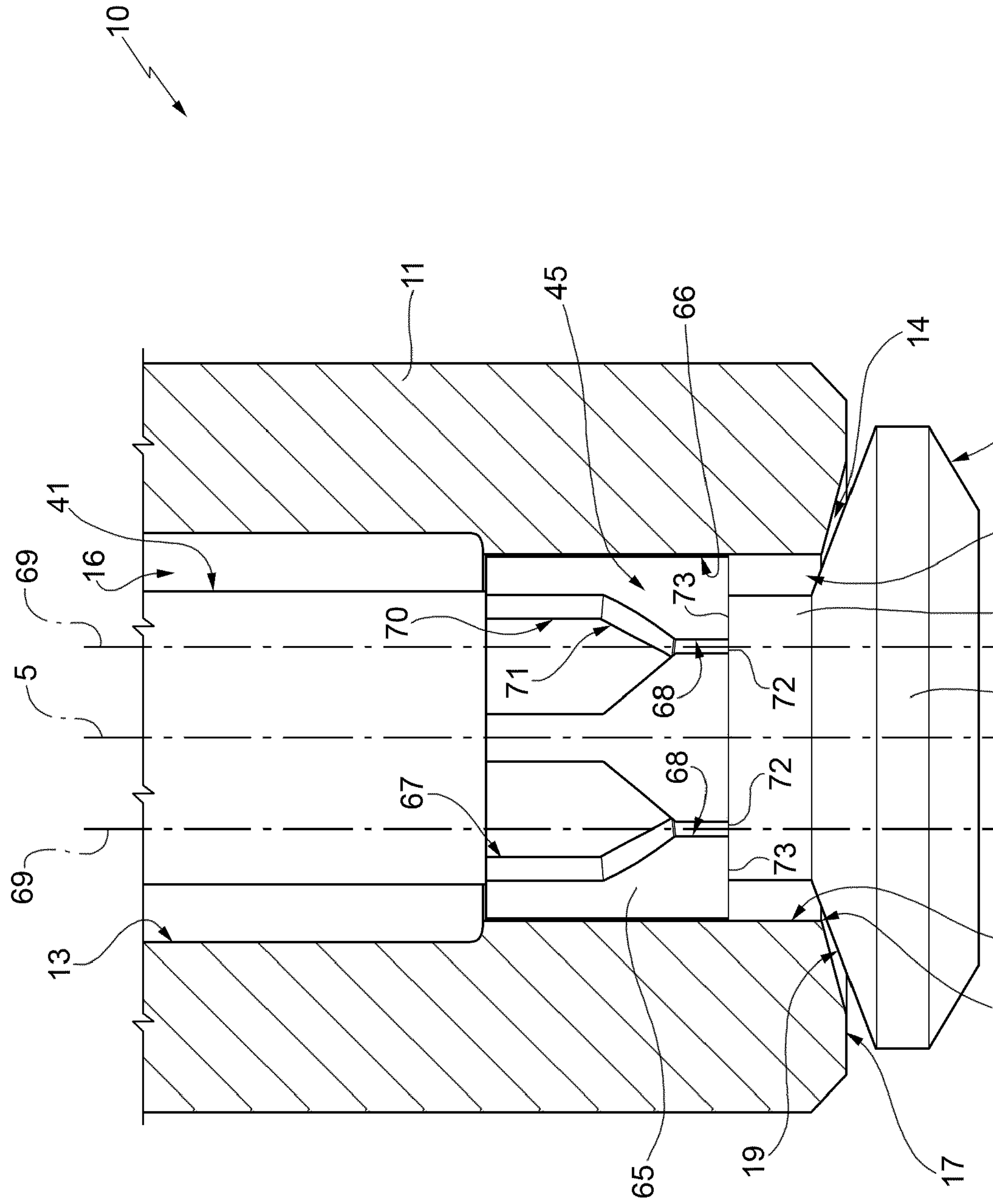
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FIG. 2

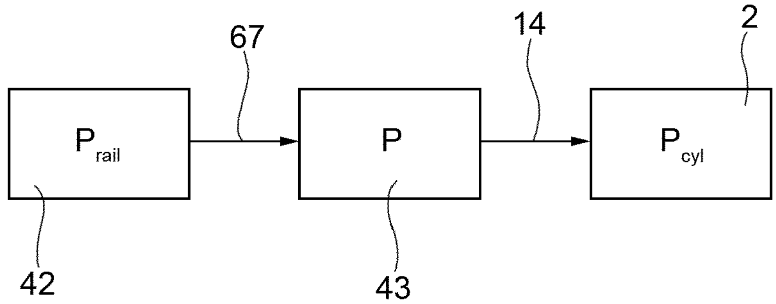

FIG. 3

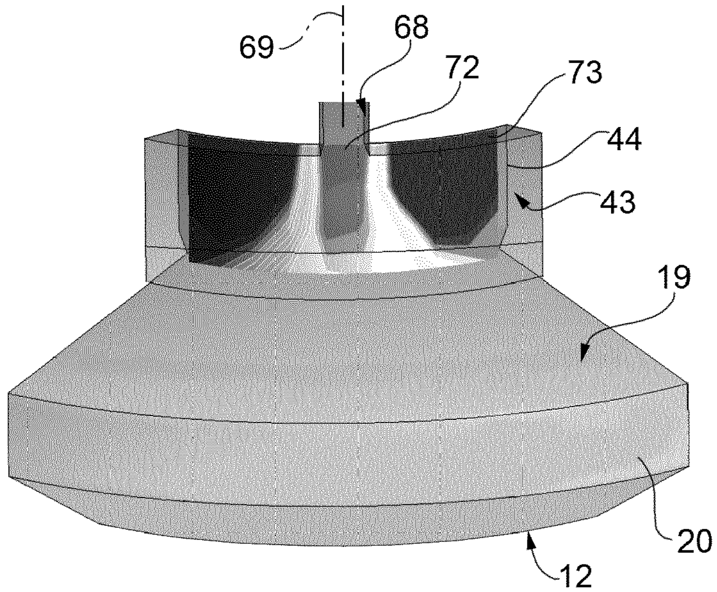

FIG. 4

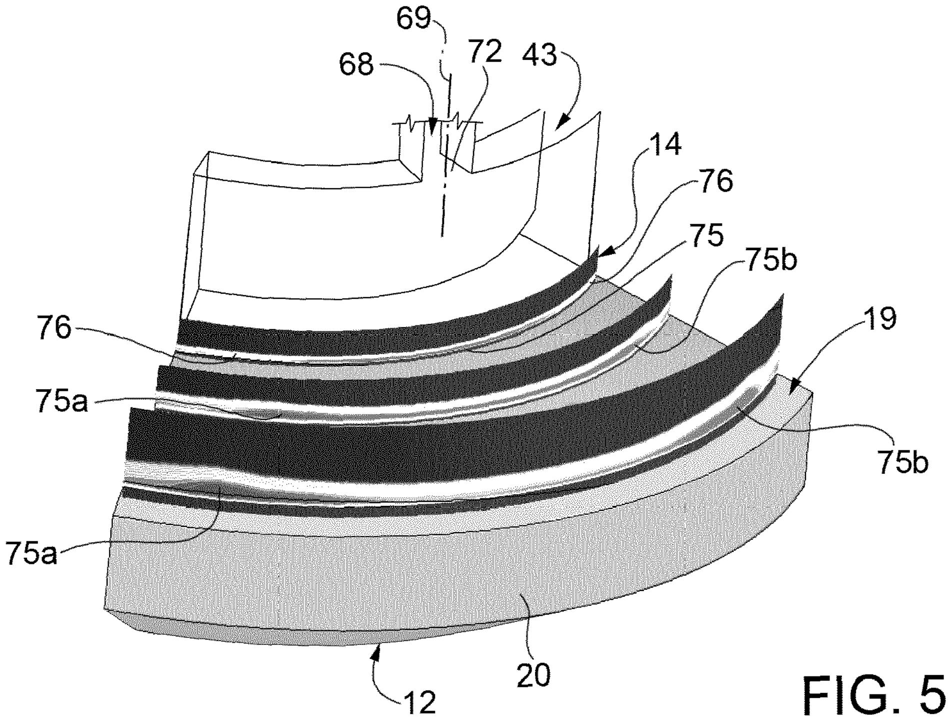
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FIG. 5

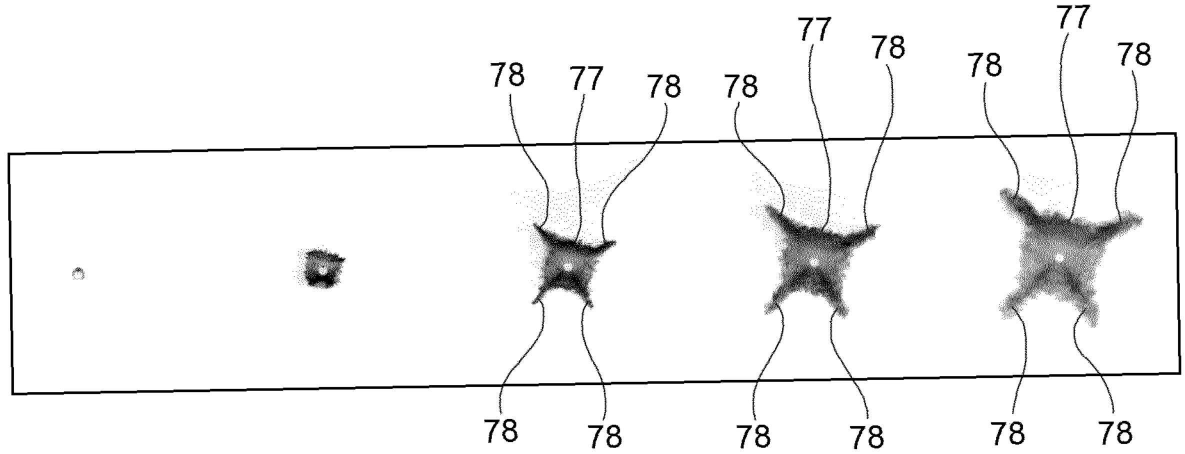
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FIG. 6

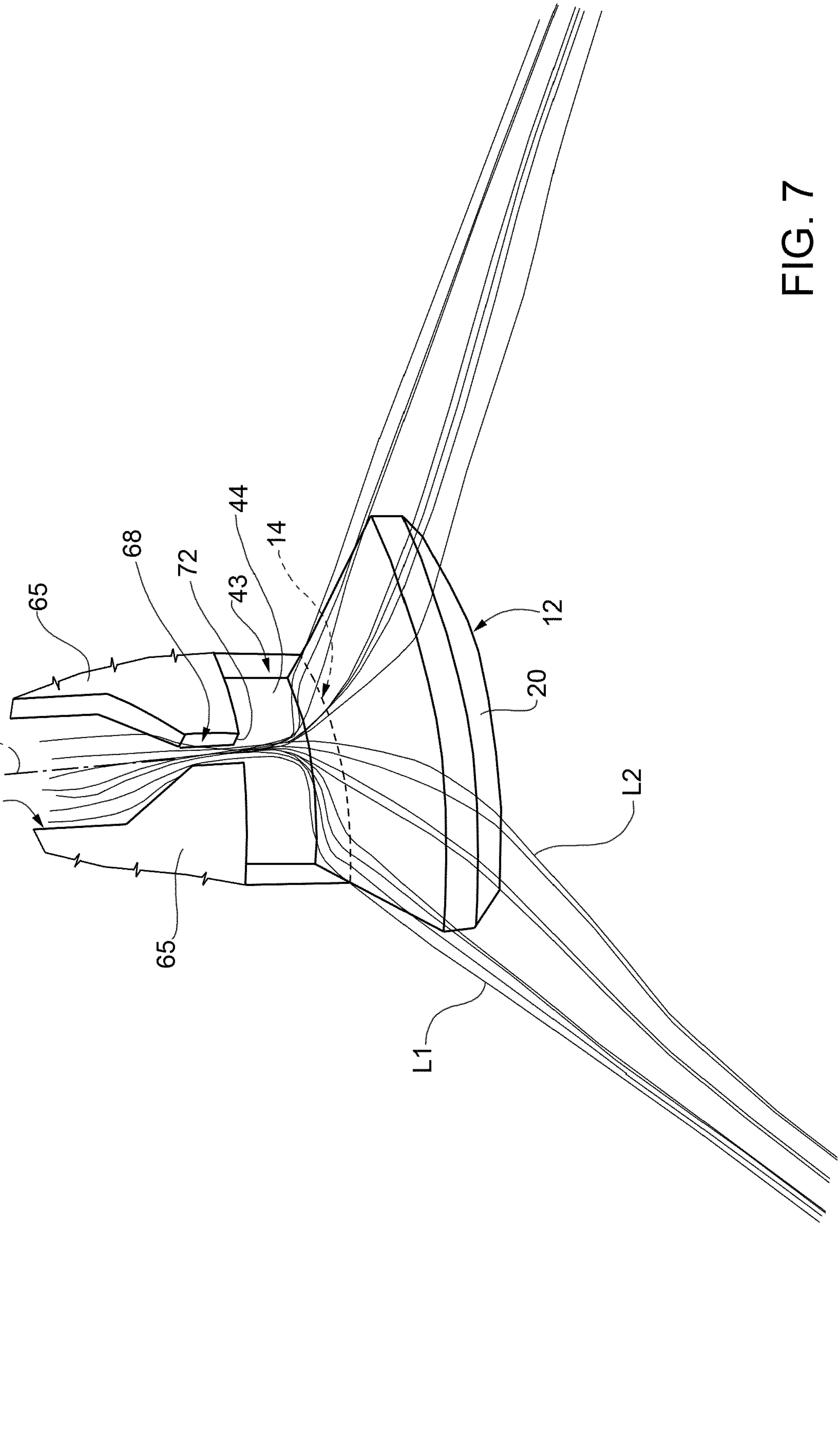

FIG. 7

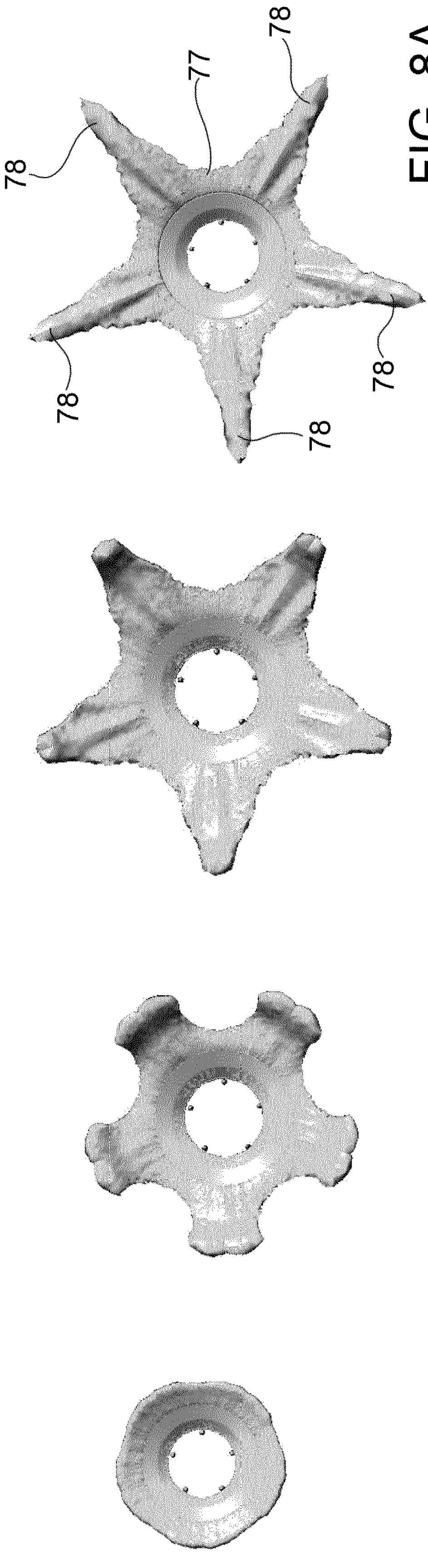

FIG. 8A

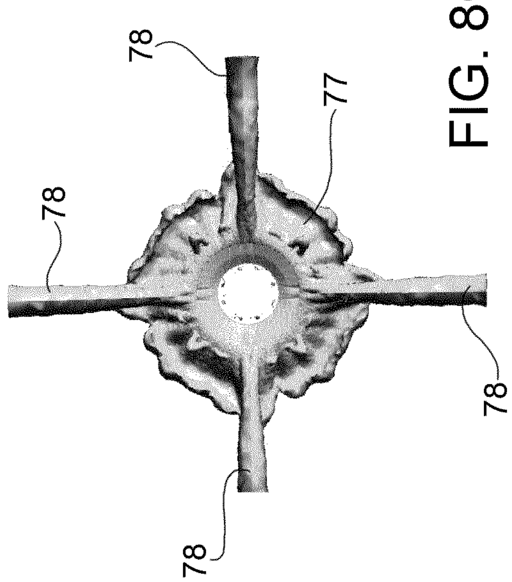

FIG. 8C

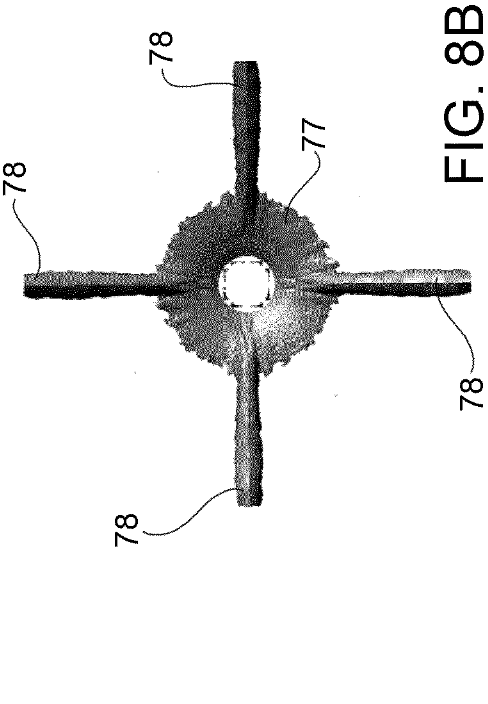

FIG. 8B

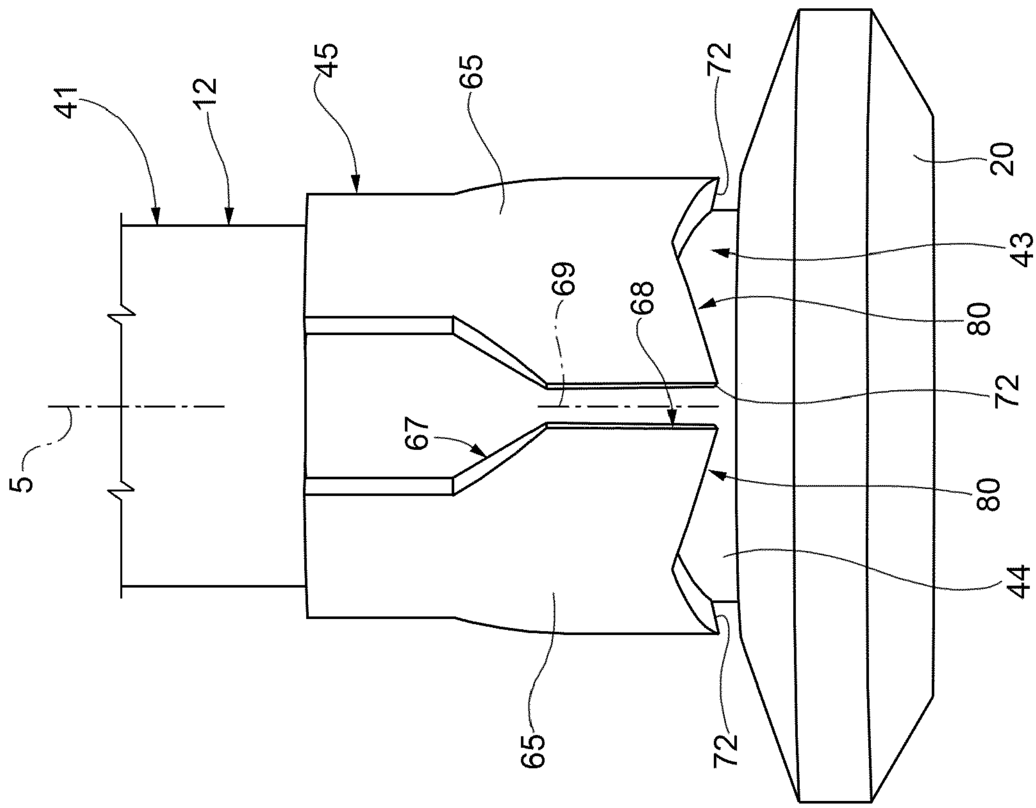

FIG. 9

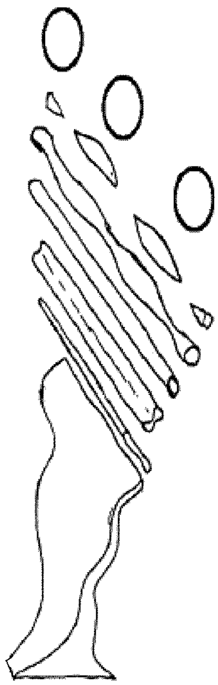

FIG. 10A

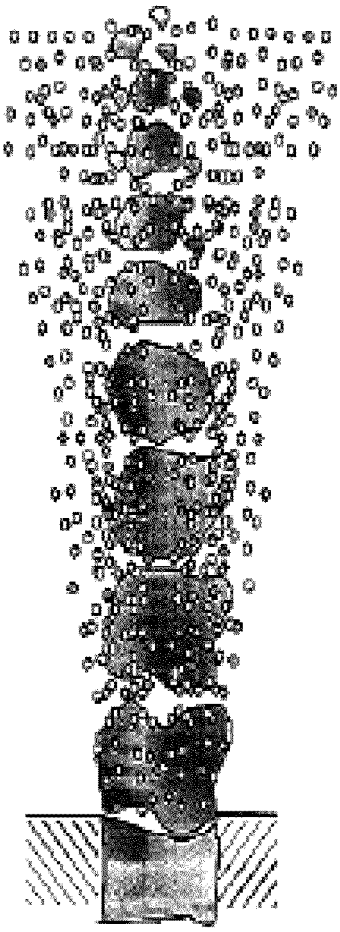

FIG. 10B

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EUROPEAN SEARCH REPORT

Application Number
EP 15 19 3750

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X A	US 5 829 688 A (REMBOLD HELMUT [DE] ET AL) 3 November 1998 (1998-11-03) * column 8, lines 5-57 * * column 9, lines 43-54; figures 1,6-9 *	1-12, 14-16 13	INV. F02M61/18 F02M61/04 F02M61/06 F02M61/08 F02M61/12
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The present search report has been drawn up for all claims

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Place of search	Date of completion of the search	Examiner
The Hague	6 April 2016	Boye, Michael

CATEGORY OF CITED DOCUMENTS

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