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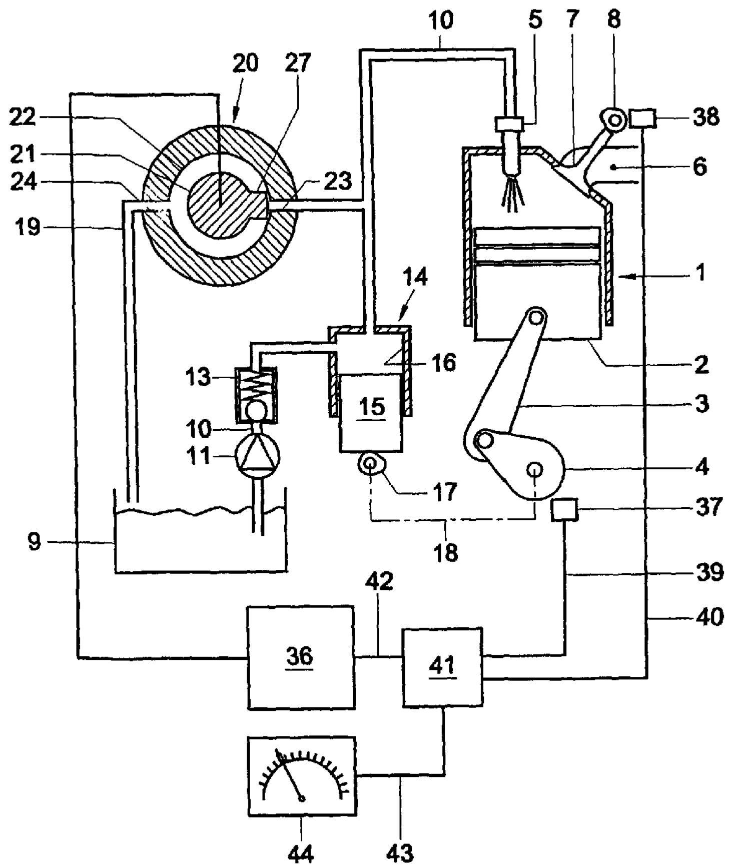
(54) **Title:** APPARATUS, FUEL INJECTION SYSTEM, AND METHOD FOR METERING FUEL



(57) Abstract: A fuel injection system has a rotatable valve body (21) which releases a spill channel (19) in a first angular range and closes off the spill channel (19) in a second angular range. Duration and times of start and end of a fuel injection are determined by duration and times of start and end of the closure of the spill channel (19). Periodic closure of the spill channel (19) occurs in accordance with the position of the engine, with the valve body (21) rotating continuously. During each working cycle of the combustion engine, the angular speed of the valve body (21) is again varied back and forth for determining duration and times of start and end of the fuel injection, but the average angular speed of the valve body (21) is in a constant ratio to the angular speed of the crankshaft. An apparatus for converting an existing fuel injection system and a method for metering fuel are also described.

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Title: Apparatus, fuel injection system, and method for metering fuel

The invention relates to an apparatus for metering the output of a fuel injection system according to the introductory portion of claim 1, to a fuel injection system according to the introductory portion of claim 7, and to a method for the metered delivery of fuel to a combustion engine according to the introductory portion of claim 13.

Such an apparatus, such a fuel injection system and such a method are known from European patent application 0 024 803.

Such apparatuses, fuel injection systems and methods are also known from European patent applications 0 026 583 and 0 027 682, international patent application WO 82/03890 and the British patent application 2147954.

In these known apparatuses, fuel injection systems and methods, the fuel pump always delivers its maximum output, and the metering of fuel, at least for the purpose of the main injection - which is variable to enable regulation of the engine power to be delivered - is carried out by draining (spilling) excess fuel at the desired metering, which excess fuel is led back to the tank again. Through such spilling, the duration and the time of start and end of the injection can be accurately controlled.

Such controlled spilling of fuel for metering the fuel delivery via an injector is then realized by two rotatable valve bodies incorporated in parallel channels. The injection duration is controlled by adjusting the mutual angular displacement of the rotatable valve bodies. One valve body is then the last to close one of the parallel channels and thus determines the start of the injection of fuel in that the spilling of fuel is terminated. The other one of the valve bodies is thereupon the first to clear one of the parallel channels and thus determines the end of the injection of fuel in that the spill of fuel is resumed again.

A drawback of this manner of metering is that the construction required therefor is fairly complicated and costly.

It is an object of the invention to provide a solution which makes it possible to realize the metering of fuel by spilling fuel while the pump delivers fuel, with a simpler and less costly construction.

According to the present invention, this object is achieved by providing an apparatus according to claim 1, with which an existing fuel injection system can be converted. This object is further achieved according to the present invention by providing a fuel injection system according to claim 7 or a method according to claim 13.

For the use of the proposed apparatus, the proposed fuel injection system or the proposed method, due to the variation, during each working cycle of the engine, of the relative angular speed of the valve body with respect to the angular speed of the crankshaft, there is no need, for the purpose of metering the main injection via a particular injector, for two rotatable valve bodies in parallel channels with provisions for controlling the mutual angular displacement.

Particularly advantageous embodiments of the invention are set forth in the dependent claims.

Further objects, features, effects and details of the invention appear from the following description of an exemplary embodiment represented in the drawing. In the drawing:

Fig. 1 is a schematic representation of an example of a fuel injection system according to the invention and parts of a combustion engine cooperating therewith;

Fig. 2 is a perspective representation of a rotatable valve body for, for instance, a fuel injection system as represented in Fig. 1;

Fig. 3 is a side elevation in cross section along the line III-III in Fig. 4 of a spilling valve structure with a rotatable valve body as represented in Fig. 2;

Fig. 4 is a side elevation in cross section along the line IV-IV of Fig. 3;

Fig. 5 is a diagram in which the operation of different functions of the method according to the invention is represented in relation to the displacement of the crankshaft of the engine at a first average speed of the rotatable valve body;

Fig. 6 is a diagram as in Fig. 5, but at a second average speed of the rotatable valve body; and

Fig. 7 is a diagram as in Figs. 5 and 6, but at a third average speed of the rotatable valve body.

The apparatus, the fuel injection system and the method for metering fuel as described hereinafter represent the embodiments of the invention that are presently preferred most.

Fig. 1 shows a cylinder 1 of a combustion engine, having therein a piston 2 which is connected through a drive shaft 3 to a crankshaft 4. Located above the piston 2 is a combustion chamber in which terminate an injector 5 for injecting fuel and ports for taking in and exhausting fuel. Of these ports, the drawing shows the inlet port 6, which is closable by a valve 7 operated by a camshaft 8. The engine represented in the drawing is a piston engine working according to the diesel principle. It is also possible, however, to apply the proposed manner of fuel metering with advantage to a different type of combustion engine with intermittent fuel delivery, such as a gasoline engine with ignition, in combination with direct fuel injection into the combustion chamber or in combination with fuel injection into an inlet channel. The proposed manner of fuel metering can also be advantageously combined with an engine of a different geometry, such as a Wankel engine.

Fig. 1 further shows a fuel tank 9 and a fuel injection system with a fuel supply channel 10 which includes a low-pressure fuel pump 11 for drawing fuel from the tank 9.

Downstream of the fuel pump 11, a check valve 13 is included in the fuel supply channel 10. To a downstream end of the fuel supply channel 10 connects the injector 5 of the fuel injection system. A high-pressure pump 14 has a piston 15 which is reciprocable in a cylinder 16 of that pump 14. Provided for the purpose of driving the movements of the piston 15 is a camshaft 17 which is coupled with the crankshaft 4, as is represented by a chain-dotted line 18. The transmission between the camshaft 17 of the pump 14 and the crankshaft 4 forms a 1:2 reduction, so that the camshaft 17 rotates at a speed which is half of the speed of the crankshaft 4. The piston 15 of the pump 14 thus makes a stroke each time when the piston 2 of the engine makes two strokes. This relationship between the movements of the pistons 2, 15 can naturally be effected in many other ways.

The cylinder 16 of the high-pressure pump 14 communicates with the fuel supply channel 10 for periodically taking up fuel supplied by the low-pressure pump 11 and periodically delivering fuel under high pressure. The injector 5 is provided, in a manner known per se, with a valve which allows fuel to pass only when it has been brought under a particular minimum pressure. However, for the purpose of feeding fuel to the high-pressure pump 14, it is also possible to apply other provisions than a low-pressure pump 11, such as bringing the tank 9 under an excess pressure.

Between the high-pressure fuel pump 14 and the injector 5, there is connected to the fuel supply channel 10 a spill channel 19 for draining fuel away from the injector 5.

The spill channel 19 extends through a spill valve structure 20 (see also Figs. 2-4) with a rotatable valve body 21 in the spill channel 19. The valve body 21 is situated in a chamber 22 with an inlet port 23 and an outlet port 24.

During rotation through a first angular range, the valve body 21 releases the spill channel 19. To that end, according to the example shown, the valve body is provided with a groove 25 extending in a circumferential sense over a portion of the circumference in a cylindrical surface 26 of the valve body 21. According to this example, the first angular range is the angular range in which the groove 25 in the cylindrical surface 26 of the valve body 21 is situated in front of the inlet port 23. Instantaneous views of this situation are represented in Figs. 3 and 4.

During rotation through a second angular range, the valve body 21 closes the spill channel 19 off. According to this example, the second angular range is the angular range in which an interruption 27 of the groove 25 in the cylindrical surface 26 of the valve body 21 is situated in front of the inlet port 23. An instantaneous view of this situation is represented in Fig. 1. It is noted that the valve body 21 also closes off the spill channel 19 if the interruption of the groove is situated in front of the outlet port 24. However, this does not lead to delivery of fuel via the injector, because this does not coincide with supply of fuel under high pressure by the high-pressure fuel pump 14. If the closure of the spill channel, adjacent the outlet port 24, were yet to lead to undesired delivery of fuel, the outlet port 24 can be provided with a slot in circumferential sense which is longer than the length of the interruption in circumferential sense, so that the interruption cannot close off the outlet port 24.

As is represented in Fig. 4, the valve body is rotatably bearing-mounted in bearings 28, 29 which, in closing pieces 30, 31, are positioned with respect to a housing 32 of the valve structure 20. The housing 32 is provided with bores 33, 34 with internal thread for connecting upstream and downstream conduits through which extends the fuel supply channel. Further, the housing 32 is provided with a bore 35 with internal thread for connecting a downstream conduit through which extends the spill channel 19.

For rotating the rotary valve body 21, the fuel injection system is equipped with a drive 36, in this example in the form of a servomotor 36, which is schematically represented in Fig. 1. For controlling the servomotor 36, there is provided a control structure with sensors 37, 38 which signal positions of the crankshaft 4 and the camshaft 8. The sensors are connected via lines 39, 40 with a processing structure, in this example in the form of a microcontroller 41, and produce a signal each time when the crankshaft 4 and the camshaft 8, respectively, have reached a particular position. Thus, the angular displacement of the running combustion engine is signaled to the processing structure 41. The microcontroller 41 is connected via a line 42 with the servomotor 36 for actuating the servomotor 36. Thus, the microcontroller 41 can control the servomotor 36 in accordance with the signaling received from the sensors 37, 38.

The microcontroller 41 is further connected via a line 43 with an operating interface 44 for operating the engine. According to this example, the interface is designed as an operating handle 44 of a type used, for instance, on the bridge of a ship.

The microcontroller 41 is arranged for actuating the servomotor 36 for the purpose of varying back and forth the angular speed of the valve body 21 during each working cycle of the combustion engine, so that the times of start and end of the traversal of the second angular range – in which the spill of fuel is blocked and fuel is injected via the injector – are determined in accordance with the signals received from the sensors 37, 38 and the setting of the operating handle 44.

Further, the microcontroller 41 is arranged for compensating variations in angular speed of the valve body 21 during the traversal of the first angular range, for the purpose of maintaining an average angular speed of the valve body 21 over at least one working cycle of the combustion engine, which corresponds with a current speed of the combustion engine.

Fig. 5 shows an example of the operation of the proposed system during an operating condition, wherein the fuel supply is to a small extent limited with respect to the maximum fuel supply.

The upper two lines of the diagram shown in Fig. 5 (S CRANKSHAFT and S CAMSHAFT) represent, in relation to the position of the crankshaft 4, pulse signals which the sensors 37, 38 produce while the combustion engine runs. The pulses coming from the crankshaft sensor 37 in each case indicate that the crankshaft 4 passes a position 90° before the position associated with the piston 2 reaching the upper dead center. To enable distinction between the suction stroke and the power stroke, the camshaft sensor 38 too generates a pulse each time when the camshaft 8 (which rotates at a speed equal to half the speed of the crankshaft) has made a complete revolution. These pulses are produced each time when the camshaft 8 reaches a position which precedes by 45 degrees' rotation of the camshaft the position of the camshaft at the moment when the piston 2 reaches the upper dead center after the compression stroke.

The piston 15 of the high-pressure fuel pump 14 is driven by the shaft 17 which likewise rotates at a speed equal to half the speed of the crankshaft 4. The piston 15 of the high-pressure fuel pump 14 makes a working stroke each time when the piston 2 of the engine completes the compression stroke and passes the upper dead center. Furthermore, a possible fuel output is involved only when the pump pressure is at least above a particular minimum value. All this leads to potentially effective pump yield during the hatched phases in the third line from the top (PUMP ACTIVE).

The instantaneous angular speed ω of the valve body is represented (not on scale) in the bottom line of the diagram (ω VALVE). In the operating condition shown by way of example, the angular speed of the rotary valve body 21, each time when it approaches the second angular range – in which the spill channel 19 is closed off – is slightly raised and reduced again after the middle of that angular range has been passed. Thus the valve body 21 is in the second angular range, in which it closes the spill channel 19, for a shorter time than would be the case if the valve body 21 were to rotate at an angular speed which is in a continuously constant ratio to the angular speed of the crankshaft. The periods in which the valve 20 is closed in the operating condition shown by way of example, are represented in the diagram in the line VALVE CLOSED. After increasing the angular speed of the valve body 21, the angular speed of the valve body 21 is each time lowered to slightly below the intended average speed over a working cycle of the combustion engine (720° rotation of the crankshaft in the proposed four-stroke combustion engine), to compensate the temporary raise of the speed for the purpose of shortening the closing time.

The periods during which the spill valve 20 is closed essentially correspond to the periods during which delivery of fuel via the injector 5 in effect takes place, as appears from the line FUEL INJECTION in Fig. 5.

As the line OVERFLOW in Fig. 5 indicates, during the periods that the high-pressure pump 14 is active for supplying fuel to be potentially injected, but the spill valve 20 is not closed, fuel is drained via the spill channel 19 and led back to the tank 9.

Inasmuch as during each working cycle of the combustion engine the instantaneous angular speed of the valve body 21 is again varied up and down for determining times of start and end of the closure of the spill channel 19, while maintaining an average angular speed of the valve body 21 per working cycle of the engine that corresponds with the current speed of the combustion engine, the main injection of fuel – which varies strongly depending on the desired engine power as indicated by the operating handle 44 – can be metered with a rotation-spill valve 20 with a single rotary valve body 21, and a single drive 36 for controlling the rotation of that valve body 21 will suffice.

If a pre-injection is carried out, this can be controlled with a separate valve, but it can also be controlled with the same valve body 21 as the main injection. In the latter case, the advantage is achieved that per injector only one rotatable valve is necessary and that the duration of the pre-injection varies along with the duration of the main injection.

The invention can be applied with particular advantage for improving the metering and the accuracy of the time of injection of fuel by an existing fuel injection system. In older fuel injection systems, it is customary for the output to be varied by varying the stroke of the high-pressure pump. In this way, however, the control of the course of the fuel injection and the moment of injection is fairly rough, so that the fuel consumption and the emissions of noxious substances are fairly high. Such existing fuel injection systems can be considerably improved, for instance, by mounting of an apparatus comprising the following parts of the fuel injection system according to the proposed example: the spill valve 20, the drive 36 for rotating the rotatable valve body 21, and the above-described control structure with the sensors 37, 38 for signaling angular displacement of a running combustion engine and with the processing structure 41 operatively coupled to the sensors 37, 38.

The control of the output of the high-pressure pump of the existing combustion engine must then be fixed at maximum, and excess fuel is led back via the spill channel to the tank.

If the proposed fuel injection system is applied to a combustion engine with several injectors, the numbers of the rotary valve bodies 21 and drives 36 are preferably each equal to the number of injectors. Then, on the one hand, the number of rotatable valve bodies 21 and drives 36 is limited to one per injector, so that a limited number of valve bodies 21 and drives 36 are needed. On the other hand, in that case, each valve body 21 only needs to control the closure and release of one injector, so that between the periods in which the valve body 21 closes off the spill channel 19 and fuel is injected, there is relatively much time available for compensating accelerations or decelerations for the purpose of controlling the duration of injection.

In combustion engines operatively running at loads varying between substantially zero-load and full-load, the required amount of fuel per injection varies quite strongly. As a consequence, the angular speed of the valve body in the extreme operating conditions must each time be varied very strongly to enable the required adjustment of the closure duration of the valve 20 to the required fuel metering.

To enable limitation of the necessary accelerations and decelerations per working cycle of the combustion engine, the control structure 37-41 and the drive 36 according to this example are arranged for switching from a first average angular speed of the valve body 21 over at least one working cycle of the engine, to a second average angular speed of the valve body 21 over at least one working cycle of the engine. The average angular speeds of the valve body 21 then each amount to half the angular speed of the crankshaft 4 or a whole multiple thereof.

In the operating condition as represented in Fig. 6, the power required from the combustion engine is lower than in Fig. 5 (assuming that the speed is approximately the same). The injection duration indicated in the line FUEL INJECTION is correspondingly shorter than the injection duration according to Fig. 5. To achieve this, the average angular speed of the valve body 21 has been raised from half of the angular speed of the crankshaft 4 to a speed equal to the angular speed of the crankshaft 4. As a result, the valve 20 closes each time when the crankshaft 4 reaches the upper dead center position, that is, also when no injection of fuel is desired. However, because in that case the high-pressure pump 14 does not force any fuel under pressure to the injector 5, this does not lead to the injection of fuel.

In the operating condition represented by way of example in Fig. 6, the required fuel delivery is greater than that which would be obtained upon rotation of the valve body 21 at an angular speed which is in a fixed ratio to the angular speed of the crankshaft. To realize the required fuel delivery, the speed of the rotatable valve body 21 has been reduced each time during traversal of the second angular range, in which the valve body

21 closes the spill channel 19, and the angular speed during the traversal of the first angular range has been raised to compensate the deceleration applied. Thus, the average angular speed of the valve body 21 in the operating condition represented in Fig. 6 is twice as high as in the operating condition represented in Fig. 5, but the fuel metering according to Fig. 6 amounts to more than half of that according to Fig. 5.

In order to further enlarge the control range of the fuel metering, in the fuel system according to this example, the average angular speed of the valve body 21 over at least one working cycle of the combustion engine can be further raised to twice the speed of the crankshaft 4. In that operating condition, of which Fig. 7 represents an example, the valve body closes the spill channel 19 four times per working cycle of the combustion engine. In this operating condition, three of the closures are without effect, because they do not coincide with the high-pressure pump 14 delivering fuel under high pressure. The operating condition represented in Fig. 7 is that in which the fuel output is minimal, in that the angular speed of the valve body 21 during traversal of the second angular range, during which it closes off the spill channel 19, is maximally accelerated if this coincides with the high-pressure pump 14 supplying fuel under high pressure (line PUMP ACTIVE).

It is also possible to provide an intermediate operating condition in which the valve body 21 rotates at an average speed 1.5 times higher than the speed of the crankshaft 4 (that is, three times the lowest average speed of the valve body 21).

In the proposed switch between different average angular speeds of the valve body 21, the fuel delivery decreases tendentially with the increase of the average angular speed of the valve body 21. As a result, extreme speeds and accelerations of the valve body 21 do not often occur. The fact is, a great demand for power (and hence a low average speed of the valve body 21 with respect to the angular speed of the crankshaft) is mostly accompanied by higher speeds of the crankshaft and vice versa. Wear of the valve 20 and the drive 36 is thus prevented.

Wear of the valve 20 and the drive 36 is further prevented in that the angular speed of the valve body 21, at an average angular speed at which it also periodically closes off the spill channel 19 without this coinciding with delivery of fuel by the high-pressure pump 14, is varied only a single time per working cycle of the combustion engine, i.e., only in relation to simultaneous delivery of fuel by the high-pressure pump 14.

Preferably, the lowest average angular speed of the valve body 21 over at least one working cycle of the combustion engine is equal to the simultaneously occurring angular speed of the crankshaft of the combustion engine. At this average speed, the valve body 21 closes the inlet port twice per working cycle of the combustion engine. This angular speed is so high that the portion of the rotatable valve body 21 that closes off the inlet port 23 can cover a relatively large angle in circumferential direction. As a result, at a given diameter of the valve body, it can be of robust design. Also, at a given robustness of the portion closing off the inlet port 23, the valve can have a relatively small diameter. A further advantage of a minimum average speed of the rotary valve higher than the lowest possible speed is that the closure and release of the inlet port 23 of the valve 20 is carried out faster. As a result, the fuel injection phase exhibits relatively steep flanks, which is advantageous for the effective atomization of the injected fuel at the beginning and the end of the injection phase and for accurately controlling the amount of injected fuel.

It will be clear to those skilled in the art that within the framework of the present invention, many other embodiments and modes are possible. Thus, for instance, the valve body can be designed such that it closes off the spill channel several times per revolution. The rotation of the valve body relative to the crankshaft may also be influenced, for instance, by depressing and releasing different sections of a loop cord or chain.

CLAIMS

1. An apparatus for metering the output of a fuel injection system, with a spill valve structure (20) comprising:

an inlet (23),

an outlet (24),

a spill channel (19) from the inlet (23) to the outlet (24),

a rotatable valve body (21) in said spill channel (19), which valve body (21) is arranged for releasing said spill channel (19) during rotation through a first angular range and for closing off said spill channel (19) during rotation through a second angular range,

a drive (36) for rotating said valve body (21), and

a control structure with at least one sensor (37, 38) for signaling angular displacement of a running combustion engine and with a processing structure (41) operatively coupled with said sensor (37, 38) for controlling said drive (36) in accordance with said signaling, such that said valve body (21) rotates continuously,

characterized in that the control structure and the drive (36) are arranged, at any rate in at least one operating condition, for varying the angular speed of said valve body (21) back and forth during each working cycle of the combustion engine, for determining times of start and end of the traversal of said second angular range and for compensating, during the traversal of said first angular range, variations in angular speed of the valve body (21) for maintaining an average angular speed over at least one working cycle of the combustion engine, which corresponds to a current speed of the combustion engine.

2. An apparatus according to claim 1, wherein said spill valve structure (20) is provided with a single rotatable valve body (21).

3. An apparatus according to claim 1 or 2, wherein said spill valve structure (20) is provided with a single drive (36).

4. An apparatus according to any one of the preceding claims, wherein the control structure and the drive (36) are arranged for switching from a first average angular speed of the valve body (21) over at least one working cycle of the combustion engine, to a second average angular speed of the valve body (21) over at least one working cycle of the combustion engine.

5. An apparatus according to any one of the preceding claims, wherein the lowest average angular speed of the valve body (21) over at least one working cycle of the combustion engine is equal to the simultaneously occurring angular speed of the crankshaft of the combustion engine.

6. An apparatus according to any one of the preceding claims, wherein the control structure and the drive (36) are arranged for driving the valve body (21) with an average angular speed over at least one working cycle of the combustion engine, while the valve body (21) closes off the spill channel (19) at least twice per working cycle, and the angular speed of the valve body (21) is raised and reduced exclusively a single time per working cycle of the combustion engine.

7. A fuel injection system comprising:

a fuel pump,

an injector,

a fuel supply channel for supplying fuel from the pump to the injector,

a spill channel (19) which is connected with the fuel supply channel between the fuel pump and the injector, for draining fuel away from the injector,

a spill valve structure (20) having a rotatable valve body (21) in said spill channel (19), which valve body (21) is arranged for releasing said spill channel (19) during rotation through a first angular range and for closing off said spill channel (19) during rotation through a second angular range, a drive (36) for rotating said rotatable valve body (21), and

a control structure having at least one sensor (37, 38) for signaling angular displacement of a running combustion engine and having a processing structure (41) operatively coupled with said sensor (37, 38), for controlling said drive (36) in accordance with said signaling, such that said valve body (21) rotates continuously,

characterized in that the control structure and the drive (36) are arranged, at any rate in at least one operating condition, for varying the angular speed of said valve body (21) back and forth again during each working cycle of the combustion engine, for determining times of start and end of the traversal of said second angular range and for compensating, during the traversal of said first angular range, variations in angular speed of the valve body (21) for maintaining an average angular speed of the valve body (21) over at least one working cycle of the combustion engine, corresponding with a current speed of the combustion engine.

8. A fuel injection system according to claim 7, wherein the number of said rotatable valve bodies is equal to the number of said injectors.

9. A fuel injection system according to claim 7 or 8, wherein the number of said drives (36) is equal to the number of said injectors.

10. A fuel injection system according to any one of claims 7-9, wherein the control structure and the drive (36) are arranged for switching from a first average angular speed of the valve body (21) over at least one working cycle of the combustion engine, to a second average angular speed of the valve body (21) over at least one working cycle of the combustion engine.

11. A fuel injection system according to any one of claims 7-10, wherein the lowest average angular speed of the valve body (21) over at least one working cycle of the combustion engine is equal to the simultaneously occurring angular speed of the crankshaft of the combustion engine.

12. A fuel injection system according to any one of claims 7-11, wherein the control structure and the drive (36) are arranged for driving the valve body (21) with an average angular speed over at least one working cycle of the combustion engine, while the valve body (21) closes off the spill channel (19) at least twice per working cycle, and the angular speed of the valve body (21) is raised and reduced exclusively a single time per working cycle of the combustion engine.

13. A method for the metered delivery of fuel to a combustion engine, comprising:

per working cycle, during a fuel supply phase, supplying fuel to a fuel supply channel leading to an injector,

continuously rotating a valve body (21) in a spill channel (19) communicating with said fuel supply channel, through a first angular range during which supplied fuel is spilled along the valve body (21) and through a second angular range during which the valve body (21) closes off the spill channel (19) and the supplied fuel is delivered via the injector,

wherein angular displacement of a running combustion engine is detected and the rotating of said valve body (21) takes place with an angular speed of the valve body (21) which corresponds to a current speed of the combustion engine,

characterized by, at any rate in at least one operating condition, varying an instantaneous angular speed of said valve body (21) back and forth again during each working cycle of the combustion engine, for determining times of start and end of closing off said spill channel (19), and

while maintaining an average angular speed of the valve body (21) per working cycle of the combustion engine, which corresponds with a current speed of the combustion engine.

14. A method according to claim 13, wherein the number of said rotatable valve bodies that is rotated is equal to the number of said injectors.

15. A method according to claim 13 or 14, wherein the number of said drives (36) that is controlled is equal to the number of said injectors.

16. A method according to any one of claims 13-15, wherein a switch is made between a first average angular speed of the valve body (21) per working cycle of the combustion engine which corresponds to a current speed of the combustion engine and a second average angular speed of the valve body (21) per working cycle of the combustion engine.

17. A method according to any one of claims 13-16, wherein the lowest average angular speed of the valve body (21) over at least one working cycle of the combustion engine is equal to the simultaneously occurring angular speed of the crankshaft of the combustion engine.

18. A method according to any one of claims 13-17, wherein the valve body (21) rotates with an average angular speed over at least one working cycle of the combustion engine, while the valve body (21) closes off the spill channel (19) at least twice per working cycle, and the angular speed of the valve body (21) is raised and reduced exclusively a single time per working cycle of the combustion engine.

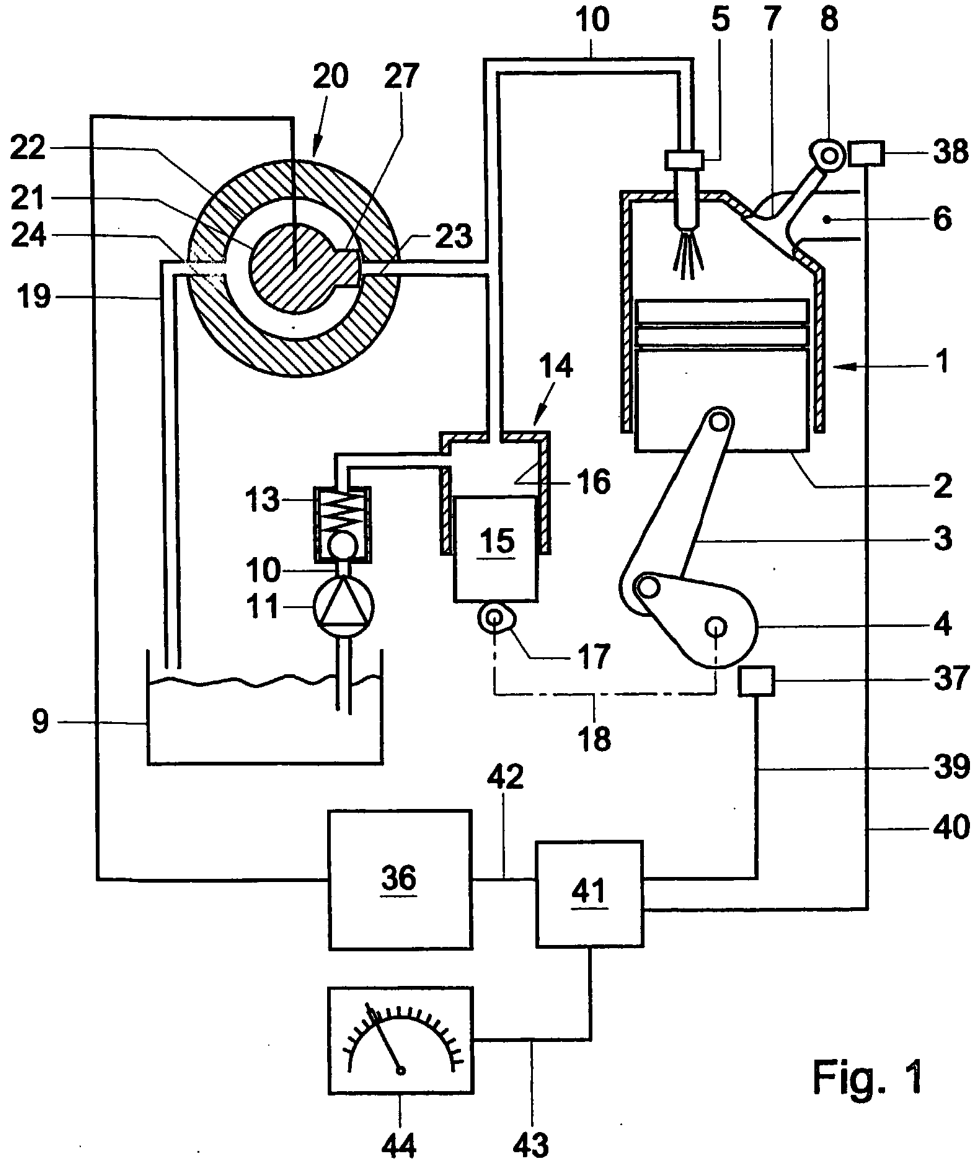
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Fig. 1

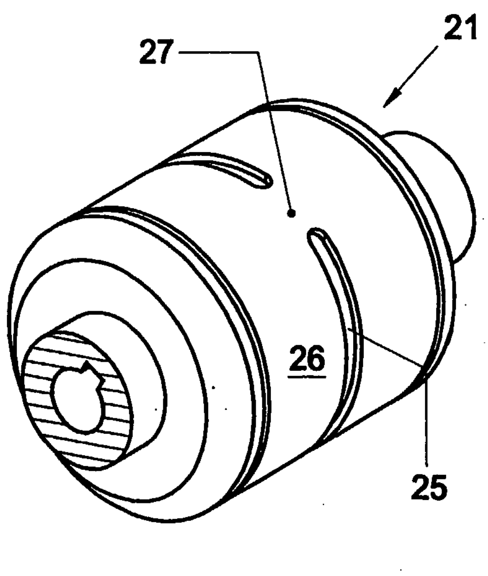
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Fig. 2

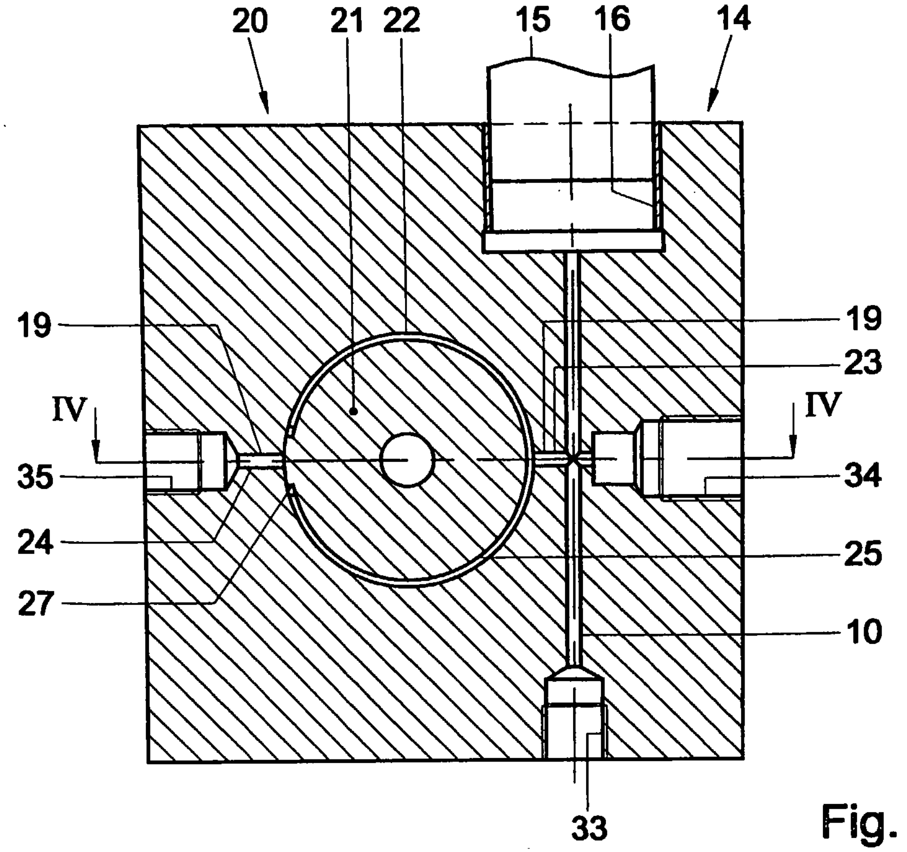
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Fig. 3

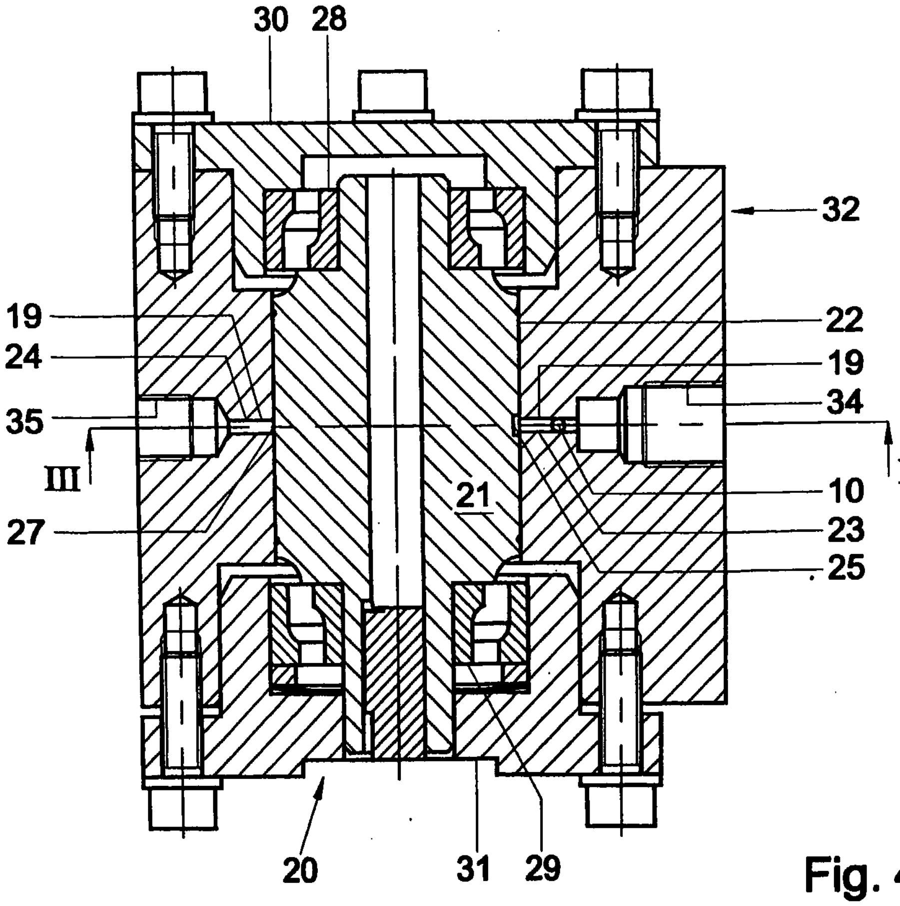
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Fig. 4

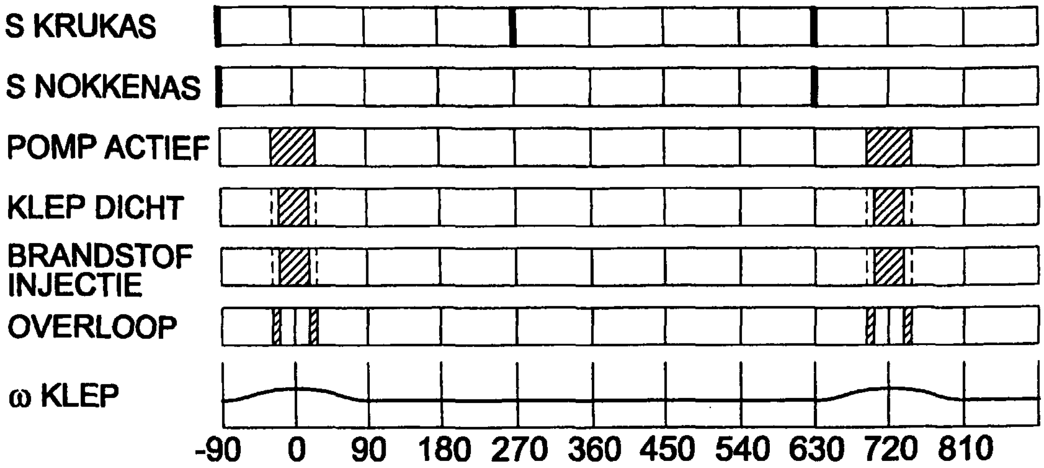

Fig. 5

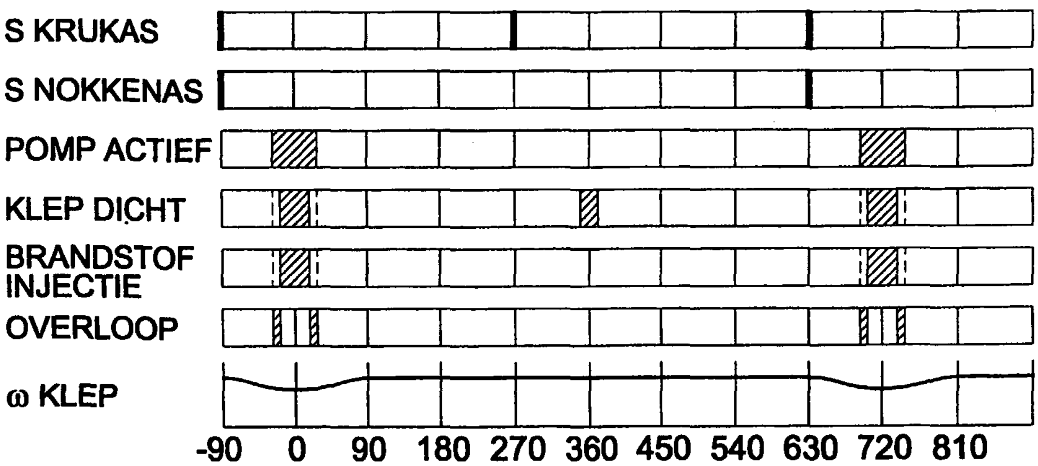

Fig. 6

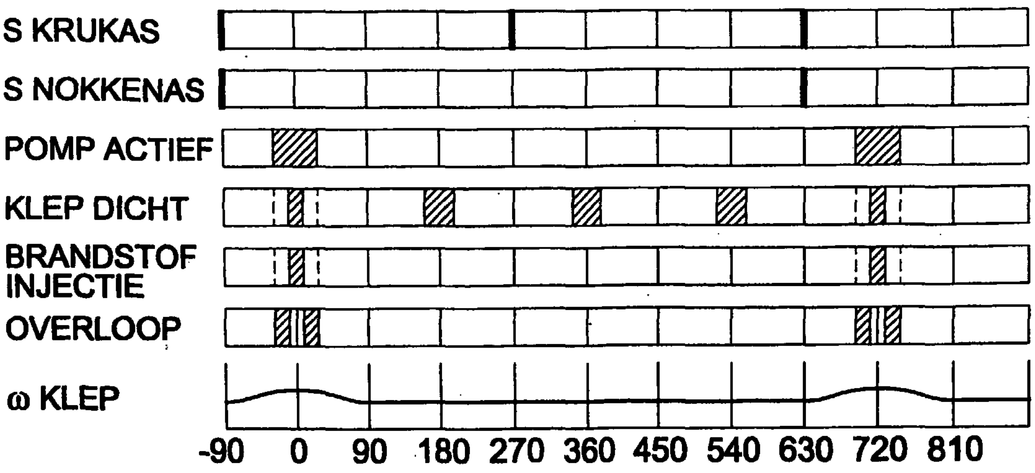

Fig. 7

INTERNATIONAL SEARCH REPORT

International Application No
PCT/NL 01/00167

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F02M59/36

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)
IPC 7 F02M F02D

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 practical, search terms used)

PAJ, EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 4 440 134 A (NAKAO ET AL.) 3 April 1984 (1984-04-03) column 3, line 21 -column 6, line 54; figures	1,7,13
A	WO 82 03888 A (GOLOFF) 11 November 1982 (1982-11-11) abstract; figures	1,7,13
A	EP 0 027 682 A (CATERPILLAR) 29 April 1981 (1981-04-29) cited in the application abstract; figures	1,7,13
A	EP 0 032 168 A (BOSCH) 22 July 1981 (1981-07-22) abstract; figures	1,7,13

-/--

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
20 July 2001	27/07/2001

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31–70) 340–2040, Tx. 31 651 epo nl, Fax: (+31–70) 340–3016	Kooijman, F

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INTERNATIONAL SEARCH REPORT

International Application No
PCT/NL 01/00167

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication,where appropriate, of the relevant passages	Relevant to claim No.
A	US 5 205 251 A (CONKLIN) 27 April 1993 (1993-04-27) abstract; figures -----	1,7,13

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INTERNATIONAL SEARCH REPORT

Information on patent family members

Internati   lication No
PCT/NL 01/00167

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