(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 4 074 945 A1

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
19.10.2022 Bulletin 2022/42

(21) Application number: **21168009.5**

(22) Date of filing: **13.04.2021**

(51) International Patent Classification (IPC):
F01L 1/344 *(2006.01)* ***F01L 13/00*** *(2006.01)*
F01L 13/06 *(2006.01)* ***F01L 9/12*** *(2021.01)*

(52) Cooperative Patent Classification (CPC):
F01L 1/3442; F01L 9/12; F01L 13/0047; F01L 13/06; F01L 13/0005; F01L 2001/0537; F01L 2001/34426; F01L 2001/34433; F01L 2001/34483; F01L 2001/34489; F01L 2810/03; F01L 2810/04

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA ME
Designated Validation States:
KH MA MD TN

(71) Applicant: **C.R.F. Società Consortile per Azioni**
10043 Orbassano (Torino) (IT)

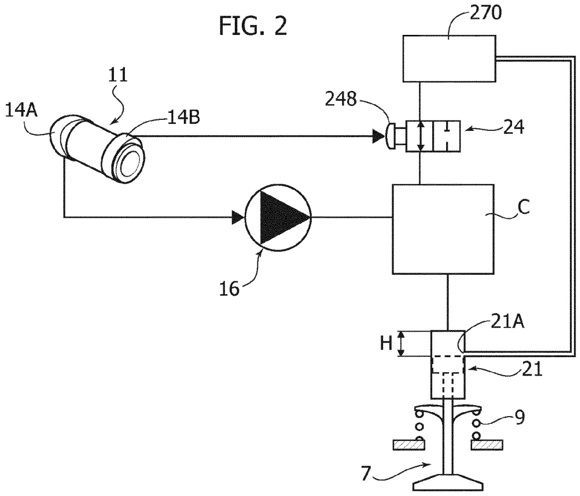
(72) Inventors:
- **RICCO, Raffaele**
  10043 Orbassano (Torino) (IT)
- **STUCCHI, Sergio**
  10043 Orbassano (Torino) (IT)
- **GARGANO, Marcello**
  10043 Orbassano (Torino) (IT)
- **DE MICHELE, Onofrio**
  10043 Orbassano (Torino) (IT)
- **ALTAMURA, Chiara**
  10043 Orbassano (Torino) (IT)

(74) Representative: **Notaro, Giancarlo**
Buzzi, Notaro & Antonielli d'Oulx S.p.A.
Corso Vittorio Emanuele II, 6
10123 Torino (IT)

(54) SYSTEM FOR ACTUATION OF AN INTAKE VALVE OF AN INTERNAL COMBUSTION ENGINE

(57) An actuation system for actuating an intake valve associated with a cylinder of an internal combustion engine comprises a slave piston (21) that operates the intake valve (7) and that is hydraulically controlled by means of a volume of pressurized fluid (C) by a master piston (16) operated by a first cam (14A) of a camshaft (11). A control valve (24) controls the communication between the volume of pressurized fluid (C) and an environment at lower pressure to which a fluid accumulator is connected (270). The control valve (24) is mechanically controlled by a second cam (14B) that is carried by the camshaft (11) and that has the sole function of controlling the control valve (24).

FIG. 2



EP 4 074 945 A1

Description

Field of the invention

[0001] The present invention relates to an actuation system for actuation of an intake valve associated with a cylinder in an internal combustion engine, of the type comprising:

- a master piston controlled, directly or indirectly, by a first cam of a camshaft of the internal combustion engine,
- a slave piston, which actuates said intake valve and which is hydraulically controlled by said master piston, by means of a volume of pressurized fluid interposed between the master piston and the slave piston,
- wherein the intake valve is biassed by at least one spring towards a closed position, and
- a control valve, which controls the communication between the volume of pressurized fluid and an environment at a lower pressure to which a fluid accumulator is connected,
- in such a way that:
    - when the control valve keeps the aforesaid communication closed, the intake valve is sensitive to the movement of said cam, while
    - when the control valve opens said communication, fluid is discharged from the volume of pressurized fluid into the aforesaid environment at lower pressure, so that the intake valve is biassed towards the closed position by said spring and remains insensitive to the movement of said cam.

Prior art

[0002] Since long, The Applicant has developed internal combustion engines provided with a variable actuation system of the engine intake valves, marketed under the trademark "MultiAir", having the above indicated characteristics. The Applicant is also the owner of various patents and patent applications relating to engines equipped with a system of the type specified above and to components of this system.

[0003] Figure 1 of the attached drawings shows a cross-sectional view of a cylinder head of an internal combustion engine according to the art described in the document EP 0 803 642 B1.

[0004] The cylinder head illustrated in Figure 1 and indicated therein with reference numeral 1 is applied to an in-line four-cylinder engine, although it is understood that the variable actuating system illustrated therein is of general application. The head 1 comprises, for each cylinder, a cavity 2 formed in the bottom surface 3 of the head and defining the combustion chamber. In the cavity 2 there are two intake ducts 4, 5 (duct 5 is illustrated with a dashed line) and two discharge ducts 6 (only one of which is visible in the drawing). The communication of the two intake ducts 4, 5 with the combustion chamber 2 is controlled by two traditional mushroom-type intake valves 7 (only one of which is visible in the figure), each comprising a stem 8 slidably mounted in the body of the head 1.

[0005] Each valve 7 is biassed towards the closed position by springs 9 interposed between an inner surface of the head 1 and a cup end 10 of the valve. The communication of the two discharge ducts 6 with the combustion chamber is controlled by two valves 70 (one of which is visible in the Figure), also of a traditional type, which also have associated return springs biassing each valve towards the closed position.

[0006] The opening of each intake valve 7 is controlled, as will be described in the following, by a camshaft 11 rotatably mounted about an axis 12 within supports of the head 1, and comprising a plurality of cams 14 for actuating the intake valves 7 of the internal combustion engine.

[0007] Each cam 14 that controls an intake valve 7 cooperates with the plate 15 of a tappet 16 slidably mounted along an axis 17 which, in the case of the example illustrated in the aforementioned document, is essentially directed at 90° with respect to the axis of the valve 7. The plate 15 is biassed against the cam 14 by a spring associated therewith. The tappet 16 constitutes a pumping piston, or master piston, slidably mounted within a bushing 18 carried by a body 19 of a preassembled unit 20 incorporating all the electrical and hydraulic devices associated with the actuation of the intake valves, according to what is described in detail below. A separate unit 20 may be provided for each cylinder of the engine.

[0008] The master piston 16 is able to transmit a thrust to the stem 8 of the valve 7, in order to cause the valve to open against the action of the elastic means 9 by means of pressurized fluid (preferably oil from the lubrication circuit of the engine) present in a volume of pressurized fluid C to which the master piston 16 faces, and by means of a slave piston 21 slidably mounted within a cylindrical body constituted by a bushing 22, which is also carried by the body 19 of the preassembled unit 20.

[0009] Still with reference to Figure 1, the volume of pressurized fluid C associated with each intake valve 7 can be put in communication with an environment at a lower pressure, constituted by a discharge channel 23, by means of a solenoid valve 24. The channel 23 is configured to receive oil from the lubrication circuit of the engine, fed by the pump of the lubrication circuit, by means of a duct having one or more air purge siphons and a non-return valve (see, for example, EP-A-1 243 761 and EP-A-1 555 398 of the Applicant).

[0010] The solenoid valve 24 can be of any known type suitable for the function illustrated herein and is controlled by electronic control means 25 according to signals S indicative of operating parameters of the engine and/or of the variable actuating system of the engine valves, such as the accelerator position and engine speed, or the temperature or viscosity of the oil in the variable actuating system of the valves.

[0011] When the solenoid of the solenoid valve 24 is energized, the solenoid valve is closed, so as to maintain the volume of fluid C pressurized and to enable the actuation of each intake valve 7 by the respective cam 14, by means of the master piston 16, the slave piston 21 and the volume of oil contained therein.

[0012] When the solenoid of the solenoid valve 24 is de-energized, the solenoid valve opens so that the volume C enters into communication with the channel 23 and the pressurized fluid present in the volume C flows into that channel. Consequently, decoupling of the cam 14 and the master piston 16 is obtained from the intake valve 7, which then returns quickly to its closed position under the action of the return springs 9.

[0013] By checking the communication between the volume C and the discharge channel 23, it is therefore possible to vary the opening moment and/or the closing moment and the opening stroke of each intake valve 7.

[0014] The discharge channels 23 of the various solenoid valves 24 all lead into the same longitudinal channel 26 communicating with pressure accumulators 270, one of which is visible in Figure 1. Each accumulator is essentially constituted by a cylindrical body within which a piston is mounted, defining a chamber of the accumulator communicating with the low pressure environment defined by the discharge channels 23, 26. A helical spring inside the accumulator recalls the accumulator piston towards a position wherein the volume of fluid reception inside the accumulator is minimal. If the solenoid valve 24 is opened at an instant in which the master piston 16 is in a compression state of the fluid present in volume C, part of the pressurized fluid present in volume C flows to the accumulator 270.

[0015] The master piston 16 with the associated bushing 18, the slave piston 21 with the associated bushing 22, the solenoid valve 24 and the channels 23, 26 are carried or formed in the aforesaid body 19 of the preassembled unit 20 to make assembling of the engine quicker and easier.

[0016] In the illustrated example, the discharge valves 70 associated with each cylinder are conventionally controlled by a respective camshaft 28 through respective tappets 29, although in principle, the application of the variable actuating system to the discharge valves is not excluded. This also applies to the present invention.

[0017] Still with reference to Figure 1, the variable volume chamber defined within the bushing 22 and facing the slave piston 21 (shown in Figure 1 in its minimum volume condition, with the slave piston 21 in its upper stroke-end position) communicates with the volume of pressurized fluid C by means of an opening 30 formed in an end wall of the bushing 22. This opening 30 is engaged by an end nose 31 of the piston 21 in order to implement the hydraulic braking of the movement of the valve 7 during closing, when the valve is next to the closed position, as the oil present in the variable volume chamber is forced to flow into the volume of pressurized fluid C by passing through the clearance existing between the end nose 31 and the wall of the opening 30 engaged therein. In addition to the communication constituted by the opening 30, the chamber of pressurized fluid C and the variable volume chamber of the slave piston 21 communicate with each other by means of internal passages formed in the body of the slave piston 21 and controlled by a non-return valve 32 that only allows the flow of fluid from the pressurized volume C to the variable volume chamber of the slave piston 21. Various alternative embodiments of the hydraulic braking device of the slave piston 21 have been proposed in the past by the Applicant (see, for example, EP-A-1 091 097 and EP-A-1 344 900). The object of the hydraulic braking device is to avoid a strong impact (and consequent noise) of the valve 7 against its seat when the valve 7 returns rapidly to the closed position as a result of an early opening of the solenoid valve 24.

[0018] During normal operation of the known engine illustrated in Figure 1, when the solenoid valve 24 excludes the communication of the volume of pressurized fluid C with the discharge channel 23, the oil present in the volume C transmits the movement of the master piston 16, imparted from the cam 14, to the slave piston 21 that controls the opening of the valve 7. In the reverse closing movement of the engine valve, as already said, during the final step, the nose 31 enters the opening 30 causing the hydraulic braking of the engine valve, so as to avoid collisions of the valve body against its seat, for example, after an opening of the solenoid valve 24 that causes the immediate return of the valve 7 to the closed position.

[0019] In the system described, when the solenoid valve 24 is activated, the engine valve follows the movement of the cam (full lift). An early closure of the engine valve can be obtained by opening the solenoid valve 24, so as to empty the volume of pressurized fluid C and to obtain closing of the valve 7 under the action of the respective return springs 9. Likewise, a delayed opening of the valve 7 can be achieved by delaying the closing of the solenoid valve 24, while the combination of a delayed opening with an early closing of the valve can be obtained by controlling the closing and opening of the solenoid valve during the thrust of the relative cam. According to an alternative strategy, in accordance with the disclosures of EP 1 726 790 A1 by the same Applicant, each intake valve can be controlled in a "multi-lift" mode, that is, according to two or more repeated opening and closing "sub-cycles". In each sub-cycle, the intake valve opens and then closes completely. The electronic control unit is, therefore, able to obtain a change in the instant of opening and/or the instant of closing and/or the lift of the intake valve, as a function of one or more operative parameters of the engine. This allows maximum efficiency of the engine and the lowest fuel consumption, in all operating conditions.

[0020] Still with reference to the characteristics of the known system already implemented by the Applicant, which are also usable within the scope of the present invention, it should be noted that a hydraulic clearance compensation device ("lash adjuster") can be interposed between the slave piston 21 and the stem of the engine valve. This solution is, for example, described in the document EP-A-1 635 045 by the same Applicant.

[0021] Figure 1A shows, by way of example, the different lift profiles of an intake valve that can be made with the Multiair system; the outermost curve, indicated by N, represents the lift profile corresponding to the profile of the cam 14, that is, to the operating mode called "Full Lift", in which the control valve 24 never discharges the chamber C of pressurized fluid, so that the intake valve has a movement corresponding to the entire profile of the cam.

[0022] Figure 1B shows the lift profile N, corresponding to the "Full Lift" case, and the EIVC *("Early Intake Valve Closing")* lift profile corresponding to the case of early closing of the intake valve.

Technical problem and object of the invention

[0023] The growing diffusion of hybrid motor-vehicles, of the type in which the internal combustion engine is essentially dedicated to recharging the power supply batteries of the electric traction motor of the vehicle, makes it increasingly convenient to produce relatively simple internal combustion engines, with operation limited to a restricted portion of the engine rpm/torque diagram, located around the point of maximum efficiency of the engine. For engines of this type there is no need for a variable actuation of the intake valves such as that achievable with the "Multiair" system. The same need for simplification may also exist in non-electric vehicles, where one wishes to favor simple and low-cost solutions.

[0024] Furthermore, the implementation - by means of a Multiair device - of EIVC-type strategies, such as the one shown in Figure 1B, involves a loss of energy due to that the compression energy of the return spring of the intake valves is not released. The energy stored by the spring is proportional to the displacement of the valve itself up to the position it reached immediately before closing: this loss, in some operating conditions, may be significant and may nullify the reduction of the suction work in the cylinder enabled by the EIVC implementation.

[0025] Figure 1C schematically illustrates the above. In the diagram on the left, which corresponds to the actuation N of Figure 1B (the law of motion is only reported in a qualitative way), a mass M compresses a spring S following the engagement of the mass M against a cam P which, in the diagram, moves to the left (the spring S corresponds to the spring 9 and the mass M corresponds to the mass of the valve 7 in Figure 1). The compression work is indicated in the lower part with a negative sign (-); during the final step of engagement with the cam, the spring returns the compression work (+) so that - in this case - there is no loss. Otherwise, in the diagram on the right, indicated with EIVC and corresponding to the EIVC actuation of Fig.1B, due to the decoupling from the cam N, the valve closes without being able to perform the compression work, therefore there is a loss equal to the area (-) of the Figure. This loss is proportional to the lift of the spring at the time of disconnection with the cam 14 (i.e. at the time when the solenoid valve is activated and discharges the oil interposed between the master piston 16 and the slave piston 21).

[0026] Another source of loss, during an EIVC actuation, consists in the volume of oil pumped by the master piston, but made to flow back towards the discharge environment without being put under pressure: the amount of this volume is given by the difference of the integral subtended by the full lift curve and the integral of the area subtended by the EIVC curve. Although this oil is not put under pressure, it represents a loss as the master piston has to push it through a series of gaps that introduce pressure drops, said loss being proportional to the volume sent to the discharge.

[0027] At the same time, however, there remains the need to create highly efficient internal combustion engines, to reduce harmful emissions and also drastically reduce fuel consumption. For the purposes of an optimal result from this point of view, the ideal objective would be to have a relatively rapid opening movement of the intake valve, maintenance of the open condition of the intake valve for a certain angular range of rotation of the crankshaft, and then a very rapid closing movement of the intake valve, limiting the movement of the intake valve as much as possible so as to reduce the compression work of the spring associated therewith. Conventional systems with cams for the mechanical actuation of the intake valves are not able to meet this requirement: in particular, the kinematic design of the cam envisages that, the faster the opening or closing step is, the larger the radius of the base circle of the cam must be, directly limiting, in fact, the opening and closing speed of the valves actuated by the cam.

[0028] There is, therefore, on the one hand the need to avoid the complexity of variable actuation systems such as the "Multiair" system, but - on the other hand - there is also the need to carry out a control of the intake valves that allows obtaining a more efficient combustion in the engine cylinders. A primary object of the present invention is to respond to this double requirement.

[0029] Therefore, the main object of the invention is to produce a simplified version of the "Multiair" system described above, which - on the one hand - drastically reduces the constructive complexity and cost of the system, but which - on the other hand - also provides for the intake valves to be controlled in a way which would not be possible with a conventional system, with cams for mechanical actuation of the intake valves, and which is optimal for maximizing engine efficiency.

[0030] In particular, the invention aims to produce an extremely simple and low cost actuation system for the intake valves, but capable of producing an opening profile of each intake valve with a relatively rapid opening movement, an intermediate step wherein the intake valve remains in the same open position for a certain angular range of the crankshaft rotation, and an extremely rapid closing movement of the intake valve.

Summary of the invention

[0031] In order to achieve one or more of the aforesaid objects, the invention relates to an actuation system for actuation of an intake valve of a cylinder of an internal combustion engine, comprising:

- a master piston controlled, directly or indirectly, by a first cam of a camshaft of the internal combustion engine,
- a slave piston, which actuates said intake valve and which is hydraulically controlled by said master piston, by means of a volume of pressurized fluid interposed between the master piston and the slave piston,
- wherein the intake valve is biassed by at least one spring towards a closed position,
- a control valve, which controls the communication between the volume of pressurized fluid and an environment at lower pressure to which a fluid accumulator is connected,
- in such a way that:
    - when the control valve keeps said communication closed, the intake valve is sensitive to the movement of said first cam, while
    - when the control valve opens said communication, fluid is discharged from the volume of pressurized fluid into the aforesaid environment at lower pressure, so that the intake valve is rapidly biassed towards the closed position by said spring and remains insensitive to the movement of said first cam.

said system being characterized in that the control valve is mechanically controlled by a second cam, which is carried by said camshaft and which has the sole function of controlling the control valve, and in that the second cam is configured and angularly positioned on the camshaft in such a way as to cause the opening of the control valve, and the resulting sudden closing of the intake valve, in advance of the closing of the intake valve which would be determined by said first cam.

[0032] As can be seen, in the system according to the invention the control valve that controls the communication between the volume of pressurized fluid and the environment at lower pressure is not a solenoid valve as in the case of the known "Multiair" system, but is a valve controlled mechanically by a dedicated cam, arranged on the camshaft. Consequently, the system according to the invention does not require any electronic control of the communication between the volume of pressurized fluid and the environment at lower pressure, so that it is not able to perform a variable actuation of the intake valves, according to operating conditions of the engine, and in particular when engine speed and engine load vary. At the same time, however, the system according to the invention is capable of producing a control of the intake valves that would not be possible with a conventional mechanical cam actuation system, and that provides, with a good approximation, as will result in more detail in the following, a profile of the intake valve lift significantly close to the ideal profile for obtaining maximum engine efficiency.

[0033] Substantially, the system according to the invention remains a hydraulic actuation system as in the case of the "Multiair" system, but replaces the electronic control provided in the "Multiair" system with a mechanical actuation of the control valve controlling the communication between the volume of fluid at high pressure and the environment at lower pressure. This is done through the provision of a dedicated cam on the same camshaft that carries the cams for actuating the intake valves.

[0034] It should be noted that, although it is not possible, with the present invention, to vary the valve lift profile as the operating conditions of the engine vary, as occurs with the Multiair system, it is anyhow possible to adjust the charge (degree of filling) of the cylinders of an engine equipped with the device of the invention, by modulating the pressure in the intake manifold with the aid of a conventional throttle valve located upstream of the intake manifold. In the same way, the device according to the invention is - in any case - compatible with other intake valve control devices, such as, for example, with timing adjusting devices - of any known type - suitable for carrying out an adjustment rotation of the entire camshaft, so as to be able to adjust the engine load.

[0035] In a preferred embodiment, the control valve has a valve member that is normally in a closed position of the control valve and that is configured to be pushed, directly or indirectly, by the aforesaid second cam towards an open position of the control valve.

[0036] The control valve can be of any known type, for example of the "poppet valve" or "slide valve" type, with a movable member, which can be moved between two operating positions in which it closes and opens the communication between an inlet and an outlet of the control valve.

[0037] In the aforesaid preferred embodiment, the first cam is configured for opening the intake valve substantially in proximity to the Top Dead Center (TDC) of the piston movable in the cylinder with which the intake valve is associated, and for closing the intake valve in advance of the Bottom Dead Center (BDC).

[0038] Also preferably, a limiting device is associated with the slave piston of the intake valve actuation system, which puts a hydraulic chamber, containing a volume of fluid that operates the slave piston, to discharge when the slave piston has moved through a predetermined distance, in such a way that the intake valve has a maximum lift that is reduced with respect to the theoretical maximum lift determined by said first cam, this maximum lift being kept constant through a determined angular range of the crankshaft rotation, up to the crank angle value at which the intake valve suddenly closes, due to the opening of the control valve.

[0039] In the case of a preferred example, the mechanical connection between the second cam and the movable member of the control valve is configured in such a way that the movable member of the control valve begins to move towards its open position after an initial lifting stage of said second cam.

[0040] In an example wherein two intake valves are associated with each cylinder of the engine, each intake valve is associated with a respective actuation system, with a respective first cam and a respective control valve: in this way it is possible to differentiate the actuation laws of the intake valves. The asymmetrical actuation of the intake valves, in particular, at low engine speeds, provides the establishment of swirl motions in the combustion chamber, which favor the mixing of the air charge with the fuel injected through a direct injection, and thus provide a reduction of particulates and HC.

[0041] In this case, preferably each of the two intake valves of the same cylinder is associated with a respective second cam for the mechanical control of the respective control valve, even if the possibility of providing a single second cam for actuation of the two control valves associated with the two intake valves of the same cylinder is not excluded.

[0042] Regarding the design of the cam that actuates the master piston, it is important to limit the maximum excursion of said master piston by appropriately dimensioning the diameter of the master piston itself: the ratio between the diameter of the master piston and the diameter of the slave piston (which is associated with the hydraulic brake of the corresponding intake valve) may be optimized to reduce said excursion by maximizing the opening speed of the intake valve.

[0043] In the event that two intake valves are associated with each cylinder of the engine, a single first cam and a single second cam can be provided to simultaneously control the two intake valves associated with the same cylinder. For example, it can be envisaged that a single master piston controls the two slave pistons of the intake valves by means of a single volume of pressurized fluid that can be discharged through a single control valve.

[0044] In the case of two intake valves per cylinder, the diameters of the slave pistons associated with the intake valves may have different values, so as to differentiate the lift profiles of the two intake valves. Similarly, different limiting devices associated with the two slave pistons may be provided, so as to differentiate the maximum lift of the two intake valves. Or again, the return springs towards the closed position of the two intake valves of each cylinder may have different characteristics, so as to differentiate the closing speed of the two intake valves.

[0045] In an example, which refers to the case in which two respective first cams are associated with the two intake valves of each cylinder, the cams are configured and arranged in such a way that one of the two intake valves of each cylinder opens after that the other intake valve has already been opened and closed.

<u>Description of an embodiment of the invention</u>

[0046] Further characteristics and advantages of the invention will become apparent from the description that follows with reference to the attached drawings, provided purely by way of non-limiting example, wherein:

- Figure 1 is a cross-sectional view of a cylinder head of an internal combustion engine provided with a variable actuating system of the intake valve, according to the prior art,
- Figures 1A, 1B, 1C are diagrams illustrating the operating modes of the known system,
- Figure 2 is a diagram of the actuation system according to the invention,
- Figures 3, 4 show an embodiment example of the control valve used in the system according to the invention, in two different operating conditions,
- Figure 5 is a diagram showing the profiles of the first cam and of the second cam forming part of the system according to the invention and the lift diagram of the intake valve obtainable with the system according to the invention, compared to the lift diagram of the intake valve in a conventional mechanical system, and
- Figure 6 is a diagram showing the advantages of the present invention.

[0047] In Figure 2, the parts corresponding to those of Figures 1 are indicated by the same reference numbers. Therefore, Figure 2 shows an intake valve 7 that is operated by a slave piston 21, which receives pressurized fluid from a pressurized fluid chamber C following the pumping action of a master piston 16, which is mechanically driven, directly or indirectly, by a cam 14A of a camshaft 11.

[0048] A first fundamental difference with respect to the "Multiair" system described above with reference to Figure 1 lies in that the control valve 24, which controls the communication between the pressurized fluid chamber C and the reduced pressure environment, communicating with the pressure accumulator 270, is not an electrically-operated valve. In the case of the invention, the valve 24 has a movable member 24A which is mechanically controlled, directly or indirectly, by a dedicated cam 14B arranged on the cam shaft 11 carrying the cams 11 A for actuating the intake valves.

[0049] Figures 3, 4 show the closed condition and the open condition of an examplary embodiment of the mechanical valve 24. In this example, the valve 24 is a "poppet valve", with a valve body 240 having an axial cavity 241 communicating with two openings 242, 243 connected, respectively, to the pressurized fluid chamber C and to the reduced pressure environment 270. Inside the cavity 241 there is a dividing wall 242 with an opening acting as a valve seat for a ball 245 pushed by a spring 246 towards a closed position of the valve seat. In the cavity 241, there is slidably mounted a movable member 247 having a pin 248 configured to push the ball 245 towards an opening position of the valve seat, against the action of the spring 246 (Figure 4) as a result of a downward movement (with reference to the figures) of the movable member 247. The movable member 247 has an end out of the valve body, indicated by 248, which is actuated, directly or indirectly (by means of a mechanical transmission of any known type) by the second cam 14B.

[0050] Naturally, the embodiment of Figures 3, 4 for the control valve 24 is given here purely by way of example. The configuration of the valve 24 could also be completely different, for example, it could be that of a slide valve or that of any other valve of a known type capable of opening or closing the communication between two ways by means of a movable member having two operating positions. The second cam 14B acts in the direction of moving the movable member of the valve towards one of its two operating positions (towards the open position in the case of the preferred example described herein).

[0051] Figure 5 refers to a preferred examplary embodiment of the invention. In this figure, the line N shows the lift diagram of an intake valve in a conventional mechanical actuation system of the intake valves. The diagram N shows the displacement of the valve as the crank angle varies. The values of 360° and 540° in the diagram of Figure 5 correspond to the TDC and BDC positions of the piston in the cylinder with which the intake valve is associated. The diagrams A and B show the profiles of the first cam 14A and of the second cam 14B, represented as a variation of the radial dimension of the cam as a function of the engine crank angle.

[0052] As can be seen in Figure 5, the first cam 14A causes an opening of the intake valve associated therewith substantially at the TDC and would cause the intake valve to close with a certain advance with respect to the BDC, in the example at about 490°. However, when the crank angle reaches a value close to 440°, the second cam 14B begins to intervene, causing a displacement of the movable member of the control valve 24 into its open position, which causes a rapid closing of the intake valve, in advance of the closing that would be dictated by the profile of the first cam 14A.

[0053] Again in Figure 5, the diagram X indicates the lift profile of the intake valve, which can be obtained with the system according to the invention. As can be seen, from a point P at which the intake valve begins to open, up to a point Q, the intake valve opens due to the hydraulic actuation by the master piston 16 caused by the first cam 14A.

[0054] The slope of the initial section of curve X can be steeper than the slope of the initial section of the curve (corresponding to the cam 14A), by suitably dimensioning the ratio of the diameters of the master piston 16 and of the slave piston 21 (with which the hydraulic brake of the intake valve is associated).

[0055] With reference again to Figure 2, the slave cylinder in which the slave piston 21 is movable is associated with a discharge port 21A, communicating with the low pressure environment 270, which puts the volume of fluid that pushes the slave piston 21 to discharge when the slave piston 21 has completed a stroke H, corresponding to an identical opening stroke of the intake valve 7. Therefore, at point Q, before that the intake valve 7 reaches the maximum lift that would derive from the profile A of the first cam 14A, the intake valve 7 stops and remains in this opened position through a determined angular range of rotation of the crankshaft (in the example from about 400° to about 450°) up to a point R. In the meantime, as can be seen in Figure 5, the effect of the second cam 14B is felt, since the second cam 14B causes a displacement of the movable member 24A of the control valve 24 towards the open position establishing the communication between the pressurized fluid chamber C and the environment at reduced pressure 270.

[0056] In the preferred embodiment, the mechanical transmission between the second cam 14B and the movable member of the control valve 24 is configured in such a way that the movable member 24A begins to move towards its open position only after the second cam 14B has reached its operating point Z in figure 5, corresponding to a lift which in the example is about 0.2 cm. Starting from this moment, the pressurized fluid discharges into the environment at reduced pressure and the intake valve 7 closes rapidly: this step corresponds to the section RS in the diagram X, which is almost vertical.

[0057] As is evident from the above description, the system according to the invention is therefore capable of producing a substantially squared lift diagram X, with a relatively steep opening section, an intermediate section with constant opening, and an extremely steep closing section, which is optimal for the purpose of improving the efficiency of the engine and in particular for the purpose of increasing the combustion speed in the cylinder.

[0058] Figure 6 shows the advantages of the invention with respect to the case of an EIVC actuation with the Multiair system: the maximum opening of the valve is reduced by Δx and consequently the compression work of the spring 9 is lower and also lower is the compression work which is lost when the motion of the valve is disconnected from the motion of the master piston 16, by putting the volume C to discharge. Furthermore, the volume of the total oil pumped by the master piston 16 (area subtended by curve A) is significantly reduced compared to that pumped in the Multiair case (area subtended by curve N) and, therefore, the pressure drop associated with the loss of said volumes through the different channels of the hydraulic circuit is significantly reduced.

[0059] It is therefore evident that - on the one hand - the system achieves a drastic simplification and a reduction in losses in the case of EIVC actuations, compared to the "Multiair" system according to the prior art, the system being intended for applications wherein a variable actuation of the intake valves is not necessary, but at the same time the system also gives rise to a control of the intake valves that would not be possible with a conventional mechanical actuation with the use of cams.

[0060] If each cylinder of the enginehas two intake valves, an actuation system of the type described above can be associated with each of said intake valves, with a respective first cam, a respective second cam, a respective master piston and a respective slave piston, as well as a respective control valve: this configuration allows asymmetrical actuation of the intake valves and the possibility of generating air motions particularly favorable to improving the mixing of air and fuel and to increasing turbulent kinetic energy.

[0061] To simplify the system and reduce the number of components, it is possible to associate the same cam 14A and the same control volume C with both intake valves: with this configuration it would in any case be possible to implement asymmetrical openings of the intake valves, by differentiating the diameters of the slave pistons and the springs of the two intake valves, or by differentiating the position of the discharge port 21A that determines the point Q in the diagram of Figure 5.

[0062] Preferably, a respective second cam is also associated with each intake valve for the mechanical actuation of the respective control valve, but the possibility of providing a single second cam 14B to control the two control valves 24 associated with the two intake valves of the same cylinder is not excluded.

[0063] In this way, it is also possible to provide two opening cycles for the two intake valves which are different versus the time, for example, to exploit in the best possible way the swirl and tumble movements of the air flows introduced into the cylinder.

[0064] Furthermore, it is possible to equip the engine with a device, of a known type, with a cam 14A having an additional portion of different profile axially adjacent to the main portion and which can be activated under certain operating conditions of the engine, so that the master piston 16 is actuated by this additional portion, so as to move the intake valve according to a curve of the type of curve N of Figure 5: in this case, therefore, the second cam 14B does not intervene and the closure takes place in a traditional manner.

[0065] As already indicated, in the event that two intake valves are associated with each cylinder of the engine, a single first cam and a single second cam may also be provided to simultaneously control the two intake valves associated with the same cylinder. For example, it can be envisaged that a single master piston controls the two slave pistons of the intake valves by means of a single volume of pressurized fluid that can be discharged through a single control valve.

[0066] Again in the case of two intake valves per cylinder, the diameters of the slave pistons associated with the intake valves may have different values, so as to differentiate the lift profiles of the two intake valves. Similarly, different limiting devices associated with the two slave pistons may be provided, so as to differentiate the maximum lift of the two intake valves. Or again, the return springs biassing the two intake valves towards the closed position may have different characteristics, so as to differentiate the closing speed of the two intake valves.

[0067] In an example, which refers to the case in which two respective first cams are associated with the two intake valves of each cylinder, the cams are configured and arranged in such a way that one of the two intake valves of each cylinder opens after that the other intake valve has already been opened and closed.

[0068] Still in a further example, which always refers to the case wherein two intake valves are associated with each cylinder of the engine, only one of said intake valves is operated by a system according to the invention, while the other intake valve is operated by a respective cam, by means of a conventional mechanical actuation device or by means of an electronically controlled hydraulic device of the "Multiair" type.

[0069] Of course, without prejudice to the principle of the invention, the details of construction and the embodiments may vary widely with respect to those described and illustrated purely, without departing from the scope of the present invention, as defined by the attached claims.

Claims

1. An actuation system for actuating an intake valve associated with a cylinder in an internal combustion engine, comprising:

- a master piston (16) controlled, directly or indirectly, by a cam (14) of a camshaft (11) of the internal combustion engine,
- a slave piston (21), which actuates said intake valve (7) and which is hydraulically controlled by said master piston (16), by means of a volume of pressurized fluid (C) interposed between the master piston (16) and the slave piston (21).
- wherein the intake valve (7) is biassed by at least one spring (9) towards a closed position, and
- a control valve (24), which controls the communication between said volume of pressurized fluid (C) and an environment at a lower pressure (23) connected to a fluid accumulator (270),
- in such a way that:
    - when the control valve (24) keeps said communication closed, the intake valve (7)

is sensitive to the movement of said first cam (14A), while

- when the control valve (24) opens said communication, fluid is discharged from the volume of pressurized fluid (C) into said environment at lower pressure (270), so that the intake valve (7) is biassed towards the closed position by said spring (9) and remains insensitive to the movement of said first cam (14A).

- said system being **characterized in that** the control valve (24) is mechanically controlled by a second cam (14B) which is carried by said camshaft (11) and which has the sole function of controlling the control valve (24), and **in that** the second cam (14B) is configured and angularly positioned on said camshaft (11) in such a way as to cause the opening of the control valve (24), and the resulting sudden closing of the intake valve (7), in advance of the closure of the intake valve which would be determined by said first cam (14A).

2. A system according to claim 1, **characterized in that** the control valve (24) has a valve member (24A), which is normally in a closed position of the control valve and which is configured to be pushed, directly or indirectly, by said second cam (14B) towards an opening position of said control valve (24).

3. A system according to claim 1, **characterized in that** the first cam (14A) is configured for opening of the intake valve (7) substantially in proximity to the TDC of the piston movable in the cylinder with which the intake valve (7) is associated and for closing the intake valve (7) earlier than the BDC.

4. A system according to claim 1, **characterized in that** a limiting device (21A) which is associated with the slave piston (21) puts a hydraulic chamber containing a volume of fluid which activates the slave piston (21) to discharge when the slave piston has moved by a predetermined distance (H), in such a way that the intake valve (7) has a maximum lift that is reduced with respect to the theoretical maximum lift determined by said first cam (14A), said maximum lift of the intake valve being kept constant through an angular range of the crankshaft rotation, up to a value of the crank angle at which the intake valve (7) closes abruptly due to the opening of the control valve (24).

5. A system according to claim 1, **characterized in that** the mechanical connection between the second cam (14B) and a movable member (248) of the control valve (24) is configured in such a way that the movable member (248) starts to move towards its open position after an initial lifting stage of the second cam (14B).

6. A system according to claim 1, **characterized in that** two intake valves (7) are associated with each cylinder of the engine and that each intake valve (7) is associated with a respective actuation system with a respective first cam (14A) and a respective control valve (24).

7. A system according to claim 6, **characterized in that** a respective second cam (14B) is provided for each of the two intake valves (7) associated with each cylinder.

8. A system according to claim 6, **characterized in that** a single second cam (14B) is provided that is common for the two intake valves (7) associated with each cylinder.

9. A system according to claim 1, **characterized in that** two intake valves (7) are associated with each cylinder of the engine, and that a single first cam (14A) and a single second cam (14B) are provided to simultaneously control the two intake valves (7) associated with the same cylinder.

10. A system according to claim 4, **characterized in that** two intake valves (7) are associated with each cylinder of the engine, and that the diameters of the slave pistons (21) associated with the intake valves have different values, so as to differentiate the lift profiles of the two intake valves.

11. A system according to claim 1, **characterized in that** two intake valves (7) are associated with each cylinder of the engine, and that different limiting devices (21A) are associated with the slave pistons (21) of the two intake valves, so as to differentiate the maximum lift of the two intake valves.

12. A system according to claim 1, **characterized in that** two intake valves (7) are associated with each cylinder of the engine, and that the springs (9) biassing the two intake valves (7) towards the closed position have different characteristics, so as to differentiate the closing speed of the two intake valves

13. A system according to claim 6, **characterized in that** the two first cams (14A) are configured and arranged in such a way that one of the two intake valves (7) of each cylinder opens after that the other intake valve (7) has already been opened and closed.

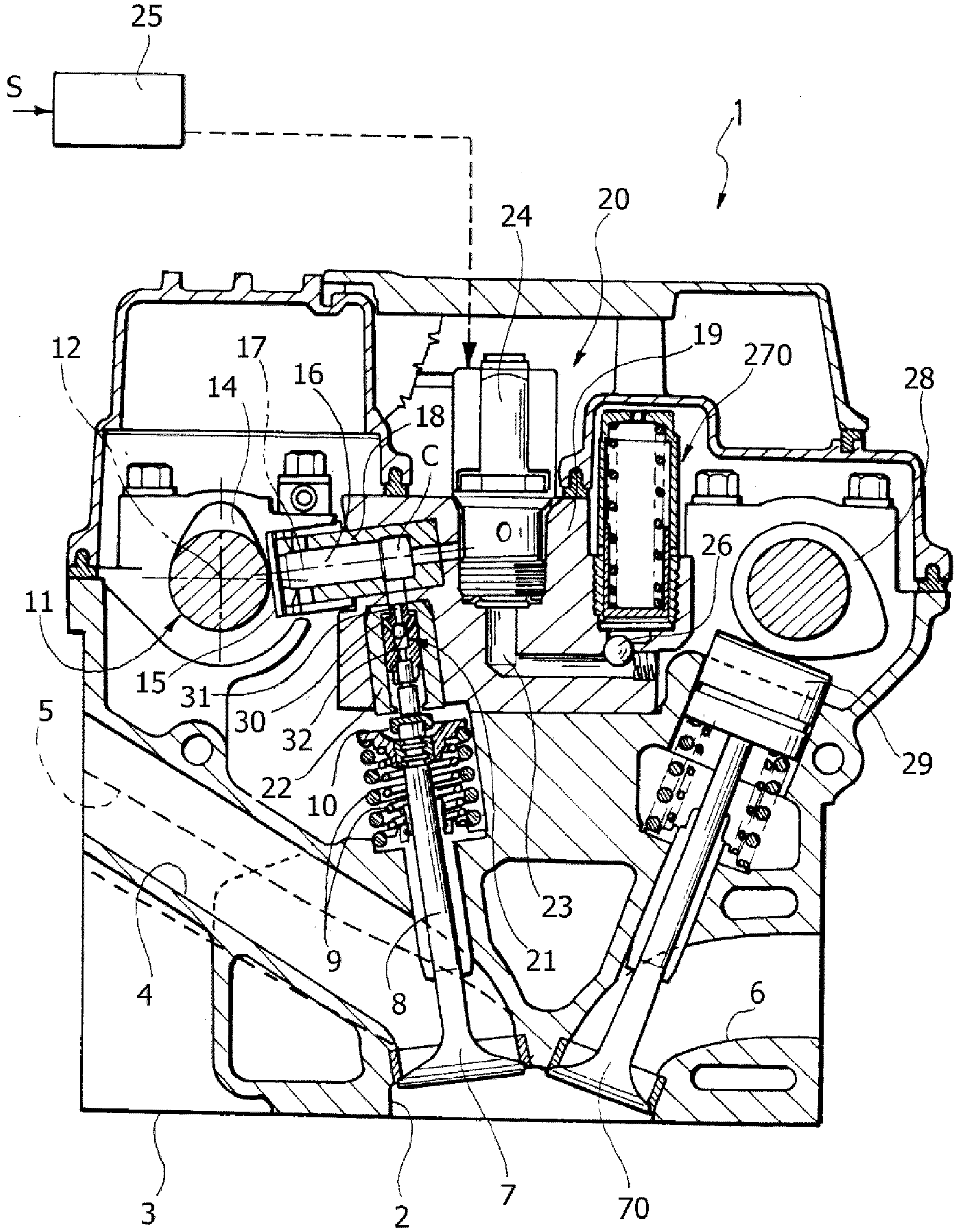
14. A system according to claim 1, **characterized in that** it comprises a timing device to vary the timing of the camshaft (11).

15. A system according to claim 1, **characterized in that**

the first cam is of a type known per se comprising portions with different cam profiles which are axially adjacent to each other and selectable to vary the mode of actuation of the intake valves between a mode of actuation with earlier quick closing and a conventional actuation mode.

16. A system according to claim 1, **characterized in that** two intake valves (7) are associated with each cylinder of the engine, and that only one of said intake valves is operated by said system, while the other intake valve is operated by a respective cam, by means of a conventional mechanical actuation device or by means of an electronically controlled hydraulic device of the "Multiair" type.

FIG. 1



Prior art

FIG. 1A

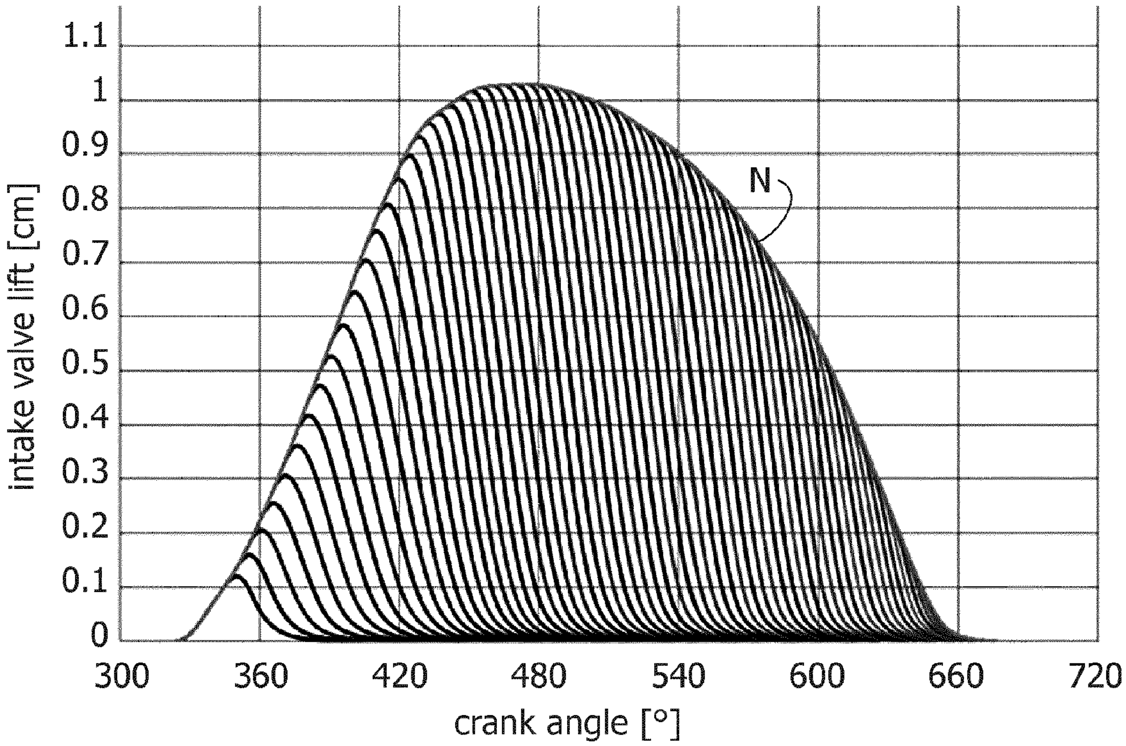

1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
intake valve lift [cm]
300
360
420
480
540
600
660
720
crank angle [°]
N

FIG. 1B

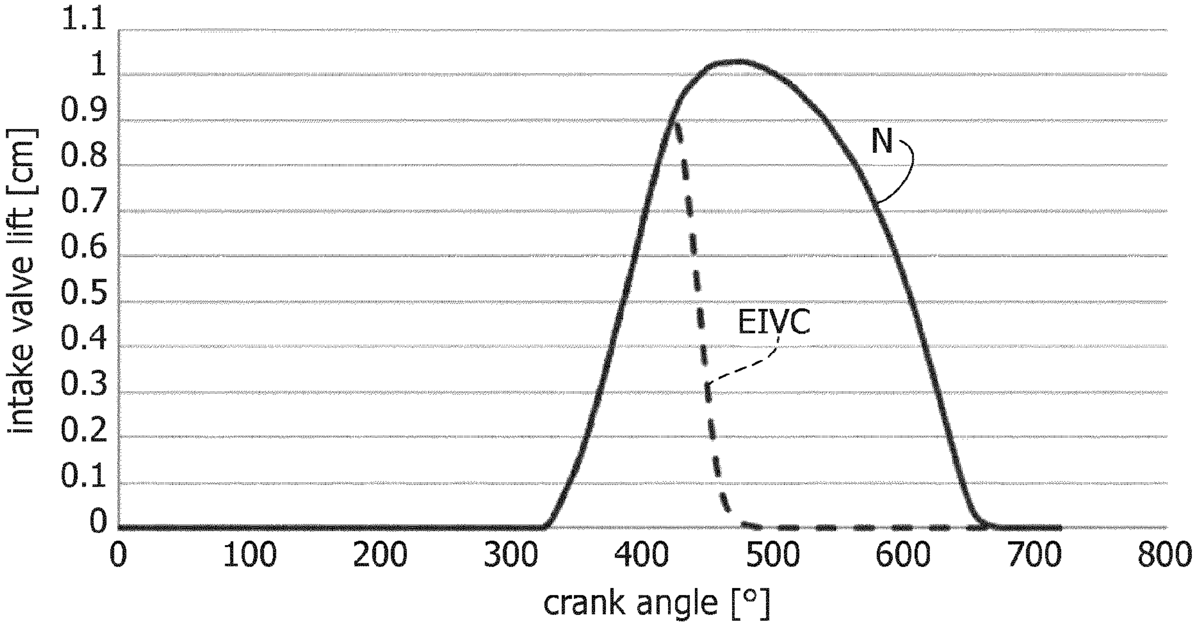

1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
intake valve lift [cm]
0
100
200
300
400
500
600
700
800
crank angle [°]
N
EIVC

FIG. 1C

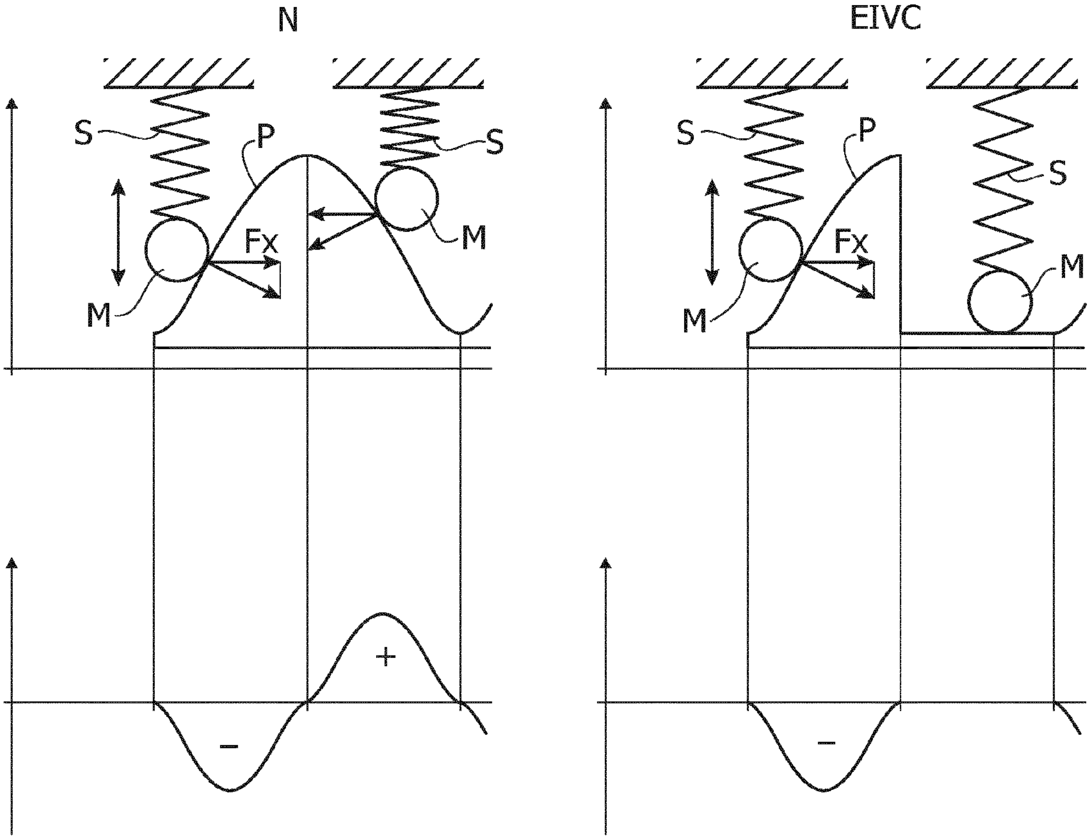


FIG. 2

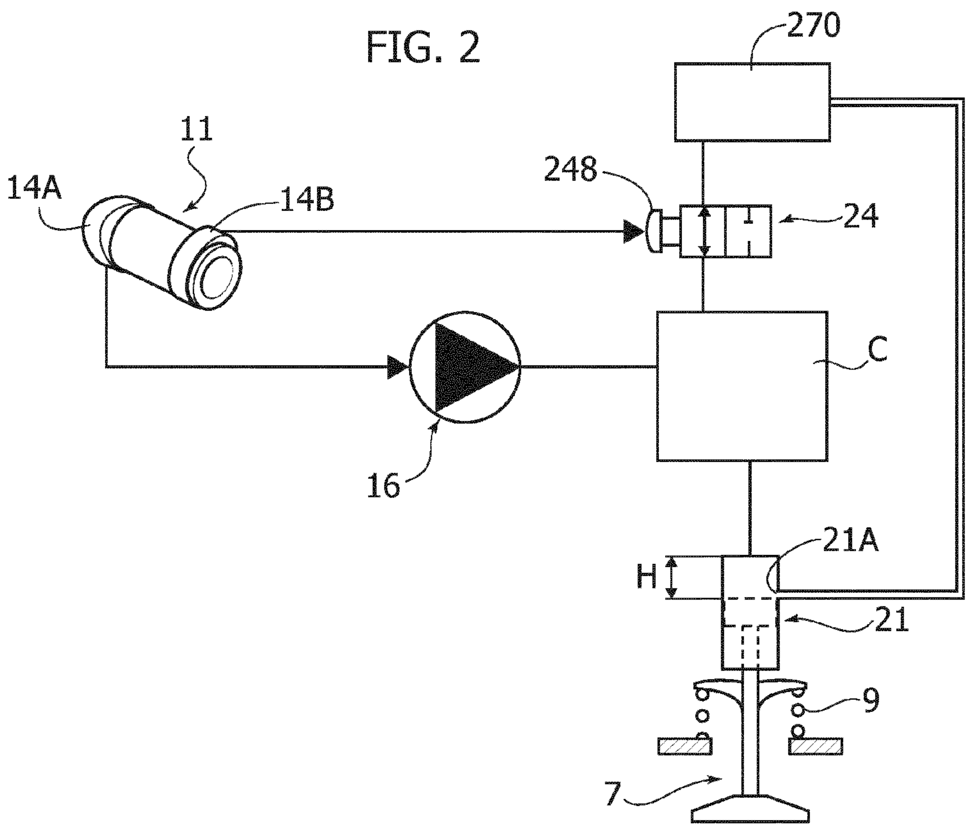


FIG. 3

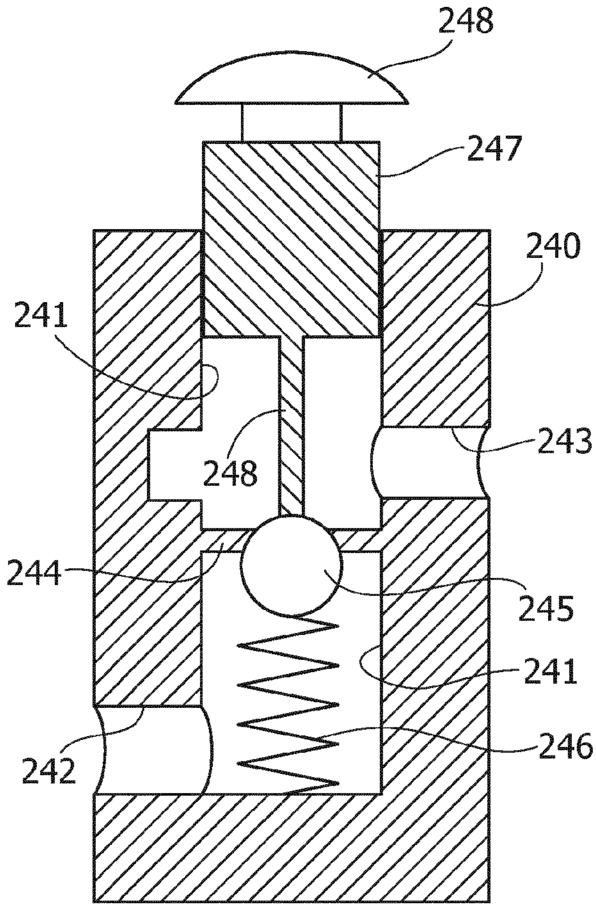

248
247
240
241
248
243
244
245
241
246
242

FIG. 4

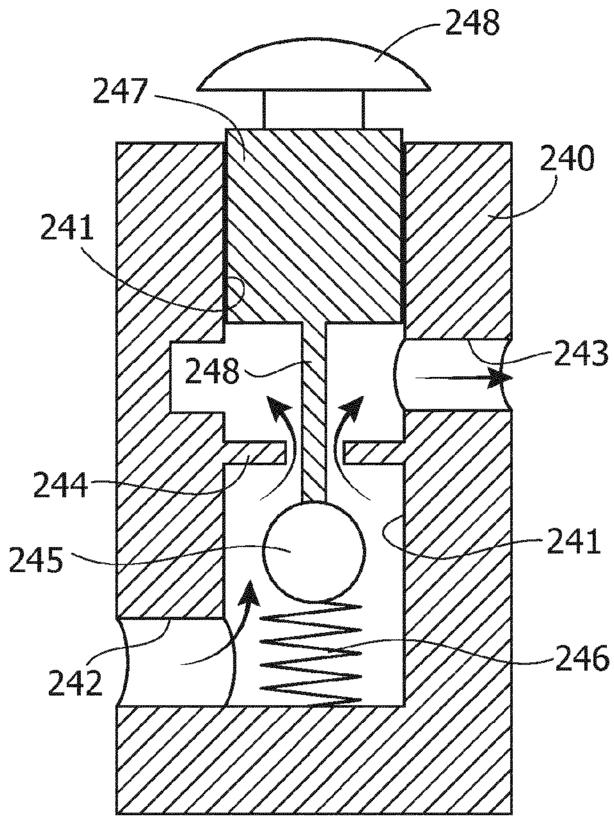
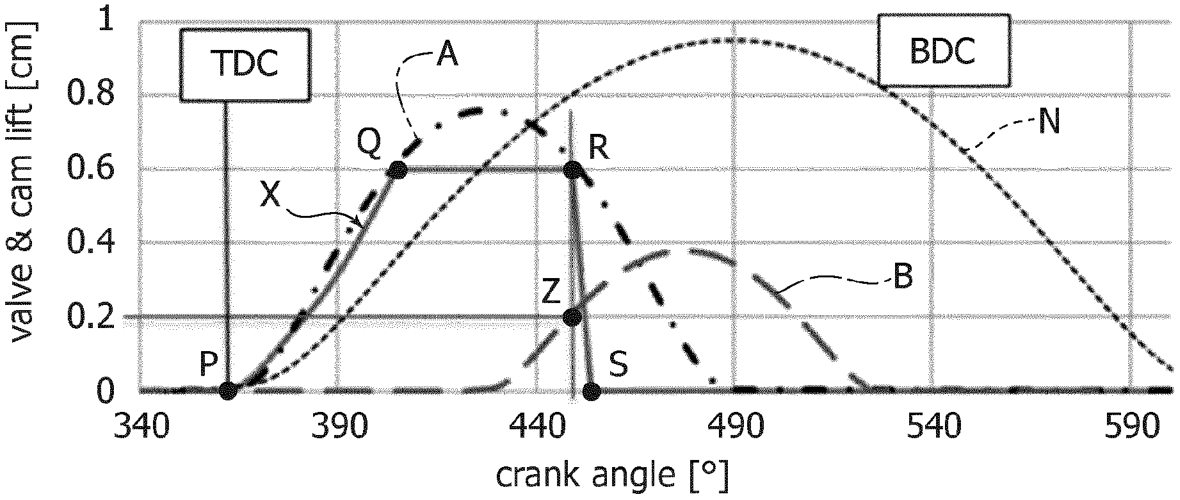
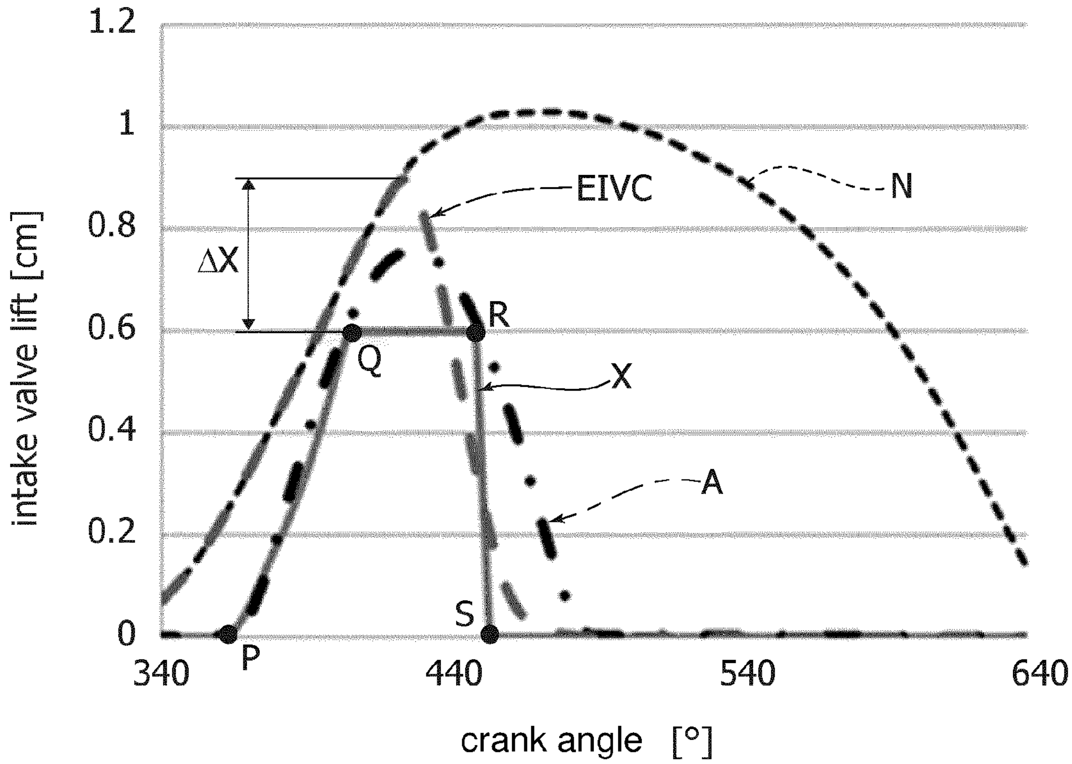

248
247
240
241
248
243
244
241
245
246
242

FIG. 5



1
0.8
0.6
0.4
0.2
0
valve & cam lift [cm]
TDC
BDC
A
Q
R
X
N
Z
B
P
S
340
390
440
490
540
590
crank angle [°]

FIG. 6



1.2
1
0.8
0.6
0.4
0.2
0
intake valve lift [cm]
ΔX
EIVC
N
R
Q
X
A
S
P
340
440
540
640
crank angle [°]

Europäisches Patentamt
European Patent Office
Office européen des brevets

EUROPEAN SEARCH REPORT

Application Number
EP 21 16 8009

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	EP 3 181 842 B1 (FIAT RICERCHE [IT]) 19 June 2019 (2019-06-19) * abstract; figures 1,4 * * paragraph [0003] - paragraph [0020] * -----	1-16	INV. F01L1/344 F01L13/00 F01L13/06 F01L9/12
X	US 2014/318484 A1 (STUCCHI SERGIO [IT] ET AL) 30 October 2014 (2014-10-30) * abstract; figures 1,3-8,16 * * paragraph [0012] - paragraph [0025] * * paragraph [0065] - paragraph [0073] * * paragraph [0205] - paragraph [0209] * -----	1-16	
X	EP 2 597 276 B1 (FIAT RICERCHE [IT]) 16 April 2014 (2014-04-16) * abstract; figures 1,3-6 * * paragraph [0004] - paragraph [0016] * * paragraph [0032] - paragraph [0035] * -----	1-16	
A	DE 10 2012 214645 A1 (MAHLE INT GMBH [DE]) 20 February 2014 (2014-02-20) * abstract; figures 1,2 * * paragraph [0024] - paragraph [0026] * -----	1	TECHNICAL FIELDS SEARCHED (IPC) F01L

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	21 September 2021	Van der Staay, Frank

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 21 16 8009

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

21-09-2021

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
EP 3181842 B1	19-06-2019	EP 3181842 A1 US 2017175592 A1	21-06-2017 22-06-2017
US 2014318484 A1	30-10-2014	EP 2796675 A1 US 2014318484 A1	29-10-2014 30-10-2014
EP 2597276 B1	16-04-2014	EP 2597276 A1 US 2013133596 A1	29-05-2013 30-05-2013
DE 102012214645 A1	20-02-2014	NONE	

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- EP 0803642 B1 **[0003]**
- EP 1243761 A **[0009]**
- EP 1555398 A **[0009]**
- EP 1091097 A **[0017]**
- EP 1344900 A **[0017]**
- EP 1726790 A1 **[0019]**
- EP 1635045 A **[0020]**