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(54) **ELECTRO-HYDRAULIC VARIABLE VALVE ACTUATOR AND METHOD TO CONTROL VALVES OF INTERNAL COMBUSTION ENGINES**

(57) The invention concerns an electro-hydraulic system for the actuation of the valves of the type comprising a cylinder-piston assembly (11-12) for each valve (13) and means for actuating said piston (12) into opening in opposition to spring means (13) recalling the engine valve (13) into its closed position; the actuation chamber (11) is in communication, on one side, with a high-pressure pipe (14), through a first semi-automatic electrovalve (16), which opens automatically when the pressure

in the pipe between said valve and the actuation chamber (11) lies above the nominal pressure in the high-pressure pipe (14) and, on the other side, with a low-pressure tank (15) through a second semi-automatic electrovalve (17), which opens automatically when the pressure in the pipe between said valve and the actuation chamber (11) lies below the nominal pressure in the low-pressure tank (15).

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## Description

**[0001]** It is known that the intake and outlet valves of internal combustion engines control the flows of gases coming into and going out from the combustion chamber of each engine cylinder. From the origins and up until today, these valves are generally controlled by cams, integral with a so-called cam-shaft, which is driven into rotation synchronically with the engine shaft movement. Each cam, upon rotation, pushes its respective engine valve into opening, while a return spring is compressed during the opening phase and extends back during the closing phase, guaranteeing this closure, as well as a constant contact between engine valve and cam. It is thereby possible to accomplish a change of both the time during which the engine valve remains open or closed, respectively (lift law determined by the cam profile), and of the timing (phase law) of the lift law of an engine valve over that of all others (timing determined by the angular position, one with respect to the other, of the cams mounted on the cam shaft).

**[0002]** Opening times and timing are established so as to optimise engine efficiency in a certain operating range and for certain load conditions and, with current cam-shaft distribution systems, they cannot be modified during engine operation. This results in the fact that, by moving away from such optimal conditions, engine efficiency drops.

**[0003]** The cam profile is designed so as to take into account some function requirements of the distribution system. In particular:

- optimisation of the engine operating conditions, especially efficiency, emissions, performance.
- reduction of impacts and, consequently, of the wear of the parts in contact, especially of the cam/tappet, and of the valve/valve seat.

**[0004]** The typical parameters in the definition of a specific law for engine valve control highlighted in fig. 1, which shows a typical law for engine valve control and the corresponding typical values, are the following:

- lift;
- time;
- phase (in the automotive field, the term "phase timing" refers to the opening advances and closing delays of the valves with respect to the dead points of the piston).

**[0005]** Each of the parameters highlighted in figure 1 plays a well-defined role in the determination of the global efficiency of the system. To each point of work, identified by the engine revolution speed and load, corresponds a set of such parameters which is optimal for the engine; it would therefore be theoretically useful to be able to continuously change all these values according to the variation of the engine rpm and load.

**[0006]** In the past, the efficiency of internal combustion engines, thus controlled, has always been considered satisfactory. However, there is currently ever-rising demand for high-performance engines compatible with the strict limits set by anti-pollution laws; more than ever, the adoption of new strategies for controlling and managing the engines is therefore required. As a matter of fact, new types of combustion strategies (for example HCCI, Homogeneous Charge Compression Ignition) are being developed, as well as new types of engine cycles which are optimised from a thermodynamic point of view, which represent an alternative to the classic OTTO cycle, for example the Miller or Atkinson cycle, together with more flexible strategies for the closing or opening of the intake and outlet valves. Conventional distribution systems are very strong, reliable and cost-effective, but display a noticeable lack of flexibility; this may be partly compensated for only by using auxiliary systems such as phase variators, which, however, rather complicate engine architecture. They implement lift laws which are optimised only in well-defined engine operating conditions, but which cannot be adapted to the various speeds and loads. The rigidity of such systems leads to the development of new solutions for the actuation of engine valves, which allow to act independently on all the typical parameters of engine distribution, so as to be able to adapt them to any operating condition.

**[0007]** Various strategies for the variable actuation of engine valves (VVA = acronym of Variable Valve Actuators) are currently under study, and are also partly actuated on prototypes of internal combustion engines, which strategies, under the control of an electronic data processing unit equipped with suitable software, provide to accomplish:

- phase variation;
- time and phase variation;
- lift change;
- combined change of lift and time;
- cyclical exclusion of one or more valves.

**[0008]** The timing variation of intake and outlet valves allows to control the crossing point between the lift curves of said valves (see fig. 1). Changing this crossing allows to continuously dose the quantity of internal EGR (acronym of Exhaust Gas Recirculation). Such strategy is exploited to adjust the temperature of the gases arrived at the cylinders, by releasing into the incoming main flow a set quantity of already burnt gases. Thereby it is possible to adjust also the maximum temperature in the cylinder, in that the already burnt gases do not take part in the combustion. This is extremely important for the building of NO<sub>x</sub>, the presence of which in exhaust gases depends on the maximum temperature reached in the combustion chamber.

**[0009]** The time change of the engine valve opening may be achieved, for example, by keeping the opening angle constant and by acting on the closing angle of the

engine valve (see fig. 2, diagram on the left-hand side of the drawing). This kind of adjustment of the outflow area during intake allows to control, during the compression stroke, the flowback of the intake gases into the intake pipe, allowing to manufacture engines having a variable actual compression ratio which differs from the expansion one. Cycles may thereby be implemented which are optimised from a thermodynamic point of view, such as Miller cycles (cycle in which the intake valves are closed before the intake stroke is completed) and Atkinson cycles (cycle in which the closure of the intake valves is delayed during the compression stroke), with noticeable improvements of the global efficiency of the engine in that it is thereby possible to manufacture engines with a compression ratio which differs from the expansion one.

**[0010]** In general, the best results are obtained by being able to vary independently the opening angle and the closing angle of the valves. It must be highlighted that the variation of the time/phase allows, in addition to the previously-cited benefits concerning consumption and polluting emissions, also to optimise the engine torque curve, whose performance improves.

**[0011]** By being able to act on lift variation, it is possible to implement a number of engine control strategies. Firstly, it is possible to control the engine load without the help of a throttle valve for reducing the fuel flow. Thanks to the removal of such member, load loss reduction in the intake pipes may be achieved with full-load operating conditions; as a matter of fact, in such operation conditions, although the throttle is fully open, there are always the load losses associated with the presence of the throttle valve itself; these losses disappear if load adjustment is carried out by acting on the lift of the intake valves, said valves becoming reducing members. The opportunity of adjusting the maximum lift value allows to obtain various speeds of gas outflow through the valves and to hence control the microturbulence in the engine cylinder, so as to optimise combustion.

**[0012]** In the case of multi-valve engines, i.e. having two or more intake valves for each cylinder, if it was possible to adjust independently the lift of each of the intake valves, control of the air swirls within the cylinder could also be obtained, positively affecting polluting emissions and consumption.

**[0013]** Through such type of adjustment it is possible to reduce the load without using auxiliary members and to optimise the engine torque curve, thereby obtaining an engine with improved elasticity and higher available torque at low rpms.

**[0014]** Finally, by "cyclic valve exclusion" the opportunity of not actuating one of the valves of a cylinder is meant, cycle by cycle. Such control strategy may be used for reducing the engine power at low loads thanks to the exclusion of one or more cylinders. In order to avoid excessive engine irregularities in these operating conditions, it is not deactivated always the same cylinder, but cyclically all cylinders.

**[0015]** Moreover, concerning multi-valve engines,

should the opportunity of excluding one only of the intake valves be provided, the same benefits on swirl control as set forth above can be obtained.

**[0016]** The strategies for varying valve actuation set forth in the preceding paragraphs cannot be accomplished with conventional, cam-shaft distribution systems. In such systems it is therefore not always possible to test new combustion techniques and new engine control patterns, which would require the option of simultaneously, and independently, varying all the fundamental values of an engine valve lift law. The use instead of "fully flexible" systems would lead to engine efficiency increase, allowing the development of units having a higher compression ratio and hence, given the same power, of smaller and lighter engines. A more effective emission control would be possible and finally a more efficient engine management, in that cyclical exclusion strategies of the cylinders may be implemented over hauls in which low power demand is found, obtaining noticeable improvements in terms of consumption.

**[0017]** The ideal system is therefore the one which, as already stated, allows to accomplish cycle by cycle and cylinder by cylinder:

- continuous control of the lift;
- continuous control of the time;
- continuous control of the phase;
- exclusion of an engine valve.

**[0018]** Attempts in this direction have been made, but all the devices suggested up until today have severe limitations. They can be classified into three subgroups:

- mechanical systems;
- electromechanical systems;
- electro-hydraulic systems.

**[0019]** Mechanical systems are by nature strong, operate with reduced power losses, but are substantially very rigid systems. They comprise phase variators and lift variators. The most evident limitations are:

- impossibility to vary the lift start/end angle in the absence of an auxiliary phase variator;
- difficulty in accomplishing Atkinson/Miller cycles;
- impossibility to vary the time of the lift law;
- impossibility to exclude one or more valves per cylinder (impossibility to control swirl and to cyclically deactivate the non-actuatable cylinders);
- impossibility to accomplish multiple openings for each cycle.

**[0020]** The electromechanical systems of the known art, also called electromechanical VVAs, essentially consist of two electromagnets attached to the cylinder head, between which an anchor made of ferromagnetic material is located, integral with the engine valve and supported by two series-arranged equal springs; the anchor run

between the two electromagnets represents the maximum value of the engine valve lift. A system of this type is described for example in US-4,777,915, please refer thereto for further details; in particular, the electromechanical VVA subject of the present patent solves some of the problems of the prior art, such as the one of the variation of the engine valve lift law, where to the problem of the "first-catching" is connected.

**[0021]** The limitations of such systems can comprise:

- difficulties in bringing back the engine valve onto its seat at modest speeds ( $<0,1$  m/s to achieve a so-called "soft landing");
- intrinsic lack of safety from a possible impact of the engine valve against the piston in case of system failure;
- complexity of system control;
- poor strength and reliability.

**[0022]** Electro-hydraulic systems can finally be divided into the following three categories:

A) "lost motion systems" - These systems - see for example the one manufactured by the company FIAT and called UNIAIR - provide that a chamber with pressurised oil is arranged between each of the cam shaft cams and the stem of the engine valve driven thereby, in which chamber a small piston works which, under the action of the pressure force, pushes on the engine valve. Thereby, cam motion is integrally transmitted by a hydraulic means (oil) to the engine valve. In order to accomplish partial lifts/times, oil is drawn from the work chamber, so that cam motion is not fully transmitted to the engine valve (hence the name "lost motion").

These "lost-motion" systems have noticeable limitations in the management of the opening and closing runs; as a matter of fact, the flexibility thereof is limited to lift laws having a well-defined shape and not allowing all the desirable strategies for engine control. For example, it is not possible to obtain valve opening laws having small lifts and long angular times, which are important to introduce a sufficient turbulence in the combustion chamber at low rpms. In any case, the lift laws achievable by this system are contained within the cam law, with the advantage of no interference problems between valves and piston, but with the limitation of not being able to implement the Atkinson cycle.

B) "Camless spring systems" - Such systems provide an actuator consisting of a cylinder-piston assembly; the cylinder chamber determines a work volume in communication with a source of high pressure and with a source of low pressure, respectively, through two driven valves which, depending on requirements, enable either source. The actuator piston opens on the one hand into such volume, on the other hand it acts on the engine valve stem. Thereby,

by enabling high pressure, the control volume becomes pressurised and the actuator piston pushes the engine valve which consequently opens. Closing is accomplished by disabling the high pressure and putting the work chamber in communication with the low work chamber in communication with the low pressure source, so that the engine valve, under the action of a respective opposing spring, can return into its seat.

Various types of systems belong to such category, each having its own peculiarity, generally stemming from the type of electrovalves enabling high and low pressure. A system of this type is the one manufactured for example by FEV-HMVT: however, it is remarkably complex from a construction point of view, requiring important structural alterations to the engine, and nevertheless with not extremely wide application fields of the system. Other systems of this type are those described in US 6,092,495, as well as in US 2004/0094104.

C) "Desmo-hydraulic camless systems" - These systems may have a general layout according to which the engine valve is driven by an actuator consisting of a double-effect piston, which runs within a cylinder, wherein each of the two faces of the piston determines a respective work chamber: one in the lower part and one in the upper part of the cylinder. The devices illustrated in DE 101 43 826, US 2004/0074456, DE 19953789 A1, EP 0915235A2 belong to this category. A multi-way, control electrovalve allows to put in communication the above-said chambers alternately with a high-pressure or with a low-pressure source. The helical spring is no longer provided to bring the engine valve onto its seat, but opening and closing of the engine valve are both actuated by using pressurised fluid. By putting the lower work chamber in communication with the high-pressure source HP, and the upper chamber with the low-pressure source LP, the engine valve closes. By keeping both paths to the two pressure sources closed, the engine valve can be kept closed for the desired time. In order to open it, it is sufficient to put the high-pressure source in communication with the upper work chamber and the low-pressure source with the lower chamber. The main limitation of such type of system is the absence of a member which, in case of failure, brings the engine valve back onto its seat avoiding the impact with the piston.

**[0023]** Currently all these electro-hydraulic systems are still in embryo, also in the light of their remarkable construction and control complexity. As said, their operation is limited in various ways, which are typical of each specific device embodiment but, in addition thereto, a serious limitation of all the types examined is the absence of solutions guaranteeing a high degree of system safety. As a matter of fact, in case of failure of the electric system or of the control device, no solutions are provided which

allow the immediate shutting of the engine valves: impacts can hence occur between piston and engine valve, with evident serious engine damage. Moreover, no reliable solutions are provided for the damping of the strong pressure oscillations in the work chamber of the hydraulic actuators, nor for the recovery of such energy, which is normally dissipated through the resulting production of heat which must be duly dissipated. Reference can be made to patents Sturman US-5,638,781 and Mercedes-Benz US-5,595,148. In addition thereto, the absence of simple and effective control strategies of the engine valve lift must be noted. Finally, it must be highlighted how the majority of current systems has serious limitations of the maximum working rpm, due to the dynamics of the above control valves, which have upper working limits beyond which they cannot act in times comparable to those of the engine cycle. Such problem is deeply felt especially at engine rpm above 6000 rpm/min.

**[0024]** In order to thoroughly understand the main steps of the opening and closing of an engine valve, it is possible to refer to the lift laws of the prior art with cam shafts, outlined in fig. 3, wherein the first diagram represents the lift position of the engine valve, the second one represents the instant speed value of the engine valve, and the third one represents the instant value of the acceleration imparted to the engine valve. The engine valve first opens starting with a certain acceleration from its closed position. Then the speed begins to increase and reaches a maximum value with 0 acceleration. Starting from that point, the engine valve must be braked (portion in which the acceleration is negative), until it reaches the maximum lift value with 0 speed and maximum deceleration. The closing step occurs in the same way. The merit of cam shafts is precisely that of "accompanying", at 0 speed, the engine valve onto its seat, thereby avoiding violent impacts and consequently noise and deterioration of the contact members. In substance first the cam imparts to the engine valve a certain acceleration; the engine valve is then braked by the helical spring, holding it in contact with the cam so as to reach its ends-of-stroke at 0 speed. In the subsequent closing step the spring will accelerate and the cam will decelerate the engine valve, which will be able to close with no impacts.

**[0025]** In the camless-type, electro-hydraulic, VVA systems there is no mechanical restraint (such as the cam) capable of imparting to the engine valve a well-defined motion law and allowing it to open and close with no impacts. In the lift law of a system of this type, 5 main steps can be identified, as outlined in sectors 1 to 5 of fig. 4 (lift stroke on the y-coordinate axis and times on the x-coordinate axis):

Sector 1) - first part of the opening run: portion with accelerating engine valve. The initial starting condition is with the engine valve closed. In such condition the high-pressure electrovalve (HP) is closed and the low-pressure valve (LP) is open; the work chamber of the actuator (regardless of it being of the type

with simple-effect piston and return spring, or of the type with double-effect piston) is sealed off from the high-pressure source and in communication with the low-pressure source. By energising the high-pressure electrovalve and keeping closed the low-pressure one, the high-pressure line communicates with the work chamber of the actuator. The engine valve moves, integrally with the actuator piston and under the action of the pressure force, from its seat at 0 speed and acceleration other than zero. Its speed begins to increase until its kinetic energy reaches a maximum value.

Sector 2) - second part of the opening run: portion with decelerating engine valve. In many systems of the prior art the acceleration step engages the entire opening run and the engine valve hence arrives at its end-of-stroke at top speed; however, it is normally suitable, if not essential, to provide that, starting from the maximum speed condition, a subsequent step begins wherein the engine valve is braked. The engine valve has a certain kinetic energy, which in some respects is desirable to be cancelled before it reaches its maximum lift value at 0 speed. In actual fact, when the engine valve has reached the desired opening value, and both high and low pressure valves are closed, in practice it occurs that, the engine valve not having been braked by any member, it will still have its kinetic energy, which allows it to continue its stroke even if the high-pressure electrovalve is closed. The low-pressure electrovalve also being closed, an abrupt volume increase due to the advance of the engine valve will occur, not accompanied by oil entry in the work chamber of the actuator. Since oil is an incompressible means, these volume increases will consequently be associated with a pressure drop; even if the actuator begins to brake the engine valve, when the pressure reaches the saturation value for the oil, steam bubbles form here. The engine valve then reaches its maximum lift value and reverts its motion, but at this stage, the steam pocket formed will be compressed again. This determines strong pressure swings, which cause the engine valve to swing as illustrated precisely in sector (2) of fig. 4. The swings continue until the entire residual kinetic energy of the engine valve is dissipated as heat.

Sector 3) - holding step: engine valve halted in open position. The engine valve is held open at the desired lift value, for a certain time of the angular rotation of the engine shaft. In such condition the engine valve will still have residual swings from the previous step, although of negligible width.

Sector 4) - first step of the closing run: portion with accelerating engine valve. The first part of the closing run of the engine valve occurs by actuating the low-pressure electrovalve: the work chamber of the actuator is sealed off from the high-pressure line and put in communication with the low-pressure one.

Thereby, under the action of the opposing spring, the engine valve will be accelerated and it will cover, at ever increasing speed, the entire closing run until it impacts with its seat.

Sector 5) - second step of the closing run: portion with decelerating engine valve. In such an electro-hydraulic system, the engine valve - in the absence of a positive braking action - inevitably impacts against its seat with all the kinetic energy accumulated under the push of the opposing spring; the valve can thereby also open again following rebound (as shown precisely in sector (5) of fig. 4), allowing gas flowback into the intake or outlet pipes towards which the engine valve opens. Thereby, the kinetic energy it has acquired is dissipated through the impacts against its seat and through the noise originated by the high speed at which the engine valve impacts with the ends-of-stroke.

**[0026]** The subject of the present invention is an electro-hydraulic device controlling the valves of internal combustion engines, of the general type referred to as "camless system with spring". As already stated, this type comprises the devices described in

US 6 092 495 - in this patent it is provided to drive the engine valve by means of a hydraulic actuator 52, which is fed by pressurised oil line 62 in opposition to return spring 56. However, the control is actuated simply by the opening and closing actions and is hence subject to all the drawbacks described above. In actual fact, US 6 092 495 concerns engine management methodologies and not a proper VVA in particular;

US 2004/0094104 - in this patent an open-and-close valve 7 is equally provided, while the exact control of the movements of engine valve 3 is demanded to a variable-flow pump 6, arranged downstream of the engine valve in the sense that it simply drives the release of the oil pressure flowing out from actuating chamber 2.

**[0027]** However, it seems that these known devices have not given satisfactory results. The object of the present invention is hence to suggest an electro-hydraulic valve system for the variable actuation of the valves of combustion engines, a servovalve structure and a method to control the same, which allow a variable actuation free from the drawbacks and limitations of the prior art. This result is obtained through the features mentioned in claims 1, 10 and 15; the dependant claims refer to the preferential features of the present invention.

**[0028]** Further features and advantages of the invention are in any case more evident from the following detailed description of a preferred embodiment, given purely by way of nonlimiting example and illustrated in the accompanying drawings, wherein:

figs. 1 to 4 are operation diagrams of prior-art engine valves, which have already been described heretofore;

fig. 5 is a general layout of the structure of a device

according to the present invention;

fig. 6 is a 3-section diagram representing the lift motion of the engine valve according to the invention, with reference to the actuation times of the control electrovalves;

fig. 7 is a diagram of a safety device associated with the engine valve control according to the invention;

fig. 8 is a diagram section of a one-directional, semi-automatic servovalve which is normally closed when in its home position; and

fig. 9 is a diagram section of a one-directional, semi-automatic servovalve, similar to the previous one, but normally open when in its home position.

**[0029]** As outlined in fig. 5, the electro-hydraulic system according to the present invention comprises:

- an actuator consisting of a cylinder 11 and of a piston 12, the latter acting directly on engine valve 13, with which it is integral, in opposition to a respective opposing spring 13a;
- a high-pressure source 14, which can be put in communication with chamber 11 of the actuator by actuating a semi-automatic electrovalve 16, of the "normally-closed" type;
- a low-pressure source 15, not necessarily at room pressure, with which chamber 11 of the actuator is normally in communication and wherefrom chamber 11 is sealed off if electrovalve 17, of the "normally-open" type, is actuated;
- a positive-displacement pump 18, which provides to maintain source 14 and the circuits connected thereto pressurised;
- a gas accumulator 19, having the function of dampening any pressure swings of high-pressure source 14.

**[0030]** The two electrovalves 16 and 17 are two semi-automatic, one-directional valves. In particular, electrovalve 16 allows the flow of fluid from the high-pressure circuit 14 to actuation chamber 11 only if it is actuated from the outside, while it automatically allows the flow of fluid in the opposite direction, only if the pressure in actuation chamber 11 is higher, by a set value established during the design stage, than that found in circuit 14. Electrovalve 17 is instead one-directional in the opposite direction, i.e. it opens automatically if the pressure in actuation chamber 11 is smaller than that found in low-pressure tank 15, while when it is actuated it prevents fluid from leaking from actuation chamber 11 to said tank 15.

**[0031]** The operation of the suggested system, as concerns the principle diagram, provides that electrovalve 16 normally be closed, when not energised, and that electrovalve 17 normally be open when not energised.

**[0032]** When both electrovalves 16 and 17 are energised, actuation chamber 11 enters into communication with high-pressure circuit 14 and is simultaneously sealed off from the low-pressure source; thereby the

pressure found in actuation chamber 11 will reach such levels as to overcome the preload of spring 13a, which tends to keep engine valve 13 closed on its seat: engine valve 13 then opens. The maximum lift value of engine valve 13 will be adjusted by acting on the opening times of electrovalve 16.

**[0033]** Closing of engine valve 13 is actuated by de-energising both electrovalves 16 and 17; actuation chamber 11 is thereby put in communication with low-pressure tank 15 while at the same time high-pressure source 14 is excluded. This causes a sudden pressure drop in actuation chamber 11, which allows the engine valve to close under the push of opposing spring 13a.

**[0034]** The management of the opening and closing runs of engine valve 13 substantially provides the same steps of the camless-type, electro-hydraulic system of the prior art, but the modes of operation differ. More specifically:

1) Opening, first step: acceleration of the engine valve. The initial starting condition is with a closed engine valve: in such condition electrovalve 16 is closed and electrovalve 17 is open, both being de-energised. Actuation chamber 11 is hence sealed off from high-pressure source 14 and in communication with low-pressure source 15. The opening run is started by simultaneously energising electrovalves 16 and 17: high-pressure circuit 14 enters into communication with actuation chamber 11, while the low-pressure tank is excluded. Engine valve 13, under the action of the pressurised fluid, moves from its seat at 0 speed and acceleration other than zero. Its speed begins to increase until its kinetic energy reaches a maximum value about halfway through the opening run.

2) Opening, second step: engine valve deceleration. From this position, about halfway through the opening run, in a condition of 0 acceleration, the subsequent step begins wherein the engine valve must be braked. As a matter of fact, in order to reach its maximum lift value at 0 speed, the kinetic energy of engine valve 13 must be reduced to 0. For this purpose, according to the present invention - by exploiting the fact that the two electrovalves 16 and 17 are semi-automatic valves - electrovalve 16 is simply de-energised, hence closing, while electrovalve 17, being actuated, remains still closed; since engine valve 13 still has all the kinetic energy acquired during the first half of the opening run, an increase of the volume of actuation chamber 11 will be produced, not associated with the entry of oil from high-pressure pipe 14. As already stated, this determines a pressure drop in actuation chamber 11; when such pressure drops below the pressure level of tank 15, semi-automatic electrovalve 17 opens and allows the oil to flow from tank 15 backwards, towards actuation chamber 11. In other words, in the actuation chamber 11 of the actuator, entry of oil from the low-pressure side, and

not from the high-pressure circuit, will be spontaneously produced; in this step, inside actuation chamber 11 there is a pressure slightly below that of tank 15 and therefore no oil vaporisation due to depression occurs. Moreover, the opposing action of spring 13a translates into braking of engine valve 13, which can reach its maximum lift value at 0 speed. It is important to stress that the deceleration of engine valve 13 does not occur by dissipating all the energy of the engine valve, but by using it to compress spring 13a. Once all its kinetic energy is depleted and once the maximum lift value is reached, engine valve 13 halts its motion and, under the push of spring 13a, begins to compress the oil found in actuation chamber 11. Thanks to this arrangement:

- there are no longer cavitation phenomena since, as already said, the volume increase of actuation chamber 11 is associated with oil entry from tank 15 to actuation chamber 11 of the actuator, thanks to the fact that semi-automatic electrovalve 17 is open in this condition;
- about half the work volume of the actuator is filled not by using the oil of high-pressure circuit 14, but by drawing it from low-pressure tank 15;

This last aspect is of particular importance from an energy point of view because, in the system according to the invention, the amount of oil required for the opening of engine valve 13, which is equal to the cylinder capacity corresponding to the stroke travelled by piston 12, is partly drawn from high-pressure circuit 14 at the expense of pump 18, but it is also partly drawn from low-pressure tank 15 and hence with no energy cost to the system.

3) Holding step: engine valve halted in an open position. In this step electromagnetic valves 16, 17 are both kept closed - electrovalve 16 de-energised and electrovalve 17 actuated - so that the engine valve is kept open around its maximum lift condition. In such condition the engine valve may still have a certain residual kinetic energy which will cause it to swing, but only slightly until the oscillations have been totally damped.

4) Closing step, first step: portion with accelerating engine valve. The holding step is followed by the engine valve closing step, for which both electrovalves 16 and 17 are de-energised. Since electrovalve 16 is normally closed and electrovalve 17 is normally open, the actuation chamber of the actuator remains sealed off from high pressure and put in communication with low-pressure tank 15. Opposing spring 13a is then free to act on engine valve 13, which will be pushed into closing. Under this push engine valve 13 is accelerated, but only until about halfway through its closing run; as can be understood, the elastic energy accumulated by spring 13a in the previous deceleration step 3) is converted back

into kinetic energy.

5) Closing run, second step: portion with decelerating engine valve. The deceleration of the engine valve is accomplished by exploiting precisely the feature of the one-directional, semi-automatic valve of electrovalve 16. At the end of the previous step, i.e. about halfway through the closing run, electrovalve 17 is actuated, so that actuation chamber 11 of the actuator is sealed off from low-pressure tank 15. The energy of engine valve 13, in this step approaching to its seat, compresses the oil in actuation chamber 11 of the actuator and, when the pressure here exceeds that found in high-pressure pipe 14, electrovalve 16 automatically opens, allowing to bring oil back to above-said pipe 14. Thereby the energy of engine valve 13, which would otherwise be dissipated into impacts with its seat and into pressure waves, is converted into pressure energy which is useful for subsequent cycles. Further energy recovery, not found in other prior art systems, is therefore obtained. In the ideal case of the absence of blow-by and friction leaks, in this step the motion of piston 12 resulting from that of engine valve 13 should pump to high-pressure pipe 14 all the fluid which has previously been drawn from low-pressure tank 15, and which is equal to the volume of actuation chamber 11 halfway through the opening run of the engine valve. Therefore, the work required from pump 18 is only that for compensating viscous, friction and aerodynamic leaks on the engine valve.

**[0035]** Taking into account the above, adjustment of the system occurs, as can be easily understood, by adjusting the actuating times of electrovalve 16 when opening and of electrovalve 17 when closing, respectively, always bearing in mind that these are semi-automatic valves whose closing and opening, respectively, occur automatically upon overstepping certain pressure threshold values in actuation chamber 11.

**[0036]** In fig. 6 an example of actuation of the engine valve and of electrovalves 16 and 17 is shown, from which it appears that to each step of the engine valve lift law corresponds an electric signal which controls the opening the high-pressure electrovalve or the closing of the low-pressure one. However, as can be observed in the last two diagrams of fig. 6, wherein the electrovalve opening laws are shown, there is no biunique correspondence between control signal and electrovalve opening or closure. This is due to the fact that the two electrovalves, being semiautomatic, open spontaneously in the cases in which the pressure gradients reach very high values. The openings or closures with which no electrical signal is associated are those concerning the "automatic" openings of the electrovalves in the opening and closing step of the engine valve, when the energy recovery occurs as mentioned in the preceding paragraphs. Hence:

- in order to vary the angular time of the opening law

of the engine valve, the holding step is adjusted, i.e. the time in which both electrovalves 16 and 17 are held closed, as stated at point 3) above;

- in order to adjust the maximum lift of the engine valve, the quantity of oil which is let into actuation chamber 11 of the actuator is instead adjusted. For this purpose, two different strategies can be followed: acting on the opening time of high-pressure electrovalve 16, always keeping constant the pressure in pipe 14, or modulating the pressure level in pipe 14 without varying the opening time of electrovalve 16.

**[0037]** As a matter of fact, it must be remembered that the mass  $M$  of the fluid having density  $\rho$  which, under a pressure difference  $\Delta P$ , flows out through a flow section  $A$  in a time  $\Delta t$  is given by:

$$M = \varphi * A * \sqrt{2\rho\Delta P} * \Delta t$$

$\varphi$  being the outflow coefficient; which shows how mass  $M$  is proportional both to the pressure time and to the pressure value.

**[0038]** An important, key advantage of the control system according to the present invention is its intrinsic safety, essentially due to the fact that - in case of failure of the device management system or of the power system or of the hydraulic circuit - it allows the valves to automatically close onto their seat, so as to avoid permanent damage to the engine in case of interference with the pistons. As a matter of fact:

a) the system according to the present invention is firstly intrinsically failure-proof as far as the power system is concerned. In this case it may happen that no signal or an incorrect signal arrives at electrovalves 16 and 17; if the engine valve remained open, it could end up interfering with its respective engine piston. However, as said, when the control electromagnets are not active, electrovalve 16 remains closed, while electrovalve 17 remains open; as a consequence, actuation chamber 11 of the actuator is in communication with low-pressure tank 15 and engine valve 13 can close with no outer intervention, but only due to the effect of the action of opposing spring 13a. The problem is therefore automatically solved in a very simple manner;

b) in addition thereto, the system according to the present invention can be integrated with an arrangement such as the one outlined in fig. 7, which allows to remedy both to a failure of the control system of the control signals of the electrovalves, and to a failure of the entire hydraulic system. The arrangement of fig. 7 in fact comprises an inhibiting assembly 20 arranged between the electronic control unit ECU and the VVA system according to the invention; a

cam 21 rigidly connected to the engine shaft and which consequently rotates therewith belongs to assembly 20. Cam 21 has a recessed circular sector 21a, which has an angular length as wide as the crank angle for which an interference between piston and engine valve would be possible. A sensor 22, for example a Hall-effect sensor, is capable of delivering a rectangular signal, whose part other than zero corresponds to length 21b of the non-recessed profile of the cam. Thereby, as long as sensor 22 is in contact with non-recessed profile 21b, there is the certainty that this is not the area of possible piston/engine-valve interference; sensor 22 will generate a nonvoid signal to the inhibiting device, which will therefore remain closed, and in such condition the control signals which the ECU unit sends will be able to duly reach the VVA. When the piston lies instead precisely in the area of possible interference with engine valve 13, then sensor 22 is in contact with the recessed profile 21a and will generate a void signal to the inhibiting device, which therefore opens and interrupts the transmission of the control signals from the ECU unit to the VVA.

**[0039]** Thanks to the arrangement of fig. 7 it is then firstly possible to remedy to a failure affecting the control system of the control signals of the electrovalves, such that the signals may be sent with wrong timelines: it is clear that this could lead to the engine valve opening when the piston is near the top dead point, hence with a serious risk of interfering with the open engine valve. For this purpose sensor 22 - which at this time is in contact with recessed area 21a - delivers a void signal, so that the inhibiting device opens and, even if the ECU was to send incorrect signals, these could not reach the VVA or, more precisely, electrovalves 16, 17 which, due to their design, remain closed.

**[0040]** It is to be noted that, in order to make the system intrinsically safe, it is provided that the inhibiting device cannot cause the opening of the engine valve, but only prevent it. This implies that, in case of failure of the inhibiting device, the event of an accidental engine valve opening and, consequently, of a possible impact with the piston cannot occur, since such a failure will cause the absence of the signal to the VVA and, as a result, the closure of electrovalve 16 and the opening of electrovalve 17, hence the closure of engine valve 13.

**[0041]** In case of failure of the hydraulic part, for example a circuit leak or a pump failure, the circuit does not become pressurised. In this event, the protection system will cause the low-pressure electrovalve to open as soon as the potential interference area is entered. Thereby, any residual fluid will be discharged into the actuation chamber, allowing the engine valve to close under the effect of the spring, avoiding the impact of the latter with the piston.

**[0042]** Finally, should an impact between valves and piston occur due to an unexpected reason, the system

according to the invention is capable of minimising damage, since it takes on a "non-rigid" behaviour. This is because a possible, unexpected lift of the engine valve will generate a pressure increase in actuation chamber 11 up until the pressure value in high-pressure pipe 14; at this point the high-pressure electrovalve will open automatically allowing oil to flow back towards pipe 14. In practice high-pressure electrovalve 16 works like a relief valve of the circuit, adjusted to the nominal pressure value in pipe 14 and, therefore, it allows to limit loads and, consequently, damage in case of accidental impact.

**[0043]** Although a system of the type described above can employ semi-automatic electrovalves which may be easily obtained on the market, these have the drawback, however, at least in general, of requiring relatively high power values to actuate the control electromagnet, especially if very short reaction times are required, as is the case for use in VVA systems.

**[0044]** Here in the following electrovalve structures are therefore described which may be employed in the VVA system for the control of the valves of internal combustion engines, which provides an electric-hydraulic actuation such as the one set forth above with reference to servovalves 16 and 17 outlined in fig. 5. The servovalves illustrated here below are particularly suited to overcome the drawbacks of the prior art and allow reduced manufacturing costs, both of the servovalve and of the corresponding control means, as well as high operation speeds.

**[0045]** The servovalve arrangement shown in fig. 8 is particularly suited to be used as a connection servovalve to high-pressure line HP. It is depicted mounted on a base body B, wherein actuation chamber 11 of piston 12 of the actuator is formed. Such a valve, connecting high-pressure circuit 14 to chamber 11 for the actuation of piston 12 controlling engine valve 13, is expected to meet the following functional requirements:

- ☐ with the electromagnet not energised, the servovalve is normally closed and does not let control fluid flow from high-pressure circuit 14 to actuation chamber 11;
- ☐ even in a non-energised condition, i.e. closed, the servovalve automatically opens when the pressure in actuation chamber 11 exceeds the pressure in high-pressure circuit 14 HP;
- ☐ very quick actuation;
- ☐ low energy consumption.

**[0046]** The servovalve illustrated in fig. 8 essentially consists of a valve body 16, within which a shutter 23 of the hydraulic valve and an actuation electromagnet EL are essentially housed.

**[0047]** Shutter 23 moves within a cylindrical chamber which forms a lower volume 24, ring-shaped around the conical lower end of shutter 23, and an upper volume 25.

**[0048]** At the lower exit of the cylindrical chamber housing said shutter 23 there is provided a pipe 26, which

puts in communication said cylindrical chamber of the shutter with the chamber 11 for the actuation of the cylinder housing piston 12 for the control of engine valve 13 (not shown in fig. 8). In a rest condition, shutter 23 closes pipe 26, pushed by a spring 37 towards the bottom in the drawing.

**[0049]** High-pressure fluid HP coming from channel 16 arrives at the servovalve through pipe 28. The flow divides itself into two pipes 29 and 30: through pipe 30 the work fluid arrives at ring-like volume 24, which is limited by the lower end of mobile shutter 23 when the latter is closed onto the seat of pipe 26. Through pipe 29 the fluid arrives instead at the upper volume 25 of the cylindrical chamber housing shutter 23.

**[0050]** A plate 31 separates the cylindrical chamber of shutter 23 from the chamber housing electromagnet EL. In plate 31 there is drilled a communication hole 32 between said two chambers. A sphere 33 rests in a conical seat formed in plate 31, at the top of hole 32. Sphere 33 is pushed onto said conical seat, sealing off pipe 32, by a small anchor 34 made of magnetic material, pushed by a helical spring 35. By adjusting the preload of this spring 35 it is possible to impart to sphere 33 a force (directed downwards in the drawing) greater than the one which the pressure of the fluid found in volume 25 can impart to sphere 33, through pipe 32, in the opposite direction.

**[0051]** Thereby it is possible to keep sphere 33 closed on its seat, with a relatively small force - due to the small flow section of pipe 32 - also in the presence of very strong pressures in volume 25. A relatively low preload of spring 35 will therefore be sufficient.

**[0052]** Small anchor 34, on the other hand, also undergoes the action of electromagnet EL which, when energised, recalls it upwards in opposition to spring 35. Due to what has been said heretofore, if the preload of spring 35 can be kept low, the force will also be modest which electromagnet EL has to impart to recall anchor 34 and release sphere 33. This implies the advantage of using a small magnet with low power absorption.

**[0053]** At this point, considering the forces acting on shutter 23, it can be observed that they consist of:

- an elastic force imparted by helical spring 37 and directed downwards in the drawing;
- a force, equally directed downwards, due to the pressure of the fluid found in volume 25, when pipe 32 is closed by sphere 33;
- an upward force, originated by the pressure of the fluid found in volume 24; since the fluid pressure here is the same as the pressure within volume 25, and nonetheless it acts on a smaller surface (only an annular portion of the upper one of shutter 23), it is clear that this upward force will have a smaller absolute value than the downward force resulting from the pressure of the fluid in volume 25; and
- an upward force, originated by the pressure of the fluid within pipe 26 and acting on the part of shutter

23 which opens into pipe 26.

**[0054]** The sum of these forces is determined so that, in the condition shown in figure 8, there is a downward resultant which keeps shutter 23 closed on the conical seat of pipe 26. Thereby, electrovalve 16 is closed and the fluid coming from high-pressure pipe 28 cannot reach the actuation chamber 11 of actuator 12. Electrovalve 16 is hence, as detailed above, a normally-closed valve.

**[0055]** When electromagnet EL is energised, it causes anchor 34 to rise, against the action of helical spring 35, which is thereby loaded. Thereby the load on sphere 33 ceases and pipe 32 is opened towards volume 25, which then appears to be in communication with the volume housing said electromagnet EL.

**[0056]** This volume, above pipe 32, however, is in communication with the low-pressure (or room pressure) line through pipes 38 and 39 so that, as soon as sphere 33 uncovers pipe 32, a sharp pressure drop occurs in volume 25. Dynamically, thereby, above shutter 23 there is established an intermediate pressure between low pressure and the pressure within high-pressure circuit 14. Thereby shutter 23 is pushed against the upper end-of-stroke found in volume 25, and pipe 26 can be connected to volume 24, allowing the high pressure to reach actuation chamber 11 through pipes 28 and 30. Valve 16, as detailed above, opens following the energisation of electromagnet EL.

**[0057]** In such opening step, hole 29 is partly covered by shutter 23; such solution helps to speed up the opening of shutter 23 since, hole 29 being partly obstructed, a load loss will occur through it, which will lead the pressure in volume 25 to drop more quickly than if said hole was fully open, hence speeding up, as said, the opening dynamics of pipe 26.

**[0058]** In addition to speeding up the subsequent closing step of shutter 23, helical spring 37 allows, through adjustment of its preload, to determine the maximum differential pressure value below which shutter 23 remains always closed, if the magnet is not energised. Such property of the system is exploited to use the same electrovalve 16 as a one-directional, automatic valve, in the direction from actuation chamber 11 to high-pressure pipe 14; i.e. a servovalve opening automatically in case of excessive pressure in pipe 26. As a matter of fact, if the pressure in actuation chamber 11 reaches very high values, exceeding by a preset value the high pressure HP in pipe 14, a pressure force will act on shutter 23 of fig. 8, through pipe 26, which will change the balance of forces on shutter 23, originating a resulting overall upward force which will cause shutter 23 to rise again allowing the fluid to flow from actuation chamber 11 to high-pressure pipe 14.

**[0059]** From what has been said above it is evident that the opening and closing of the electrovalve occurs thanks to a pressure imbalance between the lower and upper part of the shutter. The electromagnet serves the only purpose of triggering such imbalance. For such rea-

son the electrovalve is to be considered as a servovalve wherein the opening is controlled by the magnet, but is accomplished by a pressurised fluid. The magnet will have to impart only the force required to overcome the preload of the spring which keeps sphere 33 closed on pipe 32. Such preload is defined so as to overcome the force which the high pressure HP in pipe 14 imparts to said sphere 33 through pipe 32. Such pipe having a very small section, even if high pressure HP is very high, the resulting force is small and, hence, the preload of spring 35 will also be small. As a result, the energy required by said electromagnet will be modest. Such energy, however, will allow to cause pressure imbalances which will allow the opening of the valve also against high loads. Moreover, this system allows to obtain very quick opening times even with relatively long runs of the shutter, hence with significant uncovered sections[j1]. The features of the servovalve thereby allow to obtain a very quick system, but featuring modest energy consumption.

[0060] The servovalve shown in fig. 9 is suitable to be used instead as connection servovalve 17 to low pressure LP. Such a servovalve, connecting low-pressure tank 15 to chamber 11 for the actuation of piston 12 controlling engine valve 13, is expected to meet the following functional requirements:

- ☐ In de-energised conditions the servovalve is normally open; it is closed upon energisation of its corresponding electromagnet;
- ☐ The servovalve, even when energised and hence closed, opens automatically and lets work fluid flow into the actuation chamber 11 of the actuator if the pressure in said chamber is below that of low-pressure tank 15;
- ☐ very quick actuation;
- ☐ low energy consumption.

[0061] Structurally, electrovalve 17 - shown in section in fig. 9, fitted sideways to the same base body B, shown in fig. 8, wherein chamber 11 for the actuation of piston 12 of the actuator is obtained - is fully similar to the above-described electrovalve 16, except for the arrangement of magnet EL' and of corresponding anchor 34'. These elements (marked by the same references of fig. 8, but bearing an apex) appear to be inverted compared to the arrangement of fig. 8; therefore, with a de-energised magnet, helical spring 35' keeps anchor 34' raised. In these conditions shutter 33 is free and allows fluid to flow through pipe 32, to put volume 25 in communication with low pressure; the servovalve can then be considered as a normally-open valve.

[0062] The closure thereof occurs when electromagnet EL' is actuated, which leads anchor 34' to push shutter 33 into closure on the seat of pipe 32, so that the communication of actuation chamber 11 with the source 15 of low pressure LP closes. It must further be noted that in this case the actuation pressure, which allows shutter closing and opening, is the one found in the ac-

tuation chamber 11 of the actuator and not that of low-pressure tank 15.

[0063] It is understood, however, that the invention is not to be considered limited to the specific arrangements illustrated above, which represent only example embodiments thereof, but that different variants are possible, all within the reach of a person skilled in the field, without departing from the scope of protection of the invention, as defined in the following claims.

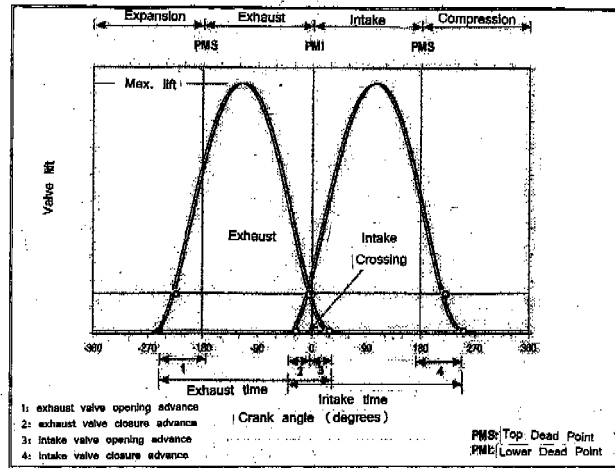
## Claims

1. Electro-hydraulic system for controlling valves of internal combustion engines, of the type comprising a cylinder-piston assembly for each engine valve (13) to be actuated, and means for transferring the piston movement to the engine valve stem, means (14, 16) for feeding control fluid to the cylinder chamber (11), for actuating said piston (12) to cause valve opening, in opposition to spring means (13a) which recall said engine valve (13) to the closed position thereof, said cylinder chamber (11) being in communication, on the one side, with a high-pressure line (14) through a first actuatable servovalve (16) and, on the other side, with a low-pressure chamber (15) through a second actuatable electrovalve (17),  
**characterised in that** said first electrovalve (16) is a semi-automatic electrovalve, normally closed when not actuated, and **in that** said second electrovalve (17) is a semi-automatic electrovalve (17), normally open when not actuated.
2. Electro-hydraulic system as claimed in claim 1), **characterised in that** said first electrovalve (16) is of the type which opens automatically when fluid pressure in the downstream line, between said electrovalve and the cylinder chamber (11), lies above the nominal pressure of the fluid in the high-pressure line (14).
3. Electro-hydraulic system as claimed in claim 1), **characterised in that** said second electrovalve (17) is of the type which opens automatically when fluid pressure in the upstream line, between said electrovalve and the cylinder chamber (11), lies below the nominal pressure of the fluid in the low-pressure chamber (15).
4. Electro-hydraulic system as claimed in claim 1), **characterised in that** said high-pressure line is connected with a high-pressure supply pump.
5. Electro-hydraulic system as claimed in claim 1) or 4), **characterised in that** said high-pressure line is connected with a gas accumulator (19) acting as a dampener of pressure variations in the high-pressure line (14).

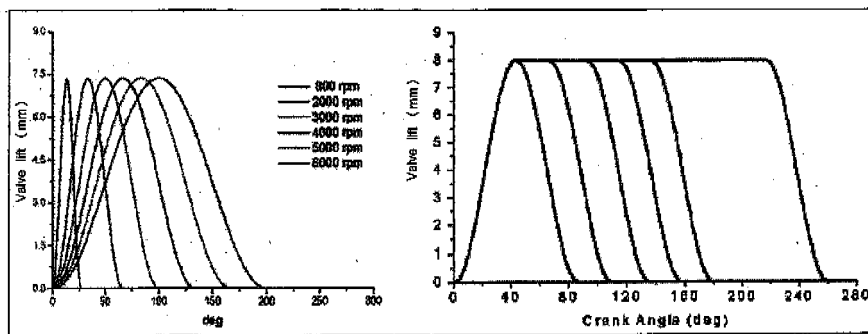
6. Electro-hydraulic system as claimed in any one of the preceding claims, **characterised in that** it comprises means (10-22) interrupting the electrovalve control signals, said means operating between an electronic control unit and the same electro-hydraulic system and acting in correspondence of the crank arc sector for which the respective piston moves in the area of possible interference between piston and engine valve.
7. Electro-hydraulic system as claimed in claim 6), **characterised in that** said interruption means comprise a cam (21), which rotates with the engine shaft, and with the profile of whose cam (21a, 21b) a position sensor cooperates, said sensor transmitting an interruption signal when it finds itself on the part (21a) of said profile corresponding to said crank arc sector.
8. Electro-hydraulic system as claimed in claim 7), **characterised in that** said sensor is a Hall-effect device, which generates a rectangular signal, the value whereof is nil in correspondence of said part (21a) of the cam profile.
9. Electro-hydraulic system as claimed in claim 7), **characterised in that** said sensor is a microswitch, which is open when the corresponding operating button thereof lies in correspondence of a recessed area (21a) of the cam profile.
10. Servovalve which may be employed in a system actuating the valves of internal combustion engines as in any one of the preceding claims, in particular as a servovalve intended to put in communication the cylinder chamber (11) of a cylinder-piston assembly (11-12) forming an actuator of each engine valve (13) to be actuated, at one end with a high-pressure line (14) and at the other end with a low-pressure tank (15), **characterised in that** it comprises a shutter (23), which is slidable, within a guide defining two volumes (24, 25) formed in correspondence of two opposite faces of the shutter (23), between a closed position and an open position of the supply line (26) to the cylinder chamber (11) of the actuator, and an electromagnet (EL) acting on a non-return valve (33), said valve acting on a passage (32) which puts in communication with low pressure the one (25) of said two volumes (24, 25) which lies on the side opposite to the volume (24) associated with said supply line (26).
11. Servovalve as claimed in claim 10), **characterised in that** said shutter (23) is formed as a cylindrical symmetry body, slidable within an axial seat of valve body (B), said seat forming said guide, said shutter (23) having a first end housed in the volume (24) associated with said supply line (26) and which forms a closing cone of said supply line (26) to the cylinder chamber (11), and a second end housed in the volume (25) associated with said passage (32), the surface area of said first end undergoing the pressure found in the first volume (24) being smaller than the surface area of said second end undergoing the pressure found in the second volume (25).
12. Servovalve as claimed in claim 10) or 11), **characterised in that** said electromagnet acts on said non-return valve (33) through a magnetic anchor (34), in opposition to the action of opposing spring means (35).
13. Servovalve as claimed in claim 12), **characterised in that**, when used as a servovalve (16) to put said supply line (26) and said cylinder chamber (11) in communication with the high pressure (HP), said opposing spring means (35) lie so as to displace said anchor (34) to control the closing of said non-return valve (33) on said passage (32).
14. Servovalve as claimed in claim 12), **characterised in that**, when used as a servovalve (17) to put said supply line (26) and said cylinder chamber (11) in communication with the low pressure (LP), said opposing spring means (35') lie so as to displace said anchor (34') to move away from said non-return valve (33) to cause said passage (32) to open.
15. Method to control an electro-hydraulic system for actuating the valves of internal combustion engines as claimed in any one of claims 1) to 9), **characterised in that** the opening of the engine valve (13) occurs in two phases:
- an engine valve (13) acceleration phase, wherein said first electrovalve (16) is open for a time corresponding to a part only of the opening run of said engine valve (13), and said second electrovalve (17) is closed,
  - and a subsequent engine valve (13) deceleration phase, wherein said first electrovalve (16) is closed for a time corresponding to at least to the remaining part of the opening run of said engine valve (13), while said second electrovalve (17) opens automatically, following the depression which is caused in said cylinder chamber (11) upon the further advance of the piston (12), due to inertia, so as to allow fluid intake from said low-pressure chamber (15) to said cylinder chamber (11).
16. Method as claimed in claim 15), **characterised in that** the closure of the engine valve (13) occurs in two phases:

- an engine valve (13) acceleration phase, wherein said first (16) and said second (17) electrovalves are de-energised, the first being closed and the second open, for a time corresponding to a part only of the closing run of said engine valve (13), and said spring means (13a) push said engine valve (13) back into its closed position,  
- and a subsequent engine valve (13) deceleration phase, wherein said second electrovalve (17) is closed for a time corresponding to at least the remaining part of the closing run of said engine valve (13), and said first electrovalve (16) opens automatically, following the overpressure generated in the cylinder chamber (11) upon advance of the piston (12) under the action of said spring means (13a), so as to allow discharge of the pressurised fluid in said cylinder chamber (11) towards the high-pressure line (HP).

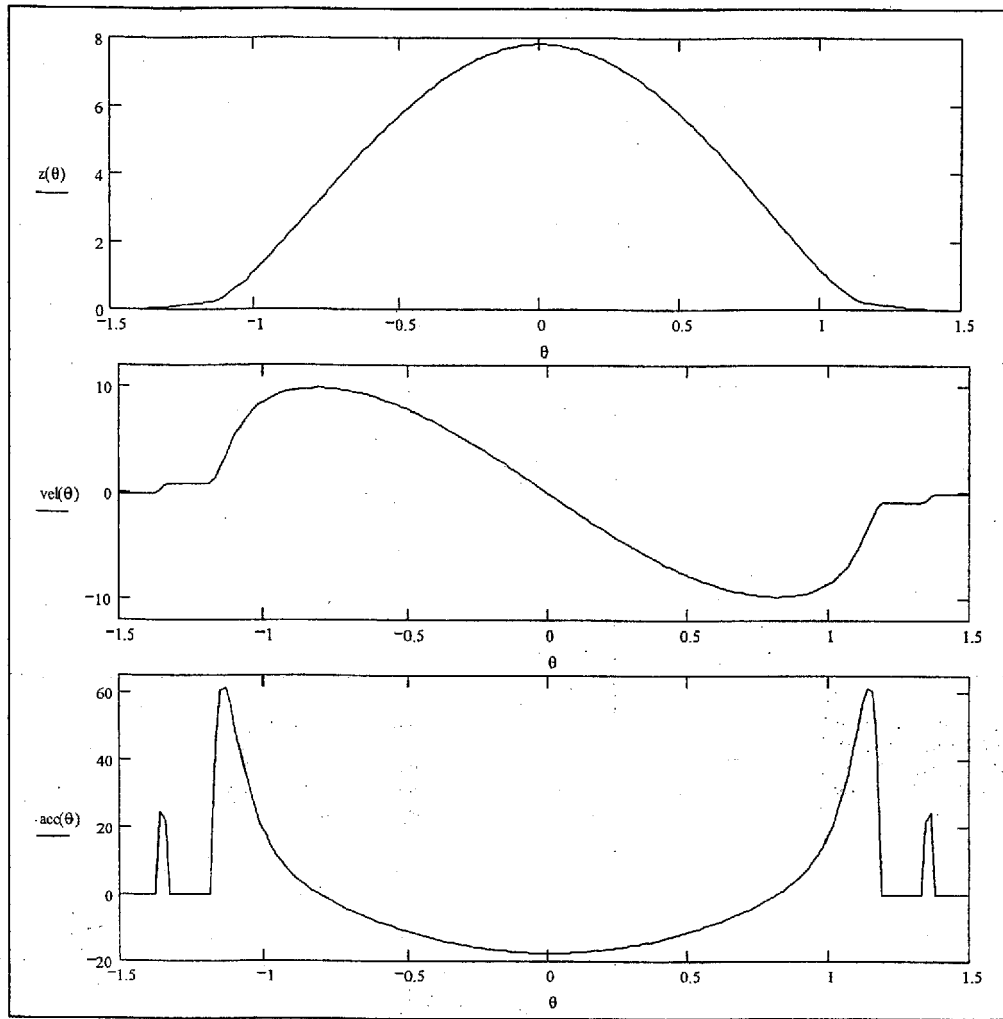
17. Method for the operation of the valves of internal combustion engines as claimed in claim 15) and 16), **characterised in that** between said opening steps of the engine valve (13) and said closing steps of the same, a step is provided during which the engine valve (13) is kept in an open position, said step being actuated by closing both said electrovalves (16, 17) for a desired length of time.



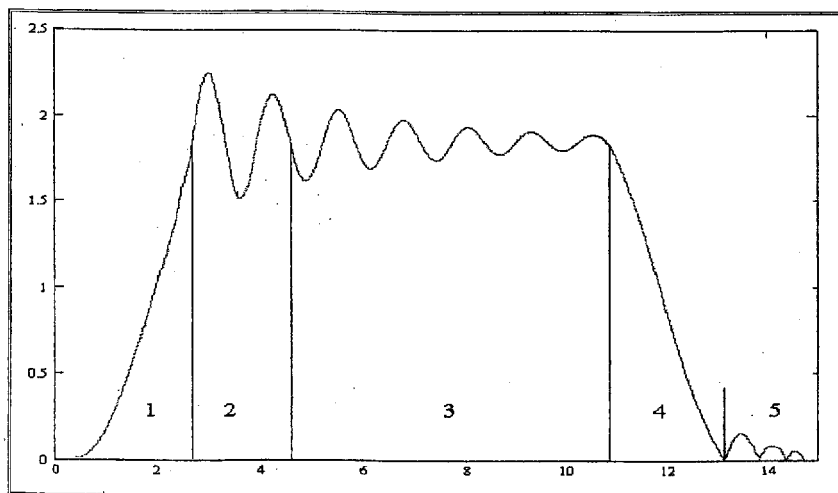
**Fig. 1**



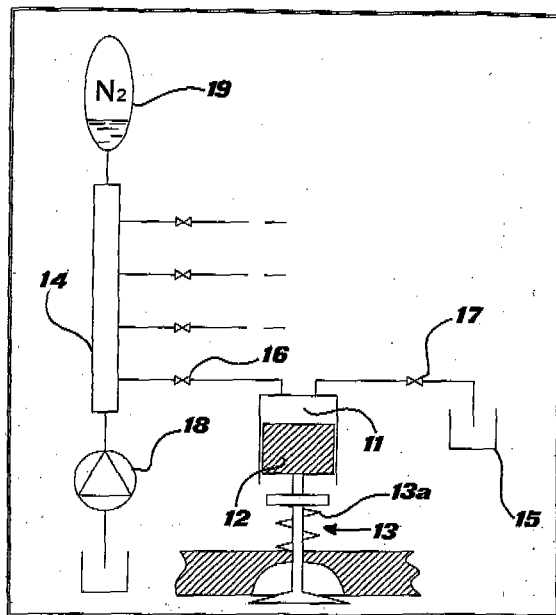
**Fig. 2**



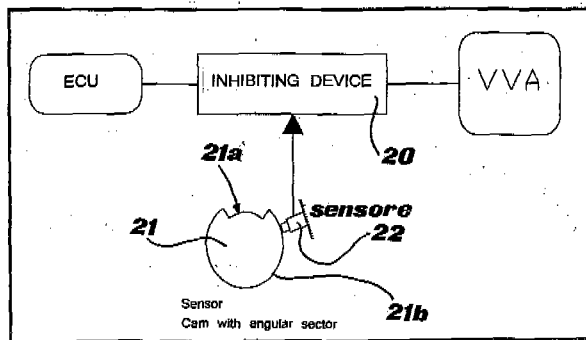
**Fig. 3**



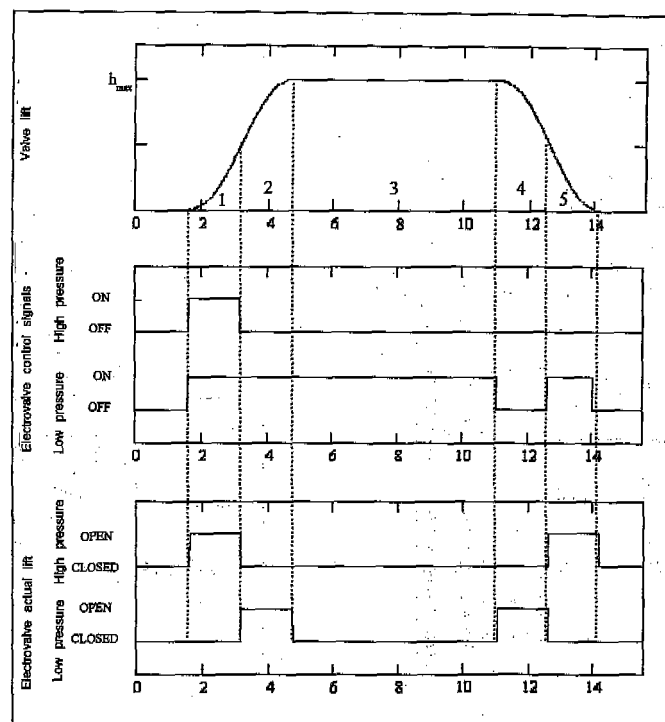
**Fig. 4**



**Fig. 5**



**Fig. 7**



**Fig. 6**

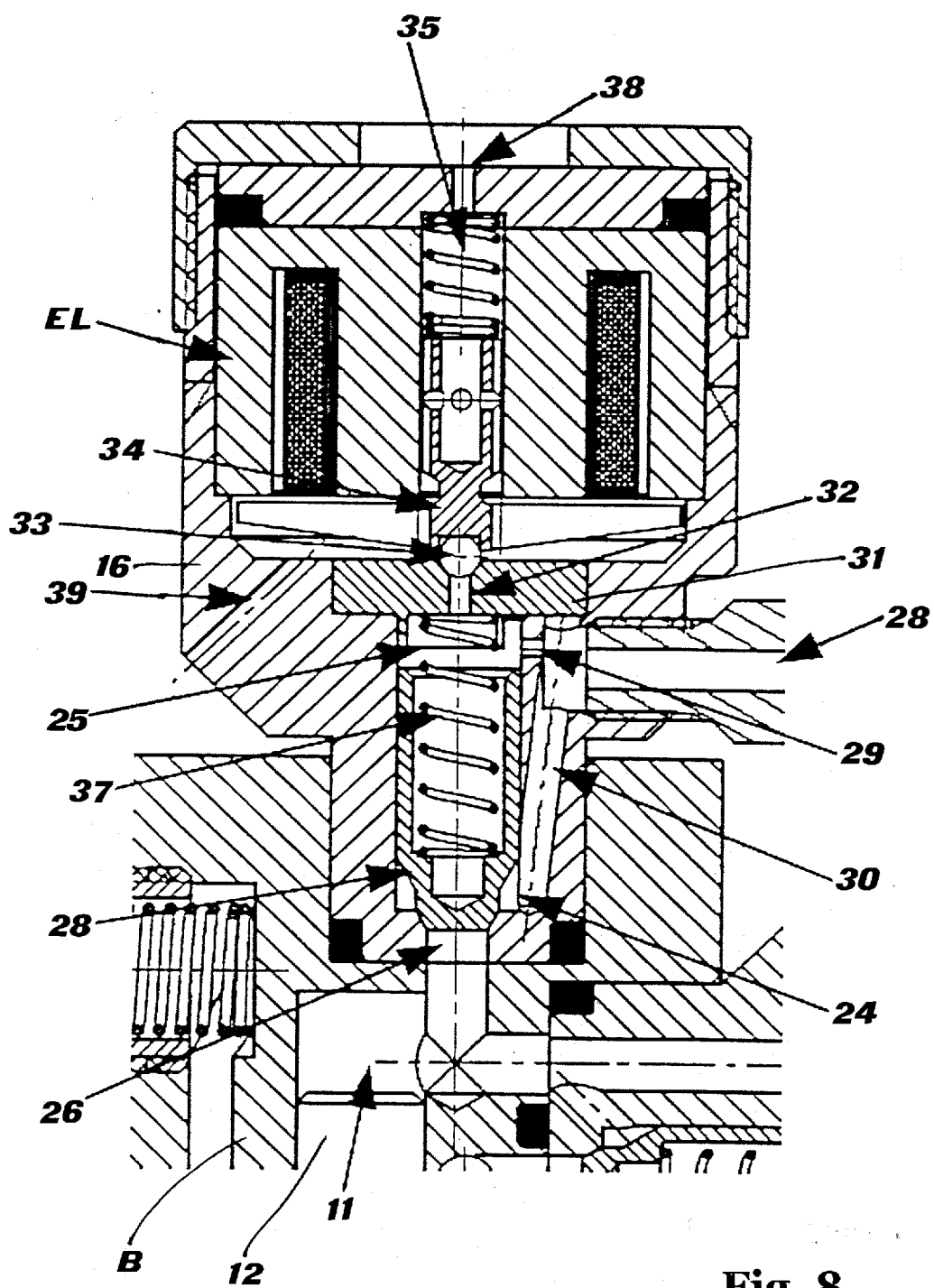
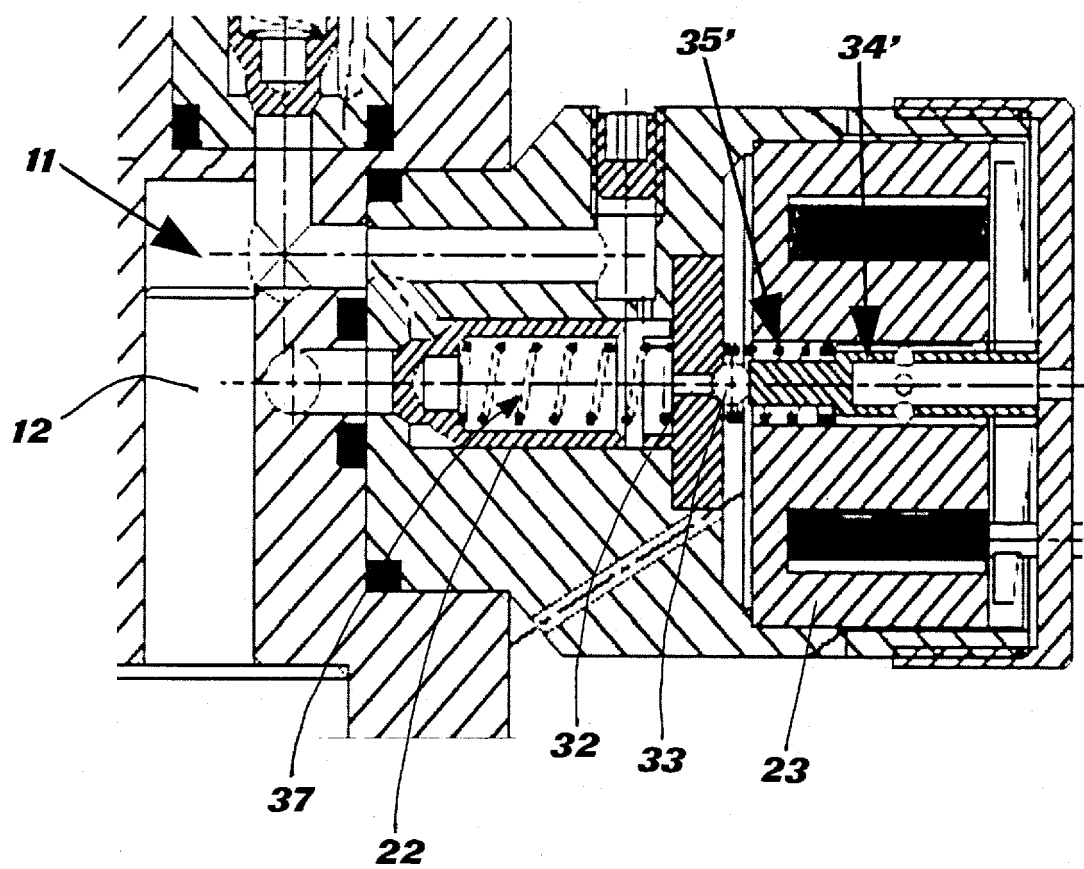


Fig. 8

**Fig. 9**



**REFERENCES CITED IN THE DESCRIPTION**

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