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(54) Hydraulic pump, in particular a fuel pump

(57) A hydraulic pump (HP; 1), in particular a fuel pump, comprising:
- an intake port (IP; 2),
- a delivery port (DP; 4), provided for hydraulic connection to a user (CR),
- at least one cylinder (CY; 20, 22, 24), movable within which is a corresponding piston (P; P1, P2, P3), wherein the piston has, during the operation of the pump (HP, 1), a reciprocating motion between a top dead centre (TDC) and a bottom dead centre (BDC),
- a delivery valve (DV; 26, 28, 30) for each cylinder (CY; 20, 22, 24) hydraulically connected thereto, movable between a closed position and an open position, wherein in the open position the delivery valve (DV; 26, 28, 30) is arranged for enabling a flow of fluid between the cylinder (CY; 20, 22, 24) and the delivery port (DP; 4), and wherein in the closed position the delivery valve (DV; 26, 28, 30) is arranged for blocking a flow of fluid between the cylinder (CY; 20, 22, 24) and the delivery port (DP; 4).

The hydraulic pump (HP; 1) comprises a regulation assembly (R; 39, 40, 42, 43, 44, 46) which, when a threshold pressure (p_{REF}) in a user (U; CR) hydraulically connected to said delivery port (DP; 4) is exceeded, is arranged for keeping the delivery valve (DV, 26, 28, 30) in the open position during at least part of a movement of the piston from the top dead centre (TDC) to the bottom dead centre (BDC) and enabling a reflux of fluid from the delivery port (4, DP) to the cylinder (20, 22, 24).

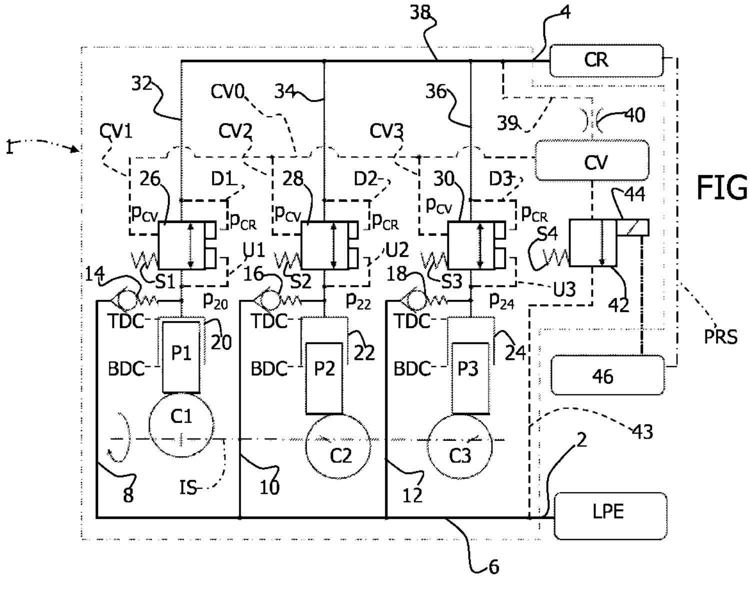

FIG.3

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Description

Field of the invention

[0001] The present invention relates to a hydraulic pump, in particular a fuel pump. In particular, the invention has been developed with reference to hydraulic pumps with devices for regulating the flowrate.

Description of the prior art and general technical problem

[0002] In the technical field of hydraulic pumps, in particular fuel pumps, there is the need to regulate the flowrate of fluid delivered by the pump to a user in a way substantially independent of the speed of rotation of the pump shaft.

[0003] In particular, in the field of compression ignition internal combustion engines (whether they be engines for road vehicles or large stationary engines for marine use), practically all the known solutions envisage hydraulic connection of a fuel pump to a pressurized-fluid accumulator (commonly referred to as "common rail"), and regulation of the flowrate delivered by the pump to the common rail is obtained substantially according to two different ways:

- by means of a regulation valve that laminates the excess flowrate delivered by the pump and not consumed by the injectors supplied via the common rail,
- by means of a lamination valve set at the inlet of the pump to cause a controlled cavitation in the fluid sucked in by the pump itself.

[0004] Evidently, the latter mode is aimed at the reduction of the mass of liquid sucked in by the pump.

[0005] However, notwithstanding their undoubted simplicity, said regulation strategies present evident disadvantages.

[0006] The first way of regulation is energetically very expensive in so far as the lamination of fluid that is obtained entails high energy losses.

[0007] The second way of regulation presents big problems of wear linked to the cavitation induced in the fluid at the pump inlet. Furthermore, said solution normally requires the use of a lamination valve of an electroactuated type with a costly proportional solenoid to be able to vary with continuity and precision the flowrate of fluid that is sucked in by the pump.

Object of the invention

[0008] The object of the present invention is to overcome the technical problems described previously.

[0009] In particular, the object of the invention is to regulate the flowrate delivered by a hydraulic pump to a user in an energetically convenient way and so as not to jeopardize the service life of the pump and of the components thereof.

Summary of the invention

[0010] The object of the present invention is achieved by a hydraulic pump having the features forming the subject of the ensuing claims, which form an integral part of the technical disclosure herein provided in relation to the invention.

Brief description of the figures

[0011] The invention will now be described with reference to the annexed figures, provided purely by way of non-limiting example, wherein:

- Figure 1 is a schematic representation of a hydraulic pump according to various embodiments of the invention;
- Figure 2 is a schematic representation of a hydraulic pump according to an embodiment of the invention;
- Figure 3 is a schematic representation of a hydraulic pump according to a preferred embodiment of the present invention; and
- Figures 4 to 8 illustrate a series of diagrams describing various characteristic quantities of operation of a hydraulic pump according to the preferred embodiment of the invention.

Detailed description

[0012] In Figure 1 the reference HP designates a hydraulic pump according to various embodiments of the present invention. The pump HP comprises an intake port IP, a delivery port DP, and at least one cylinder CY, set within which is a piston P that moves with reciprocating motion by means of a mechanism K (for example, a cam or a crank mechanism), which is in turn actuated by means of an input shaft IS. Each piston P moves with reciprocating motion between a top dead centre TDC and a bottom dead centre BDC.

[0013] The intake port IP is arranged for connection to an intake environment (not illustrated) and is in fluid communication with the cylinder CY by means of a intake valve IV, in itself known.

[0014] The cylinder CY is moreover in fluid communication with the delivery port DP by means of a delivery line DL, set on which is a delivery valve DV that can be controlled by means of a regulation assembly R.

[0015] The delivery valve DV is movable between an open position, where it is arranged for enabling a flow of fluid between the cylinder CY and the delivery port DP, and a closed position, where it prevents the aforesaid flow of fluid between the cylinder CY and the delivery port DP. The pump HP is moreover arranged for hydraulic connection to a user, here represented schematically and designated by the letter U. In one embodiment, the user U can be a fuel-accumulation injection system commonly known as "common-rail injection system ".

[0016] Operation of the pump HP is described in what follows.

[0017] The piston P in reciprocating motion between the top dead centre TDC and the bottom dead centre BDC describes a working cycle comprising a sequence of five phases, namely,

- intake of fluid, in particular liquid, within the cylinder CY from the intake port IP (intake),
- compression of the liquid within the cylinder CY (compression),
- delivery of the liquid to the delivery line DL (delivery),
- reflux of the liquid from the delivery line DL to the cylinder CY (reflux),
- expansion of the residual liquid present in the cylinder CY (expansion).

[0018] Of these, the compression and delivery phases develop substantially during the rise of the piston from the bottom dead centre BDC to the top dead centre TDC, whereas the reflux, expansion, and intake phases develop substantially during descent of the piston from the top dead centre TDC to the bottom dead centre BDC. In the ensuing description, the former will be referred to briefly as "rising phase", and the latter as "descent phase".

[0019] During the expansion phase, descent of the piston P to the bottom dead centre BDC creates a reduction of pressure within the cylinder CY, which brings about the opening of the intake valve IV. The intake phase thus starts, with the consequent entry of fluid into the cylinder CY itself. The intake valve IV substantially recloses at the end of the descent phase of the piston P.

[0020] There follows the compression phase, which terminates when the pressure in the cylinder CY has reached a value such that the regulation assembly R opens the delivery valve DV, thus enabling transfer of the pressurized fluid through the delivery line DL to the delivery port DP.

[0021] For this purpose, the regulation assembly is arranged for receiving a first driving signal PS1, preferentially corresponding to a delivery pressure immediately downstream of the cylinder CY and issuing, as a function of said driving signal PS1, an actuation signal AS1 to the delivery valve DV, for example an actuation force developed with mechanical actuators or with hydraulic means. The actuation force developed on the valve DV causes the opening thereof, thereby enabling the passage of fluid through the delivery line DL and the delivery thereof to the user U through the delivery port DP.

[0022] At the end of the delivery phase, the regulation assembly R ceases issuing the actuation signal AS1, enabling closing of the delivery valve DV.

[0023] When the flowrate of fluid sent by the pump HP to the user U exceeds the demand of the latter, the pressure of the fluid in the user U (and at the delivery port DP) increases on account of accumulation thereof.

[0024] The regulation assembly R is arranged for receiving a second driving signal PS2 corresponding to a pressure of the fluid in the user U. In case the pressure detected in the user U is higher than a threshold value p_{REF}, the regulation assembly R continues to issue the actuation signal AS1 for keeping the actuation force on the delivery valve DV, and in turn keeping the delivery valve DV in the open position even after the end of the delivery phase. In other words, the regulation assembly R keeps the delivery valve open also during at least part of the descent phase of the piston P.

[0025] In this way, a reflux of fluid is enabled from the delivery port DP to the cylinder CY through the delivery valve DV. It should be noted that during the reflux of fluid to the cylinder CY, this develops a motive work on the piston P, substantially returning the work of compression previously accumulated by the fluid. Consequently, said mechanism of regulation, which envisages delivering a full flowrate of fluid and then causing reflux of the excess amount to the cylinder CY, does not penalize the overall energy efficiency of the pump.

[0026] In effect, the reflux phase performed by the piston P, which in conditions of operation at full flowrate is substantially an undesirable effect due to the inertia of the system (delivery valve above all), becomes in conditions of regulation an effect that is sought and develops over a much wider angular range (with reference to an angle of rotation of the input shaft IS) that extends, in the limit, as far as the bottom dead centre BDC (there is thus witnessed a substantial dilation of the reflux phase, with delay and substantial contraction of the intake phase).

[0027] In this way, the pressure of the fluid in the user U decreases (as the pressure on the delivery port DP) returning into the neighbourhood of the threshold value p_{REF}. The regulation assembly R is thus arranged for ceasing the maintenance of the actuation signal AS1 on the delivery valve DV when the pressure of the fluid in the user U drops below the threshold pressure p_{REF}, so that the valve DV is able to reclose.

[0028] The operation of the regulation assembly R is such that, in conditions of regulation, the pressure of the fluid in the user U oscillates with respect to the reference value, with reiteration of the regulation process just described. In this way, it is possible to guarantee continuously that a flow-rate value equal to the one required reaches the user U.

[0029] With reference to Figure 2, in one embodiment the pump HP comprises an actuator A1, arranged for the actuation of the delivery valve DV, and an elastic positioning element S, whose action tends to bring the delivery valve DV into the closed position.

[0030] The operation is similar to what has been described previously because the regulation assembly R is arranged for controlling the actuator A1 so as to bring the delivery valve DV into the open position during the delivery phase (as a function of the driving signal PS1) and so as to keep it open during the reflux phase as a function of the driving signal PS2, i.e., of the pressure in the user U.

[0031] With reference to Figure 3, the reference number 1 designates as a whole a hydraulic pump according to a preferred embodiment of the present invention.

[0032] The schematic representation with a dashed and double-dotted line indicates a body of the hydraulic pump 1. The hydraulic pump 1 comprises an intake port 2 and a delivery port 4.

[0033] A first manifold channel 6 is in hydraulic communication with the intake port 2, and branching off therefrom, in this embodiment, are a first intake line 8, a second intake line 10, and a third intake line 12 in hydraulic communication with the manifold 6.

[0034] Set respectively on the intake lines 8, 10, 12 are a first intake valve 14, a second intake valve 16, and a third intake valve 18. The intake valves 14, 16, 18 are arranged for enabling or disabling a hydraulic connection between the corresponding intake lines 14, 16, 18 and a first cylinder 20, a second cylinder 22, and a third cylinder 24, respectively.

[0035] Movable respectively within the cylinders 20, 22, 24 are a first piston P1, a second piston P2, and a third piston P3. Each of the pistons P1, P2, P3 is actuated with reciprocating motion by means of a mechanism.

[0036] In the particular case analysed here, three cams C1, C2, C3 (with corresponding tappets) have been provided, uniformly offset in an angular direction.

[0037] In alternative embodiments it is possible to provide the pump 1 with pistons actuated by crank mechanisms.

[0038] It should moreover be noted that the embodiment of the pump 1 described herein is, by way of example, of the three-piston type, but the person skilled in the art will appreciate that the present invention applies irrespective of the number of the pistons of the pump 1, which can be different from three, even just one.

[0039] In addition to being in hydraulic communication with the corresponding intake line 8, 10, 12 and the corresponding intake valve 14, 16, 18, each of the cylinders 20, 22, 24 is in hydraulic communication, respectively, with a first delivery valve 26, a second delivery valve 28, and a third delivery valve 30, which are each movable between a closed position and an open position.

[0040] Each delivery valve 26, 28, 30 is hydraulically connected, by means of a respective delivery channel 32, 34, 36, to a second manifold channel 38, which is in turn hydraulically connected to the delivery port 4.

[0041] Each delivery valve 26, 28, 30 is normally in a closed position and is designed to enable, in the open position, a hydraulic connection of the corresponding cylinder 20, 22, 24 associated thereto with the corresponding delivery channel 32, 34, 36 and hence enable a flow of fluid from the corresponding cylinder 20, 22, 24 to the delivery port 4. In a closed position each delivery valve prevents the aforesaid flow of fluid.

[0042] Each of the delivery valves 26, 28, 30 is controlled by means of a number of actuation signals, which result in corresponding actuation forces. In the embodiment described herein said actuation forces are obtained by hydraulic driving lines, or, more briefly, driving lines. In other embodiments, it is possible to obtain the actuation by means of actuators that exert a mechanical (instead of hydraulic) action on a corresponding delivery valve.

[0043] With reference to the embodiment illustrated in Figure 3, each delivery valve 26, 28, 30 is controlled, respectively, by:

- a first hydraulic driving line U1, U2, U3 in fluid communication with the corresponding cylinder 20, 22, 24;
- a second hydraulic driving line CV1, CV2, CV3; and
- a third hydraulic driving line D1, D2, D3 in fluid communication with the corresponding delivery channel 32, 34, 36; it should be noted that said driving line can be optional; consequently, in some embodiments, such as the one described herein, the pump 1 comprises the driving lines D1, D2, D3, whilst in other embodiments only the driving lines U1, U2, U3 and CV1, CV2, CV3 are present.

[0044] Each delivery valve 26, 28, 30 moreover comprises a respective elastic element S1, S2, S3, the action of which is aimed at keeping the corresponding delivery valve in a closed position. The force developed by each elastic element S1, S2, S3 is chosen in such a way as to be substantially negligible with respect to the forces developed by the hydraulic driving lines.

[0045] In the present description, as will be evident to a person skilled in the art, the terms "hydraulic driving line" or "driving line" are used to indicate synthetically a hydraulic line with driving function, i.e., a hydraulic line that in general is able to process a negligible flowrate of fluid and in which the pressure of the fluid within it is exploited as signal of a hydraulic type within a component or a circuit.

[0046] The first and second driving lines U1, U2, U3 and D1, D2, D3 (if present) act on respective surfaces of influence of the corresponding delivery valves 26, 28, 30 to cause opening thereof.

[0047] The hydraulic driving lines CV1, CV2, CV3, like the elastic elements S1, S2, S3 (albeit, as mentioned, to a lesser extent), develop an action aimed at keeping the corresponding delivery valves in the closed position.

[0048] Furthermore, each hydraulic driving line CV1, CV2, CV3 acts on a respective influence area chosen preferentially in such a way that it is substantially equal to the sum of the influence areas on which the remaining hydraulic driving lines of the corresponding delivery valve act, i.e., the driving lines U1, U2, U3 and D1, D2, D3 (if present).

[0049] The hydraulic driving lines CV1, CV2, CV3 branch off from a control channel CV0 hydraulically connected to a control volume CV. The control volume CV is hydraulically connected to the second manifold channel 38 and to the delivery port 4 by means of a hydraulic control line 39, set on which is a choke 40, preferentially with fixed geometry. The choke 40 is set hydraulically upstream of the control volume CV.

[0050] The control volume CV is moreover hydraulically connected to the first manifold channel 6 and to the intake port 2 by means of a control valve 42 set fluid-dynamically downstream thereof in a return channel 43 hydraulically connected to the intake port 2.

[0051] The control valve 42 is functionally a pressure-regulating valve, which in this embodiment is kept in a closed position by means of an elastic element S4. Furthermore, in this embodiment the control valve 42 can be actuated by means of a solenoid 44 operatively connected to an electronic control unit 46.

[0052] In other embodiments it is possible to adopt a hydraulically or mechanically actuated control valve 42.

[0053] There is thus defined a regulation assembly of the pump 1 comprising in this embodiment the hydraulic control line 39, the choke 40, the control volume CV, and the control valve 42 (and, in this embodiment, also the control unit 46), which enables, as will emerge clearly from the ensuing description, a variation in the flowrate sent by the pump 1 to a user that is connected to the delivery port 4.

[0054] Operation of the pump 1 is described in what follows.

[0055] The ensuing description will be developed with particular reference to the application of the hydraulic pump 1 to a fuel-injection system of an internal-combustion engine, in particular as a highpressure pump for an accumulation injection system (the so-called "common rail" injection system). Of course, it is possible to implement what is described herein also in the case of a generic user connected to the delivery port 4.

[0056] The intake port 2 is hydraulically connected to a low-pressure environment LPE that is located fluid-dynamically upstream thereof. For the application considered, the low-pressure environment LPE comprises, for example, a hydraulic inflow line, in which the fuel is moderately pressurized by means of a low-pressure pump that sucks in fuel directly from a tank.

[0057] The delivery port 4 is instead hydraulically connected to a fuel accumulator, commonly referred to as "common rail" (represented schematically and designated by the reference CR), to which one or more fuel injectors (not illustrated) are hydraulically connected. On the modalities with which the injectors operate in a common-rail injection system, reference shall be made to the specific literature in so far as operation of this subsystem is widely known to the person skilled in the branch.

[0058] During operation of the pump 1, the input shaft IS is driven in rotation and actuates each of the pistons P1, P2, P3 with reciprocating motion thanks to the respective cams C1, C2, C3.

[0059] Operation at full flowrate, i.e., without regulation of the flowrate delivered to the common rail CR, will now be described with specific reference to the components associated to the piston P1, without prejudice to the fact that operation of the further pistons and of the homologous components operatively associated thereto is identical.

[0060] Reference is moreover made to the diagrams illustrated in Figures 4 to 8. Each of them describes the evolution of a quantity characteristic of operation of the pump 1 as a function of an angle of rotation of the input shaft IS, designated by CA_{IS}. In greater detail:

- the diagram of Figure 4 comprises three distinct curves describing the plot of the position S of each piston (P1, P2, P3, respectively) expressed as percentage of the total stroke (designated by S_{MAX}); for this reason each curve is labelled with the same reference number as that of the corresponding piston;
- the diagram of Figure 5 comprises three curves that describe an opening DVL of each of the delivery valves 26, 28, 30 (expressed as percentage of a maximum opening value DVL_{MAX}); for this reason, each of the curves is labelled with the same reference number as that used for the corresponding delivery valve;
- the diagram of Figure 6 comprises three curves that describe the plot of a flowrate Q_{DV} (expressed as percentage of a maximum flowrate value $Q_{DV,MAX}$) through the delivery valves 26, 28, 30, respectively; for this reason each curve is labelled with the same reference number as that used for the corresponding delivery valve;
- the diagram of Figure 7 illustrates the plot of the pressure within the common rail CR, designated by p_{CR} (expressed as percentage of the threshold pressure p_{REF}), which is moreover substantially equal to the pressure on the delivery port 4; and
- the diagram of Figure 8 illustrates the plot of an instantaneous flowrate Q_P through the delivery port 4 (i.e., the flowrate delivered to the common rail CR).

[0061] Each piston P1, P2, P3 describes a working cycle comprising five phases, in a way identical to what was described previously for the piston P. Each piston P1, P2, P3 varies, with its reciprocating motion, the volume of the fluid chamber defined with the corresponding cylinder 20, 22, 24 and is movable between a top dead centre TDC and a bottom dead centre BDC.

[0062] The angular range of observation has been chosen in such a way as to have an origin coinciding with the top dead centre TDC of the piston P1 and in such a way as to cover a number of work cycles sufficient for describing a transition from a full-flowrate operating condition to a operating condition in a flowrate-regulation regime.

[0063] Starting from the origin of the axes of the diagram of Figure 4, the piston P1 is in a position corresponding to the top dead centre TDC, where the reflux phase starts, which in these conditions is due to the inertia of the delivery valve that causes delayed closing thereof with respect to the top dead centre TDC. This is followed by the expansion phase, during which the piston

P1 descends to the bottom dead centre BDC, causing a pressure drop within the cylinder 20. When said pressure reaches a value lower than the one inside the low-pressure environment LPE, the intake valve 14 opens, with consequent entry of fluid into the cylinder 20 from the low-pressure environment LPE through the first manifold channel 6 and the first intake channel 8.

[0064] The intake phase terminates substantially when the piston P1 reaches the bottom dead centre BDC. At this point the intake valve 14 closes, isolating the volume 20 from the low-pressure line 8, after which rising of the piston to the top dead centre TDC causes the pressurization (compression phase) of the fluid within the cylinder 20. It should be noted, with reference to Figure 6, that the flowrate through the delivery valve 26 is zero during the compression phase.

[0065] The pressurization of the fluid in the cylinder 20 causes the opening of the delivery valve 26, with the modalities described in what follows (the delivery phase starts).

[0066] The delivery valve 26 is subject to the action of the first driving line U1 and of the second driving line CV1, and of the possible third driving line D1. The driving line U1 conveys, on the corresponding influence area, a pressure signal substantially corresponding to the pressure in the cylinder 20 (designated in what follows by p_{20} - likewise, p_{22} and p_{24} will be used for the cylinders 22, 24); namely, it substantially conveys information regarding a delivery pressure of the cylinder 20 immediately downstream thereof and upstream of the corresponding delivery valve 26.

[0067] The driving line CV1 conveys on the corresponding influence area a pressure signal corresponding to the pressure inside the control volume CV, designated in what follows by p_{CV}.

[0068] The driving line D1, if present, conveys on the corresponding influence area a pressure signal corresponding to the pressure that is exerted on the delivery port 4, which in this case is substantially equal to the pressure within the common rail CR, designated in what follows by p_{CR}. The common rail CR is in fact hydraulically connected to the delivery port 4, which is in turn hydraulically connected to the delivery line 32 by means of the second manifold channel 38.

[0069] It should be noted that the actuation force associated to the hydraulic driving line D1 can be the result, in some embodiments, of the geometry of the delivery valve that offers to the fluid within it an influence area by means of which it is possible to exert a force on the mobile element of the delivery valve itself.

[0070] The control volume CV is in this operating condition a stagnant environment, in so far as there is no passage of flow through the hydraulic control line 39 since the control valve 42 is in a position of complete closing.

[0071] In this way, there is a static transmission of pressure between the common rail CR and the control volume CV, since the choke 40 does not introduce any pressure drop, there not being a significant passage of flow through it and to the control volume CV.

[0072] This means that the pressure p_{CV} is equal to the pressure p_{CR}; consequently, the pressure signal transmitted by the driving line CV1 is identical to the pressure signal transmitted by the driving line D1.

[0073] In effect, in conditions of full-flowrate operation, the delivery valve 26 can be assimilated to a conventional non-return valve; i.e., the delivery valve 26 remains open until the pressure within the cylinder 20 (p_{20}) exceeds the pressure p_{CV} (which is equal, in these conditions, to the pressure p_{CR}) by an amount sufficient to overcome the action of contrast of the elastic element S1, which can be expressed, as is known, by means of an equivalent pressure. In other words, the difference between the pressure p_{20} and the equivalent pressure associated to the elastic element S1 must be equal to or higher than the pressure p_{CR}.

[0074] The operation of the delivery valves 28, 30 is the same; of course the events described above are phase shifted with respect to the ones that involve the delivery valve 26 on account of the timing of the pistons P1, P2, P3 set by the cams C1, C2, C3.

[0075] With reference to Figure 5, it may moreover be noted graphically how the opening of the delivery valves 26, 28, 30 occurs.

[0076] Opening of each delivery valve starts at the end of the compression phase performed by the corresponding piston, and terminates a few instants after the top dead centre TDC, at the end of the reflux phase.

[0077] Once the first working cycles of the angular range considered for this description are completed, the pressure p_{CR} has undergone an increase such that it is very close to the threshold pressure p_{REF} (Figure 7).

[0078] Basically, the excess flowrate delivered by the pump 1 to the common rail CR is not consumed by the injectors connected thereto and results in an accumulation of fluid that raises the level of pressure within the common rail CR and, ultimately, on the delivery port 4.

[0079] The overstepping of the threshold pressure p_{REF} occurs, with reference to Figures 4 to 8 and in particular to Figure 7, at an angle of rotation $CA_{IS,R}$. In the example presented in the diagrams of Figures 4 to 8 the angle $CA_{IS,R}$ falls within the delivery phase of the piston P3.

[0080] The overstepping of the threshold pressure p_{REF} within the common rail CR is detected, in this embodiment, by the electronic control unit 46 by pressure-sensor means of a known type operatively connected thereto and installed on the common rail CR (a schematic functional representation of the operative connection between said sensor means and the control unit 46 is represented by a dashed and dotted line joining the control unit 46 and the common rail CR and designated by the reference PRS). In said condition, the control unit 46 hence controls an actuation of the solenoid 44 that causes the opening of the control valve 42.

[0081] In the above situation, a flow passage is set up in the control line 39 from the delivery port 4 to the control volume CV through the choke 40 and from the control volume CV to the manifold channel 6 through the return line 43. The flow passage through the choke 40 causes a descent of the value of the pressure p_{CV} certainly below the value of the pressure p_{CR} since on account of the flow passage through the choke 40 a pressure drop is induced into the fluid, which upstream of the choke has precisely the pressure p_{CR}.

[0082] The presence of the choke 40 enables said regulation to be obtained by draining flowrates that are very small as compared to the ones processed by the pump, to the advantage of the hydraulic efficiency of the pump itself.

[0083] When the pressure p_{CR} in the common rail CR exceeds the threshold pressure p_{REF}, the control assembly of the pump 1 controls the valve 42 so as to prevent the pressure p_{CV} in the control volume CV from exceeding the pressure p_{REF}. In other words, the control valve 42 is arranged, once the threshold pressure p_{REF} in a user connected to the delivery port 4 of the pump 1 (in this case, the common rail CR) is exceeded, for regulating the pressure in the control volume CV, limiting it to a value at the most equal to p_{REF}.

[0084] In general, any known means designed to regulate the pressure in the control volume CV, preventing the rising thereof beyond a threshold value (p_{REF}, in the case in subject) can be used for implementing the regulation mechanism described herein.

[0085] The delivery valve 30 is always subject to the action of the hydraulic driving lines U3, D3 (if present) and CV3, but now the pressure signal conveyed onto the valve 30 by the driving line CV3 is lower than the one conveyed by the driving lines U3 (and D3, if present), which are always equal to the pressure p_{CR}. The conditions for equilibrium of the delivery valve 3 hence change and are such that it remains open even when the pressure p_{24} drops below the pressure in the delivery environment (p_{CR}). In other words, the delivery valve 30 is kept in the open position also during at least part of the descent phase of the piston P3.

[0086] This causes a reflux of fluid from the delivery port 4 to the cylinder 24, causing, in the last analysis, a reduction of the pressure p_{CR} within the common rail CR on account of the reduction of the accumulation of fluid.

[0087] When the pressure in the common rail CR reaches the level of pressure existing in the control volume CV (i.e., p_{REF}) the delivery valve 30 closes, and in the corresponding cylinder 24 the phase of expansion of the fluid contained therein starts, followed by the intake phase.

[0088] Hence, the control assembly of the pump 1 is substantially arranged for keeping the delivery valve 30 (and, with the due timing, the delivery valves 26, 28) in the open position also following upon the delivery phase, i.e., during at least part of the descent phase of the piston P3, by regulation of the pressure p_{CV} (associated to the driving lines CV1, CV2, CV3), in particular by limitation of the pressure p_{CV} to the value of the threshold pressure p_{REF} if said threshold pressure p_{REF} is exceeded in a user (in this case, the common rail CR) hydraulically connected to the delivery port 4.

[0089] In brief, the regulation system proposed herein is such that in conditions of full-flowrate operation each delivery valve remains open for a little less than half of the working cycle, basically corresponding to the delivery phase (the reflux phase is substantially negligible). In regulation conditions, instead, the instant of closing is decidedly delayed, and this results in an additional angular range of opening, where there is witnessed an increase in the reflux of flowrate through the delivery valve 30 (points of the curve with negative ordinates), which indicates the occurrence of a considerable reflux of fuel back into the cylinder 24.

[0090] During this phase, as described previously, the fluid returns to the pump the excess work of compression performed in the preceding delivery phase, thus guaranteeing a good level of energy efficiency of the entire system.

[0091] Of course, with due timing, the same applies to the delivery valves 26 and 28. Basically, the regulation of the flowrate sent by the pump 1 is obtained by reflux of fluid (in particular fuel) to the delivery of each piston.

[0092] With reference to Figure 7, it should moreover be noted that the regulation strategy described herein functions on the condition of tolerating an overpressure in the common rail CR (or in general in a user connected to the delivery port 4). Simplifying the treatment (it is important for the purposes of this description to understand what are the physical quantities involved), the maximum overpressure that is to be withstood is equal to:

$$\Delta p = \frac{V_{cyl}}{V_{CR}} \cdot E$$

where:

Δp is the overpressure in the common rail CR

V_{cyl} is the unit displacement of the pump

V_{rail} is the internal volume of the common rail CR E is the elasticity modulus of the fluid

[0093] Finally, with reference to Figure 8, illustrated therein is the curve of instantaneous flowrate Q_P delivered by the pump 1 through the delivery port 4 plotted as a function of the angle CA_{IS}. The values of flowrate Q_P are expressed as percentage of a maximum flow-rate value designated by $Q_{P,MAX}$. There should be noted, as on the other hand has already been pointed out as regards Figure 6, a marked increase in the angular ranges (referred to the angle $CA_{IS,R}$) , where the values of flowrate have a negative sign, i.e., where a reversal of flow and a return of fluid to the cylinder (or cylinders) operatively associated to the delivery valve in regulation occur.

[0094] The pump 1 according to the present invention hence presents a series of considerable advantages over pumps of a known type.

[0095] In the first place, the pump 1 enables the elimination of the need for expensive regulations by lamination at the delivery or for dangerous regulations by lamination at the intake, which are the source of problems connected to cavitation in the fluid induced thereby.

[0096] Furthermore, the control assembly of the pump 1 is easy to manage and moreover enables regulation of the threshold pressure within the common rail CR in a simple way by acting on the calibration/threshold of opening of the control valve 42.

[0097] The flowrate consumed by the control system is moreover very small and substantially negligible as compared to the one typically processed by the pump 1, which minimizes the impact of the operation of the control system on the operation of the pump 1 itself.

[0098] Of course, the details of construction and the embodiments may vary widely with respect to what is described and illustrated herein by way of non-limiting example, as defined by the annexed claims.

Claims

1. A hydraulic pump (HP; 1), in particular a fuel pump, comprising:

- an intake port (IP; 2),
- a delivery port (DP; 4), arranged for hydraulic connection to a user (U; CR),
- at least one cylinder (CY; 20, 22, 24), movable within which is a corresponding piston (P; P1, P2, P3), said piston having, during the operation of the pump (HP, 1), a reciprocating motion between a top dead centre (TDC) and a bottom dead centre (BDC),
- a delivery valve (DV; 26, 28, 30) for each cylinder (CY; 20, 22, 24) hydraulically connected thereto, movable between a closed position and an open position, wherein in said open position said delivery valve (DV; 26, 28, 30) is arranged for enabling a flow of fluid between said cylinder (CY; 20, 22, 24) and said delivery port (DP; 4), and wherein in said closed position said delivery valve (DV; 26, 28, 30) is arranged for blocking a flow of fluid between said cylinder (CY; 20, 22, 24) and said delivery port (DP; 4),

the pump (HP, 1) being **characterized in that** it comprises a regulation assembly (R; 39, 40, 42, 43, 44, 46) which, when a threshold pressure (p_{REF}) in a user (U; CR) hydraulically connected to said delivery port (DP; 4) is exceeded, is arranged for keeping said delivery valve (DV, 26, 28, 30) in said open position during at least part of a movement of said piston from said top dead centre (TDC) to said bottom dead centre (BDC) and enabling a reflux of fluid from said delivery port (4, DP) to said cylinder (20, 22, 24).

2. The hydraulic pump (1) according to Claim 1, **characterized in that** said delivery valve (26, 28, 30) is controlled by means of hydraulic driving lines (U1, U2, U3; D1, D2, D3; CV1, CV2, CV3).

3. The hydraulic pump (1) according to Claim 2, **characterized in that** said regulation assembly (R; 39, 40, 42, 43, 44, 46) comprises:

- a control volume (CV) in fluid communication with said delivery port (4) by means of a hydraulic control line (39), set on which is a choke (40); and
- a control valve (42), which is arranged for regulating a pressure (p_{CV}) in said control volume (CV) and can be actuated for enabling a fluid communication between said control volume (CV) and said intake port (2),

wherein said choke (40) is set hydraulically upstream of said control volume (CV).

4. The hydraulic pump (1) according to Claim 3, **characterized in that** said hydraulic driving lines comprise:

- a first hydraulic driving line (U1, U2, U3) that conveys on said delivery valve (26, 28, 30) a pressure signal substantially corresponding to a pressure (p_{20}, p_{22}, p_{24}) in said at least one cylinder (20, 22, 24),
- a second hydraulic driving line (CV1, CV2, CV3) that conveys on said delivery valve (26, 28, 30) a pressure signal substantially corresponding to a pressure (p_{CV}) inside said control volume (CV).

5. The hydraulic pump (1) according to Claim 4, **characterized in that** said control valve (42) is arranged for limiting the pressure (p_{CV}) in said control volume (CV) substantially to the value of said threshold pressure (p_{REF}) upon overstepping of said threshold pressure (p_{REF}) in a user (U; CR) hydraulically connected to said delivery port (DP; 4).

6. The hydraulic pump (1) according to any of Claims 2 to 5, **characterized in that** said delivery valve (26, 28, 30) comprises a third hydraulic driving line (D1, D2, D3) that conveys on said delivery valve (26, 28, 30) a pressure signal substantially corresponding to a pressure (p_{CR}) on said delivery port (4), wherein said third hydraulic driving line (D1, D2, D3) is arranged for bringing said delivery valve (26, 28, 30) into said open position.

7. The hydraulic pump (1) according to Claim 6, **characterized in that** said second hydraulic driving line (CV1, CV2, CV3) acts on an influence area equal to the sum of respective influence areas on which said first (U1, U2, U3) and third (D1, D2, D3) hydraulic driving lines act.

8. The hydraulic pump (1) according to Claim 3, **characterized in that** said control valve (42) is controlled by means of an electronic control unit (46) arranged for co-operating with sensor means for detecting the pressure in a user (U; CR) connected to said delivery port (4).

9. The hydraulic pump (1) according to any one of the preceding claims, **characterized in that** it comprises a first piston (P1), a second piston (P2), and a third piston (P3), which are operatively associated, respectively, to a first delivery valve (26), a second delivery valve (28), and a third delivery valve (30), wherein said first, second and third pistons (P1, P2, P3) are driven with reciprocating motion through an input shaft (IS) of said hydraulic pump (1).

Amended claims in accordance with Rule 137(2) EPC.

1. A hydraulic pump (HP; 1), in particular a fuel pump, comprising:

- an intake port (IP; 2),
- a delivery port (DP; 4), arranged for hydraulic connection to a user (U; CR),
- at least one cylinder (CY; 20, 22, 24), movable within which is a corresponding piston (P; P1, P2, P3), said piston having, during the operation of the pump (HP, 1), a reciprocating motion between a top dead centre (TDC) and a bottom dead centre (BDC),
- a delivery valve (DV; 26, 28, 30) for each cylinder (CY; 20, 22, 24) hydraulically connected thereto, movable between a closed position and an open position, wherein in said open position said delivery valve (DV; 26, 28, 30) is arranged for enabling a flow of fluid between said cylinder (CY; 20, 22, 24) and said delivery port (DP; 4), and wherein in said closed position said delivery valve (DV; 26, 28, 30) is arranged for blocking a flow of fluid between said cylinder (CY; 20, 22, 24) and said delivery port (DP; 4), wherein during the operation of the pump (HP, 1) said delivery valve is in said open position during a portion of the movement of said piston from the bottom dead centre (BDC) to the top dead centre (TDC) and closes after the top dead centre (TDC) of the reciprocating motion of the piston (P; P1, P2, P3),the pump (HP, 1) being **characterized in that** it comprises a regulation assembly (R; 39, 40, 42, 43, 44, 46) which, when a threshold pressure (p_{REF}) in a user (U; CR) hydraulically connected to said delivery port (DP; 4) is exceeded, is arranged for keeping said delivery valve (DV, 26, 28, 30) in said open position during at least part of a movement of said piston from said top dead centre (TDC) to said bottom dead centre (BDC), thereby delaying the closure of the delivery valve (DV, 26, 28, 30) and enabling a reflux of fluid from said delivery port (4, DP) to said cylinder (20, 22, 24).

2. The hydraulic pump (1) according to Claim 1, **characterized in that** said delivery valve (26, 28, 30) is controlled by means of hydraulic driving lines (U1, U2, U3; D1, D2, D3; CV1, CV2, CV3) .

3. The hydraulic pump (1) according to Claim 2, **characterized in that** said regulation assembly (R; 39, 40, 42, 43, 44, 46) comprises:

- a control volume (CV) in fluid communication with said delivery port (4) by means of a hydraulic control line (39), set on which is a choke (40); and
- a control valve (42), which is arranged for regulating a pressure (p_{CV}) in said control volume (CV) and can be actuated for enabling a fluid communication between said control volume (CV) and said intake port (2) ,

wherein said choke (40) is set hydraulically upstream of said control volume (CV).

4. The hydraulic pump (1) according to Claim 3, **characterized in that** said hydraulic driving lines comprise:

- a first hydraulic driving line (U1, U2, U3) that conveys on said delivery valve (26, 28, 30) a pressure signal substantially corresponding to a pressure (p_{20}, p_{22}, p_{24}) in said at least one cylinder (20, 22, 24) ,
- a second hydraulic driving line (CV1, CV2, CV3) that conveys on said delivery valve (26, 28, 30) a pressure signal substantially corresponding to a pressure (p_{CV}) inside said control volume (CV).

5. The hydraulic pump (1) according to Claim 4, **characterized in that** said control valve (42) is arranged for limiting the pressure (p_{CV}) in said control volume (CV) substantially to the value of said threshold pressure ($_{PREF}$) upon overstepping of said threshold pressure (p_{REF}) in a user (U; CR) hydraulically connected to said delivery port (DP; 4).

6. The hydraulic pump (1) according to any of Claims 2 to 5, **characterized in that** said delivery valve (26, 28, 30) comprises a third hydraulic driving line (D1, D2, D3) that conveys on said delivery valve (26, 28,

30) a pressure signal substantially corresponding to a pressure (p_{CR}) on said delivery port (4), wherein said third hydraulic driving line (D1, D2, D3) is arranged for bringing said delivery valve (26, 28, 30) into said open position.

7. The hydraulic pump (1) according to Claim 6, **characterized in that** said second hydraulic driving line (CV1, CV2, CV3) acts on an influence area equal to the sum of respective influence areas on which said first (U1, U2, U3) and third (D1, D2, D3) hydraulic driving lines act.

8. The hydraulic pump (1) according to Claim 3, **characterized in that** said control valve (42) is controlled by means of an electronic control unit (46) arranged for co-operating with sensor means for detecting the pressure in a user (U; CR) connected to said delivery port (4).

9. The hydraulic pump (1) according to any one of the preceding claims, **characterized in that** it comprises a first piston (P1), a second piston (P2), and a third piston (P3), which are operatively associated, respectively, to a first delivery valve (26), a second delivery valve (28), and a third delivery valve (30), wherein said first, second and third pistons (P1, P2, P3) are driven with reciprocating motion through an input shaft (IS) of said hydraulic pump (1).

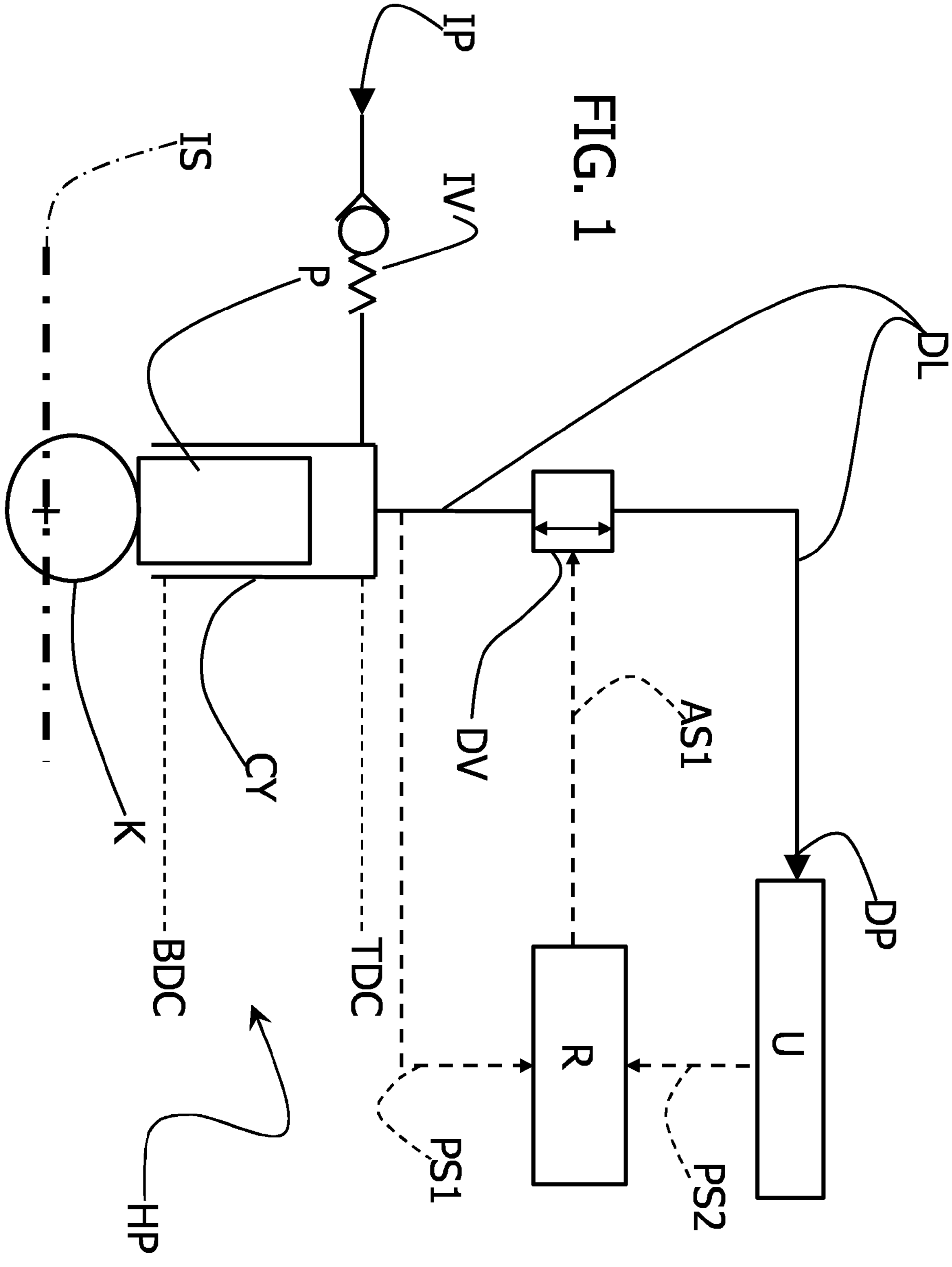


FIG. 1

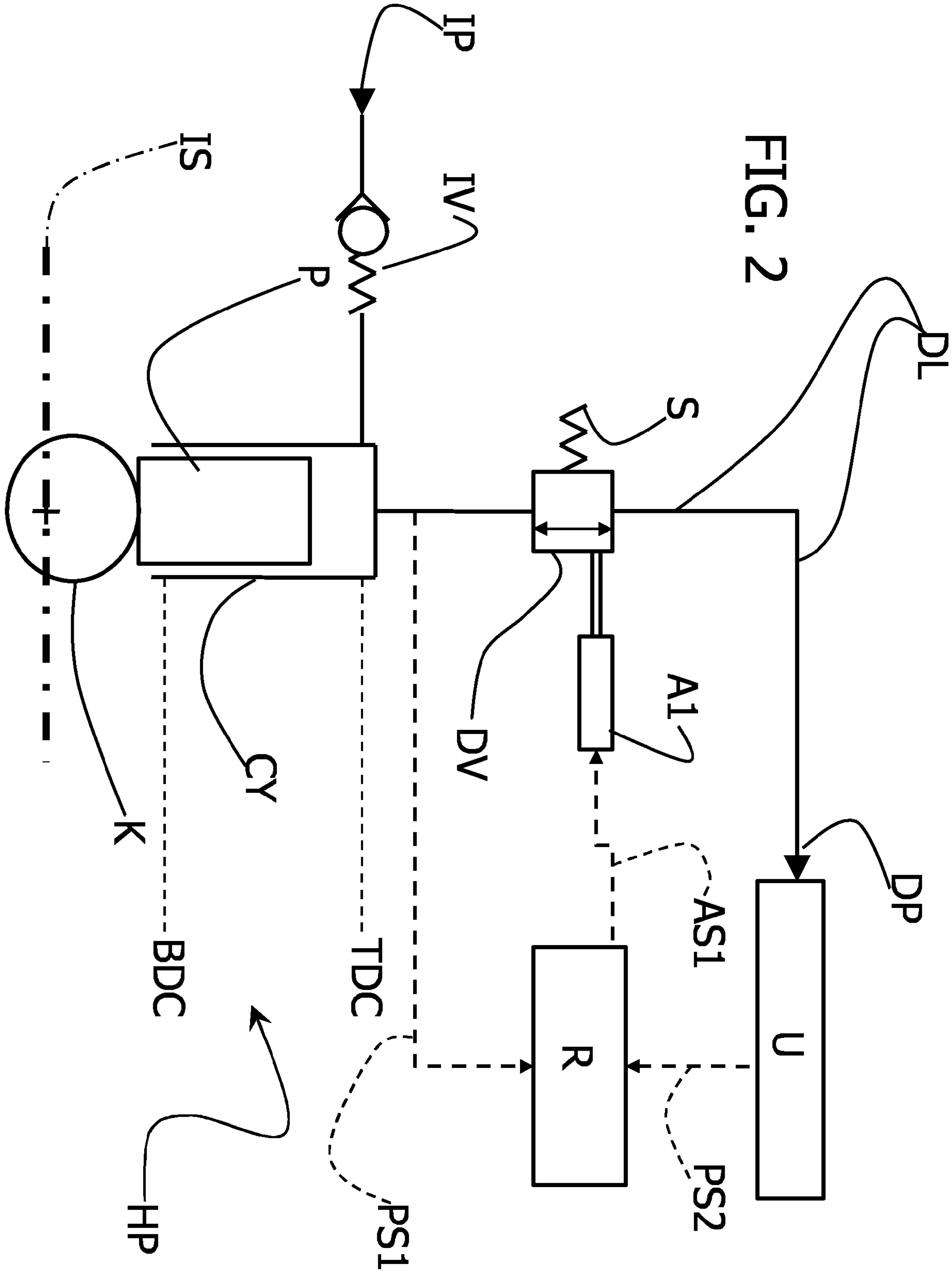


FIG. 2

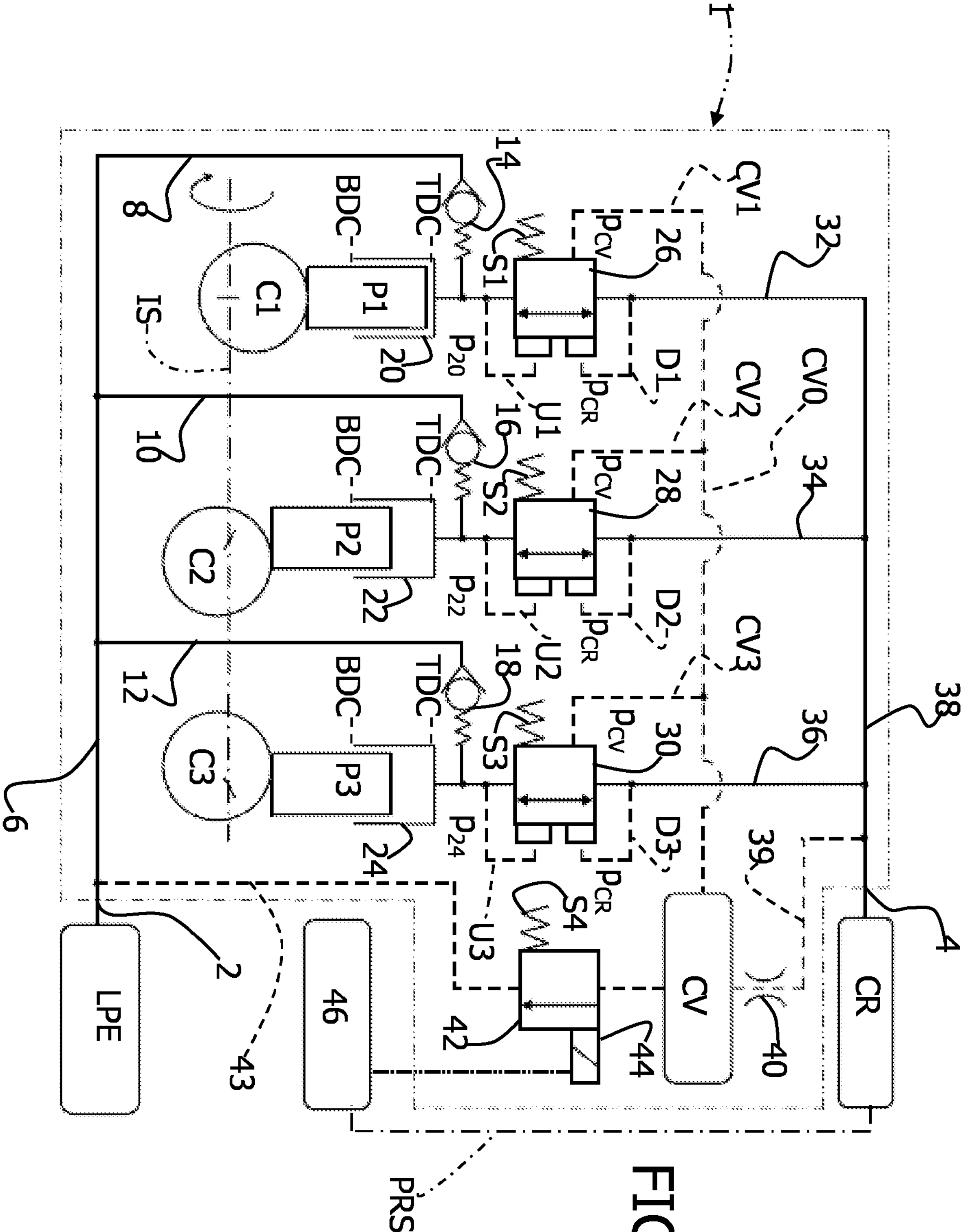

1
CV1
32
26
14
8
TDC
BDC
S1
C1
P1
IS
20
P20
U1
16
D1
CV2
CV0
10
TDC
BDC
S2
28
34
C2
P2
22
P22
D2
U2
CV3
18
12
TDC
BDC
S3
30
36
38
C3
P3
24
P24
D3
39
6
S4
U3
CV
CR
4
2
LPE
46
42
44
40
43
PRS

FIG.3

FIG. 4

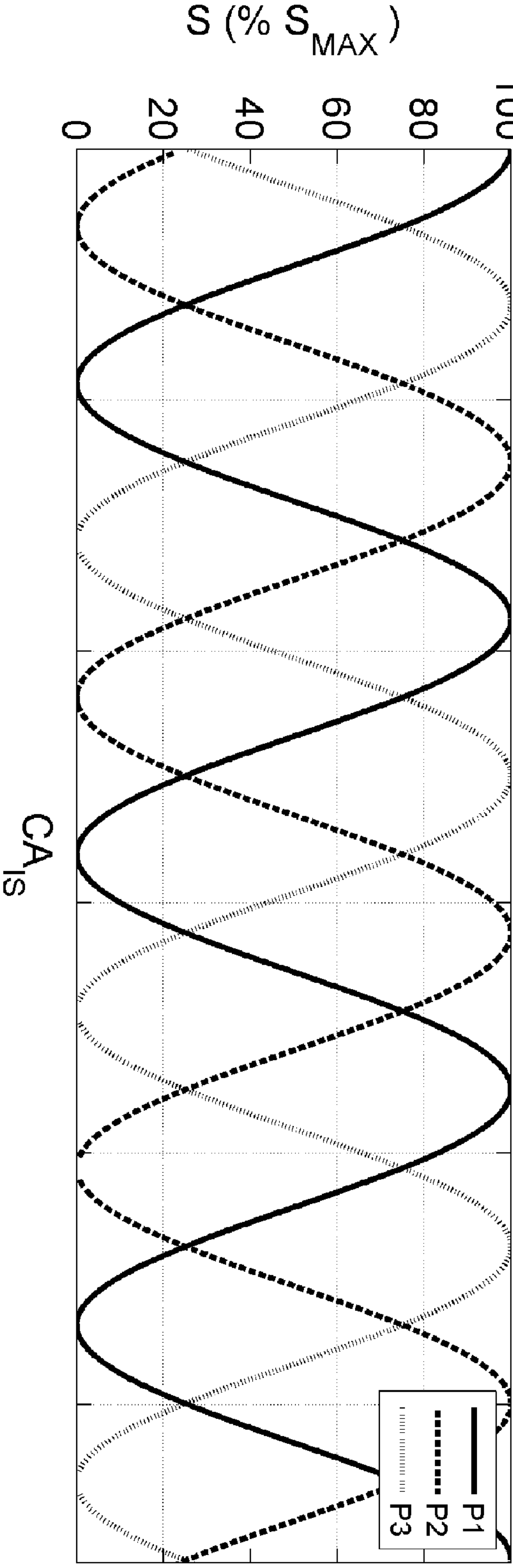

S (% S_MAX)
0
20
40
60
80
100
CA_IS
P1
P2
P3

FIG. 5

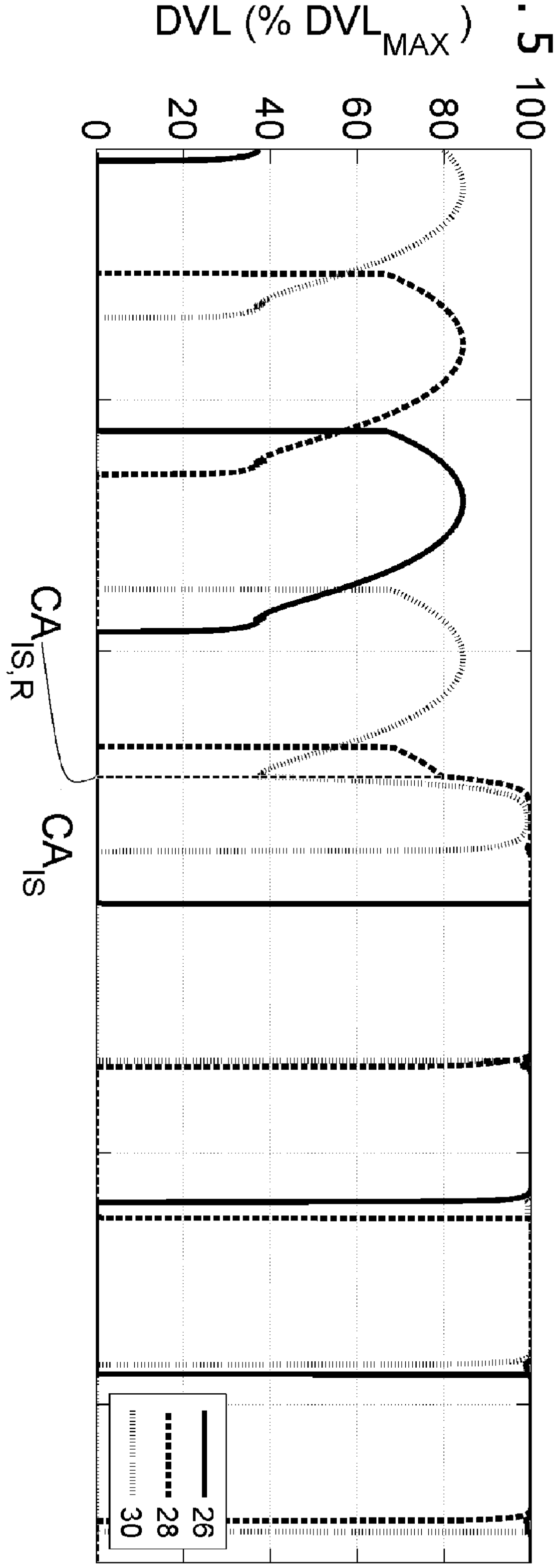

DVL (% DVL_MAX)
0
20
40
60
80
100
CA_IS,R
CA_IS
26
28
30

FIG. 6

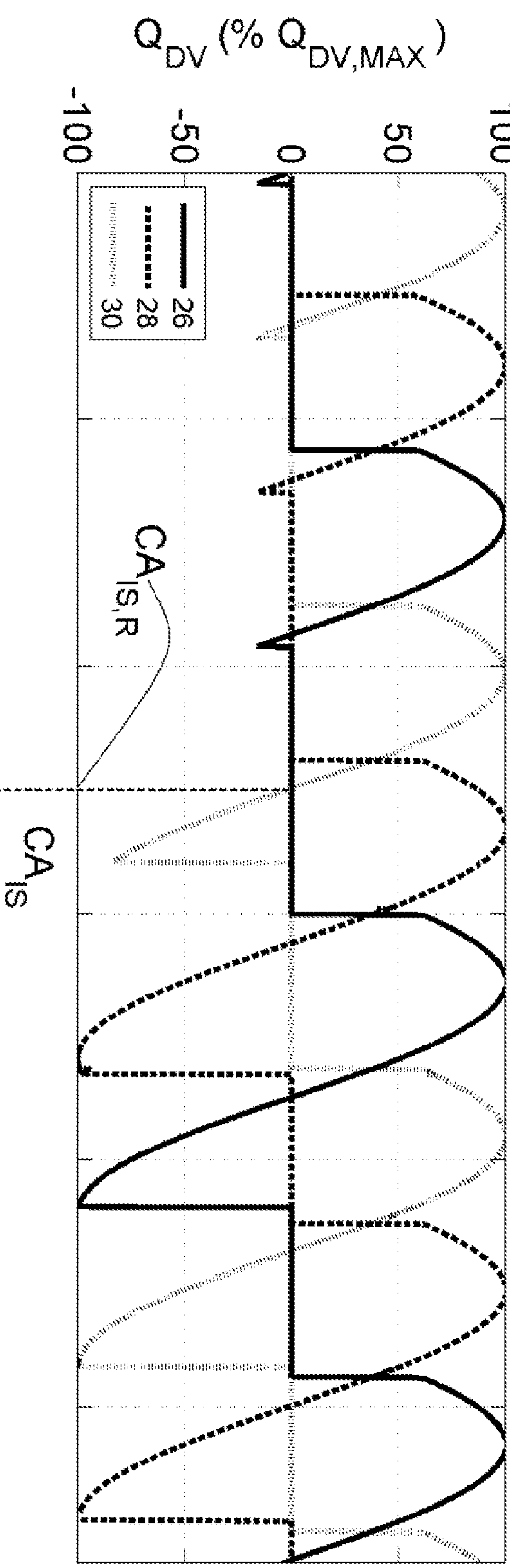
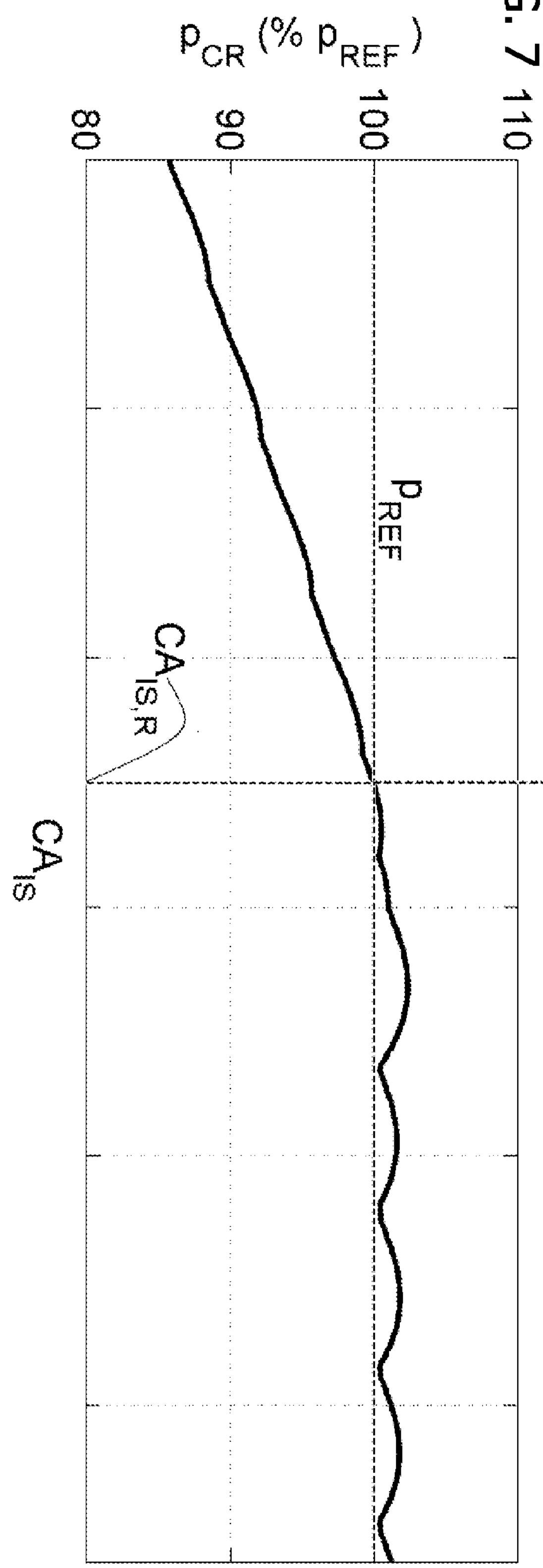

Q_{DV} (% $Q_{DV,MAX}$)
100
50
0
-50
-100
26
28
30
$CA_{IS,R}$
CA_{IS}

FIG. 7


p_{CR} (% p_{REF})
110
100
90
80
p_{REF}
$CA_{IS,R}$
CA_{IS}

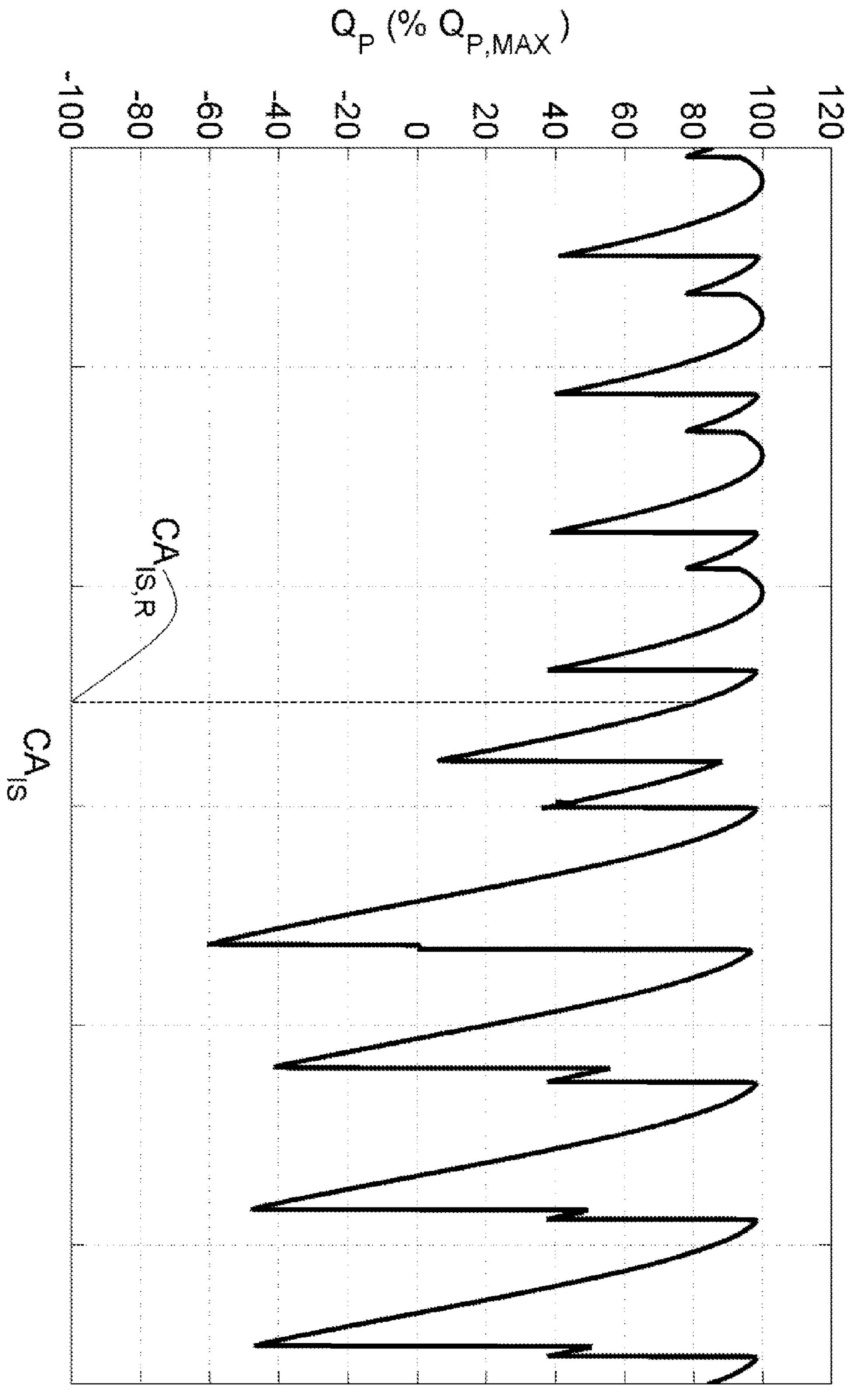


FIG. 8

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EUROPEAN SEARCH REPORT

Application Number
EP 11 15 7051

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F02M
F04B

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	1 September 2011	Schmitter, Thierry

CATEGORY OF CITED DOCUMENTS

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