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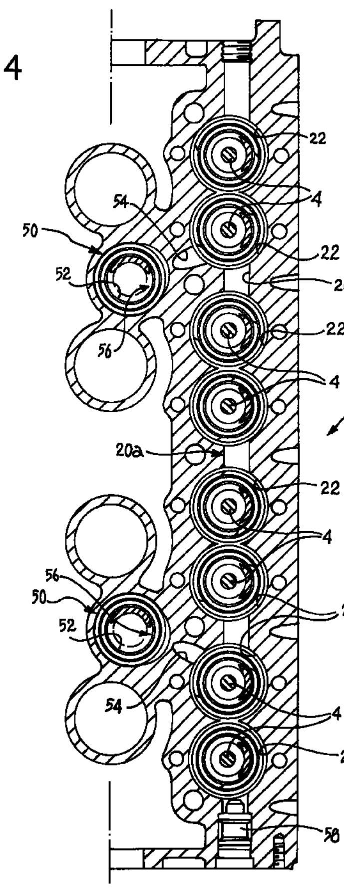
(54) A valve control system for an internal combustion engine

(57) A cam (5) having an asymmetric profile (25, 27, 29, 31) controls the movement of a valve (3) of an internal combustion engine and cooperates with a tappet (6) whose active surface (15) in contact with the cam (5) has a convex shape. The profile of the cam (5) is such that, when the engine speed exceeds a threshold value, the closure of the valve member (3) occurs without contact between the valve member (3) and the cam (5), in a period of time which is substantially fixed as the engine speed varies, whereby the closure angle of the cam (5) increases as the speed of rotation of the cam increases. One particular geometry of the cam profile (5), specifically the shape of its tip (27) and its ratio to the shape of the root (25), ensures a broad automatic variation of the closure angle of the valve member (3) and at the same time allows the cam (5) to turn in the opposite sense from that expected in normal engine running.

FIG. 4



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Description

The present invention generally concerns valve control systems for internal combustion engines.

More specifically, it concerns a control system of the type described in European Patent Application No. EP-A-0 574 867 which is considered to be an integral part of the present description.

In particular, the present invention concerns a valve control system for an internal combustion engine, of the type including:

- a valve movable between a closed position and an open position of a duct;
- resilient biasing means for biasing the valve towards its closed position;
- valve control means for driving the valve to effect cyclic movements to the open position against the action of the resilient means, in which the control means include a rotatable cam with an asymmetric profile which includes a first eccentric profile portion for controlling the movement of the valve towards its open position and a second eccentric profile portion, steeper than the first eccentric profile portion, for controlling the movement of the valve towards its closed position, and a root portion and a tip portion extending between the first and second eccentric profile portions and connected to the first and second eccentric profile portions on opposite sides of the axis of rotation of the cam, the root and tip portions both having respective constant radii of curvature, and a tappet operatively interposed between the cam and the valve and provided with an active surface facing the cam, the active surface being engageable by the cam profile at least during the opening movement of the valve in such a way that, when the rotational speed of the cam exceeds a threshold value, the active surface of the tappet loses contact with the cam profile, thereby uncoupling the closing movement of the valve from the rotation of the cam so that the closing movement occurs in a substantially fixed time period whatever the rotational speed of the cam, the active surface of the tappet having a flat portion which is substantially orthogonal to the direction of movement of the tappet and a curved portion connected to the flat portion in such a way as to form a convex zone facing the cam.

In a system of the type defined above a serious problem can, however, arise from the fact that the rotation of the camshaft which controls the timing may be "irreversible". More specifically, due to the geometry of the cam and of the tappet which cooperates with it, it is possible for a rotation of the cam in the opposite sense from its rotation in normal engine running conditions to lead to the striking of the cam, specifically its tip portion, against the tappet.

Such an anomalous running condition can, for example, occur when a motor vehicle provided with the device in its engine is on an upwardly inclined stretch of road with its engine off and a high gear is engaged. In this case, the engine's components, in particular the crankshaft and the camshaft rotatably coupled to it, can be rotated in the opposite sense from that expected under normal running conditions, this being made possible if the slope of the road is such as to permit the resistance caused by the friction within the engine and the reduced engine braking effect to be overcome because of the high gear ratio.

In such conditions the cams keyed to the camshaft, in turning in the opposite sense from normal, cooperate with the active surfaces of their respective tappets but move in the opposite way from usual. This situation can become critical if the cams have asymmetric profiles which cooperate with tappets other than the type with plate-like active surfaces, as in the present invention. In these circumstances, it is in fact possible for the cam tips to strike against the convex profiles of the tappets, consequently causing damage to the engine, in particular the breaking or stripping of the timing belt which transmits rotational drive to the camshaft from the crankshaft.

To overcome this problem, the present invention provides a system of the type indicated above characterised in that the ratio of the radius of curvature of the tip portion to the radius of curvature of the root portion is between approximately 0.35 and 0.40.

By virtue of the above characteristic, the system according to the invention is provided with cams whose asymmetric profiles guarantee a very broad automatic variation in the valve closure angle in order to maximise the filling of the cylinders at all engine speeds, achieving a high volumetric efficiency and optimum specific horsepower at every engine speed, and which further ensure that the cams can rotate in the opposite sense from their normal sense of rotation, that is, ensuring the "reversibility" of the rotation of the camshaft, thus avoiding any risk of damage to the engine parts as a result of reverse rotation of the camshaft.

Further characteristics and advantages of the invention will become clearer from the following detailed description, made with reference to the appended drawings, given purely by way of non-limitative example, in which:

Figure 1 is a sectioned side view of part of the system according to the invention in which the valve is closed;

Figure 2 is an enlarged view similar to Figure 1 with the valve open;

Figure 3 is a sectioned side view of part of a cylinder head provided with a system according to the invention; and

Figure 4 is a sectioned view taken on the line IV-IV of Figure 3.

With reference to the drawings, reference numeral 1 indicates a cylinder head of an internal combustion engine (illustrated only partially in the drawings).

Reference number 2 indicates an inlet duct associated with one of the cylinders, the outlet of which is controlled by a valve member 3, the head of which, when in its closed position, abuts a seat 3a. Although reference is made specifically in the course of the present description to the inlet valves, the invention can equally be applied to the control of either inlet valves or exhaust valves.

Each valve member 3 has a stem 4 which is displaceable axially along an axis D to control the opening and closing of the duct 2 and which is guided by a guide sleeve 4a, of known type, associated with the cylinder head 1. The movement of each valve member 3 is driven cyclically by an associated cam 5 mounted on the camshaft of the internal combustion engine and which rotates about the axis F of the camshaft in an anti-clockwise sense (with reference to the drawings), as indicated by the arrow E, during normal running of the engine.

A tappet 6, which has a substantially cup-shaped body, is interposed between the cam 5 and that end of the stem 4 closer to the cam 5. The tappet 6 is slidable axially in a bush 8 co-axial with the axis D and rigidly fixed to the cylinder head 1.

A collet element 10, of known type, is fixed axially to the valve stem 4 and is engaged by a pair of concentric helical springs 11 and 12 whose function is to bias the valve member 3 towards its position closing the duct 2.

The tappet 6 includes a head 14 facing the cam 5, which has an active surface 15 for cooperating with the profile of the cam 5. The head 14, rigid with the tappet 6, is prevented from rotating in the bush 8 by means of a "stirrup-like" guide element 16 having, in plan, a substantially arcuate shape corresponding to the shape of the adjacent edge of the head 14, and being rigidly fixed to the cylinder head 1 by means of a screw 17 which also has the function of retaining the bush 8.

A pad 18 is interposed between the end of the stem 4 facing the cam 5 and the head 14 of the tappet 6, the replacement of which allows the relative positions of the cam 5 and the tappet 6 (tappet play) to be finely adjusted. To this end the tappet 6 has an aperture 18a at the base of the head 14, which allows the extraction and insertion of the pad 18 with an appropriate tool (not illustrated in the drawings). Naturally, as an alternative to the adjustment pads 18, known hydraulic devices may be interposed between the stem 4 of every valve member and the respective tappet 6 to compensate the tappet play automatically.

In the cylinder head 1 is a duct 20 which is supplied with oil under pressure for lubricating the engine. This duct 20 is part of an hydraulic circuit 20a of the cylinder block and communicates with a peripheral annular chamber 22 which forms part of the same circuit and surrounds the bush 8, the function of which will become clearer in the following description.

The cam 5 has an asymmetric profile which includes a root portion 25 having a constant radius of curvature R_o and, on the opposite side, a tip portion 27, also having a constant radius of curvature R_1, the centre of which is spaced from the axis of rotation F of the camshaft.

Between the root portion 25 and the tip portion 27 of the cam 5 is a "less steep" profile portion 29 which controls the opening of the valve and a "steeper" profile portion 31 which controls the closure of the valve. The expression "steep profile" used in the present description and in the following claims is intended to indicate a profile which, in a system of polar coordinates, displays a greater variation of the radial coordinate for a given increment in the angular coordinate. Because of the shape of the profiles 29 and 31, the opening phase of the valve, that is, the phase in which the valve moves from zero lift to maximum lift, involves an angular movement of the cam which is greater than that necessary to bring the valve back to its closed position.

In particular a substantial part of the profile portion 31 is a straight line tangential to a circle having a radius of curvature R_2 less than the radius of curvature R_0 of the root portion 25 and also having its centre on F. This straight line is therefore connected at one end to the root portion 25 and at the other end to the tip portion 27.

The cam 5 cooperates with the active surface 15 of the tappet 6, which includes a first flat portion 33 substantially perpendicular to the axis D along which the valve stem 4 moves. The flat profile 33 is connected to a curved profile 35 to form a convex portion facing the cam 5, whose constant radius of curvature is indicated R_3.

The radii of curvature R_0, R_1, R_2, and R_3 are preferably linked by numeric relationships which allow the control system to be so dimensioned as to optimise its operation:

- the ratio of the radius of curvature R_1 of the tip portion 27 to the radius of curvature R_0 of the root portion 25 is between 0.35 and 0.40, and is preferably between 0.36 and 0.398;
- the ratio of the radius of curvature R_2 of the circle touching the "steeper" straight profile of the profile portion 31 of the cam 5 to the radius of curvature R_0 of the root portion 25 is between 0.5 and 0.8;
- the ratio of the radius of curvature R_3 of the curved portion 35 of the active surface 15 of the tappet 6 to the radius R_1 of the tip portion of the cam 5 profile is between 0.8 and 1.2.

The control system described above is designed in such a way that, when the engine speed exceeds a critical threshold value, the cam 5 loses contact with the tappet 6 in the valve closure phase whilst, when the rotational speed of the engine is less than or equal to the above threshold value, the tappet 6 and, more particularly its active surface 15, stays in contact with the profile of the cam 5.

When the engine speed exceeds the above threshold, the active surface 15 of the tappet 6 loses contact with the profile portion 31 of the cam 5 during the valve closure phase, thereby causing "ballistic" operation, due to the "steepness" of the profile portion 31 and the high speed of the cam, which is naturally a direct function of the engine speed since the camshaft is driven by the crank shaft. In these conditions, the way in which the valve closes will be determined solely by the mass of the moving parts, the thrust of the springs 11 and 12 and any inertial and damping actions to which the valve member 3 is subject.

Since all of the above factors are independent of the engine speed, it follows that, from the moment that contact between the tappet and the cam is lost, the return of the valve to its closed position will also be independent of the engine speed and will therefore occur in a substantially constant time period whatever the engine speed above the said threshold.

The control system according to the invention therefore allows the efficient and simple resolution of the need both for a small valve closure angle at low engine speeds and for a large valve closure angle at high engine speeds with an automatic increase in the closure angle as the engine speed increases through a threshold value.

In particular, the range of values indicated above for the ratio of the radius of curvature R_1 to the radius of curvature R_0 is that which provides, as simply as possible, the best compromise between the need for a very broad automatic variation in the valve closure angle, thus maximising the filling of the cylinders at all engine speeds so as to optimise the specific horsepower of the engine at all running speeds, and the need to ensure the "reversibility" of the rotation of the camshaft, similar to that normally seen in engines provided with conventional cams with symmetric profiles and tappets with plate-like active surfaces.

The system according to the invention is further provided with a hydraulic braking device, the function of which is to slow the movement of the valve member in the last stage of its closure travel so as to avoid sharp contact between the tappet 6 and the surface of the cam 5.

This hydraulic braking device includes a variable-volume chamber 40 which extends between the tappet 6 and the bush 8 and is bounded axially by a greater-diameter portion 6a of the tappet 6 and, at the opposite end, by a smaller-diameter portion 8a of the bush 8. The volume of the chamber 40 varies in dependence on the relative positions of the tappet 6 and the bush 8.

The chamber 40 has a base area shaped as a circular ring concentric with the axis D and bounded internally by a circle of radius R_4 and externally by a circle of radius R_5.

Two sets of radial passages are formed in the bush 8 in correspondence with the peripheral annular chamber 22 and put the chamber 22, supplied with oil under pressure through the duct 20, in communication with the variable volume chamber 40.

More particularly, these are circular-section outlet passages 46 and cylindrical hydraulic brake holes 48, the diameter of the holes 48 being appreciably less than the diameter of the passages 46. The passages 46 are relatively short in comparison with their diameters, that is, they have the characteristics of holes in thin walls, whereby fluid flowing through them produces a damping effect which is independent of the viscosity of the fluid used. Otherwise such a damping effect would be affected by the temperature of the oil flowing through the passages 46, which varies with the engine temperature.

The flowing of fluid through the ducts 48 produces a hydraulic braking action to slow the travel of the valve member in the last part of its closure phase so as to avoid the sudden contact between the tappet 6 and the surface of the cam 5 which can occur in "ballistic" operation. The shape and the number of the holes 48 of the hydraulic brake device and of the passages 46 may differ from that indicated by way of example in the present description and in the accompanying drawings.

The total cross-sectional area of the holes 48 and the total cross-sectional area of the passages 46 are preferably related numerically to the radii R_4 and R_5 to allow the hydraulic braking device to be dimensioned in such a way as to optimise its functional characteristics; in particular:

- the ratio of the total cross-sectional area of the holes 48 to the area of the base surface of the variable-volume chamber 40 is between 0.002 and 0.016;
- the ratio of the total cross-sectional area of the passages 46 to the base area of the variable-volume chamber 40 is between 0.3 and 1.

Figures 3 and 4 illustrate the cylinder head 1 of a multi-cylinder engine provided with the system according to the invention, the head including a plurality of valve pairs of valve members 3, respectively one pair for each engine cylinder, with associated tappets 6 and bushes 8, each of which is surrounded by an associated peripheral chamber 22. All of the chambers 22 are supplied with pressurised engine lubricating oil through the circuit 20a which includes a single duct 20 which is common to all of the inlet valves.

The circuit 20a includes a pair of accumulators 50, each being formed by a cylindrical chamber 52, which are connected to the duct 20 by means of service lines 54. Each chamber 52 is provided with a damper device constituted by a resiliently deformable member 56, of known type, which occupies a volume within the respective chamber 52 which varies in dependence on the variation in the oil pressure in the circuit 20a as a consequence of the cyclical variation in the volume of the variable-volume chambers 40 associated with each of the valve members 3.

A non-return valve 58 is interposed between the circuit 20a and the source of pressurised engine lubricating oil. In this way, in ideal functioning without oil leakage, the hydraulic circuit 20a, including the duct 20, the chambers 22 and 40, the lines 54 and the accumulators 50, forms a closed hydraulic circuit which is independent of the engine lubrication circuit. When there is fluid loss from the circuit 20a, the normal oil level is restored by the opening of the valve 58 which allows new oil taken from the engine lubrication circuit to be supplied to the duct 20.

During operation, when the valve member 3 moves from its closed position (Figure 1) to its open position (Figure 2) relative to the duct 20, pressurised fluid flows from the peripheral annular chamber 22 associated with each valve member to the duct 20 leading to the variable volume chamber 40 as a result of the variation in volume of the chamber 40 caused by the movement of the tappet 6, and thus its larger portion 6a, relative to the bush 8 and, in particular, its portion 8a. In these conditions, the pressurised oil can flow through the passages 46 and 48 and refill the chamber 40. When the valve member returns towards its closed position, the chamber 40 progressively decreases in volume until the larger tappet portion 6a blocks the passages 46. During the remaining valve closure movement, the fluid in the chamber 40 can flow only through the holes 48, thereby producing a damping effect which slows the final phase of the valve closure so as to reduce the impact of the tappet 6 against the cam 5 both in operating conditions in which the cam 5 is separated from the tappet 6 and in operating conditions at low speeds when the cam 5 is in constant contact with the tappet 6.

The oil which leaks cyclically from the chamber 40 during its contraction phase is restored to the hydraulic circuit 20a and flows through the duct 20 either towards other chambers 40 in their expansion phases or towards the accumulators 50 in order to be returned to the circuit 20a in a phase in which it is drawn by the expansion of one or more chambers 40.

Claims

1. A valve control system for an internal combustion engine, of the type including:

   - a valve (3) movable between a closed position and an open position of a duct (2);
   - resilient biasing means (11, 12) for biasing the valve (3) towards its closed position;
   - valve control means for driving the valve (3) to effect cyclic movements to the open position against the action of the resilient means (11, 12), in which the control means include a rotatable cam (5) with an asymmetric profile which includes a first eccentric profile (29) portion for controlling the movement of the valve (5) towards its open position and a second eccentric profile portion (31), steeper than the first eccentric profile portion (29), for controlling the movement of the valve (5) towards its closed position, and a root portion (25) and a tip portion (27) extending between the first and second eccentric profile portions (29, 31) and connected to the first and second eccentric profile portions (29, 31) on opposite sides of the axis of rotation (F) of the cam (5), the root and tip portions (25, 27) both having respective constant radii (R_0, R_1) of curvature, and a tappet (6) operatively interposed between the cam (5) and the valve (3) and provided with an active surface (15) facing the cam (5), the active surface (15) being engageable by the cam profile at least during the opening movement of the valve (3) in such a way that, when the rotational speed of the cam (5) exceeds a threshold value, the active surface (15) of the tappet (6) loses contact with the cam profile, thereby uncoupling the closing movement of the valve (3) from the rotation of the cam (5) so that the closing movement occurs in a substantially fixed time period whatever the rotational speed of the cam (5), the active surface (15) of the tappet (6) having a flat portion (33) which is substantially orthogonal to the direction of movement (D) of the tappet (6) and a curved portion (35) connected to the flat portion (33) in such a way as to form a convex zone facing the cam (5), characterised in that the ratio of the radius of curvature (R_1) of the tip portion (27) to the radius of curvature (R_2) of the root portion (25) is between approximately 0.35 and 0.40.

2. A system according to Claim 1, characterised in that the ratio of the radius of curvature (R_1) of the tip portion (27) to the radius of curvature (R_2) of the root portion (25) is between approximately 0.36 and 0.398.

3. A system according to either Claim 1 or Claim 2, characterised in that the second eccentric profile portion (31) includes a straight portion tangential to a circle having a radius of curvature (R_2) which is smaller than the radius of curvature (R_0) of the root portion (25) of the cam (5), in which the ratio of the said smaller radius (R_2) to the said root radius$_0$ (R) is between approximately 0.5 and 0.8.

4. A system according to Claim 3, characterised in that the radius of curvature (R_3) of the curved portion (35) of the active surface (15) of the tappet (6) is constant and the ratio of this radius of curvature (R_3) to the radius of curvature (R_1) of the tip portion (27) of the cam (5) is between approximately 0.8 and 1.2.

5. A system according to any one of Claims 1 to 4, characterised in that it includes a hydraulic braking device for slowing the movement of the valve (3) in the last phase of its closure movement.

6. A system according to Claim 5, characterised in that the said tappet (6) includes a cup-shaped member

which is slidable axially in a fixed bush (8) and the hydraulic braking device includes a variable-volume chamber (40) defined between the cup-shaped member and the bush (8), the variable-volume chamber (40) being in constant communication with a pressurised oil supply through at least one hydraulic brake hole (48) which passes radially through the bush (8) and at least one outlet aperture (46) which puts the variable-volume chamber (40) in communication with the pressurised oil supply only when the valve (3) is spaced from its closed position.

7. A system according to Claim 6, characterised in that the variable-volume chamber (40) has a constant section in the direction of movement of the valve (3) and in that the ratio of the total area of the at least one outlet aperture (46) to the base area of the variable-volume chamber (40) is between approximately 0.3 and 1.

8. A system according to Claim 7, characterised in that the ratio of the total area of the at least one hydraulic brake duct (48) to the base area of the said variable-volume chamber (40) is between approximately 0.002 and 0.016.

9. A system according to any of Claims 1 to 8, characterised in that the hydraulic braking device is associated with a hydraulic circuit (20a) including at last one fluid accumulator (50).

10. A system according to Claim 9, characterised in that the at least one fluid accumulator (50) includes a resilient damper device (56).

11. A system according to either Claim 9 or Claim 10, characterised in that a non-return valve (58) is interposed between the hydraulic circuit (20a) and the pressurised fluid supply.

FIG. 1

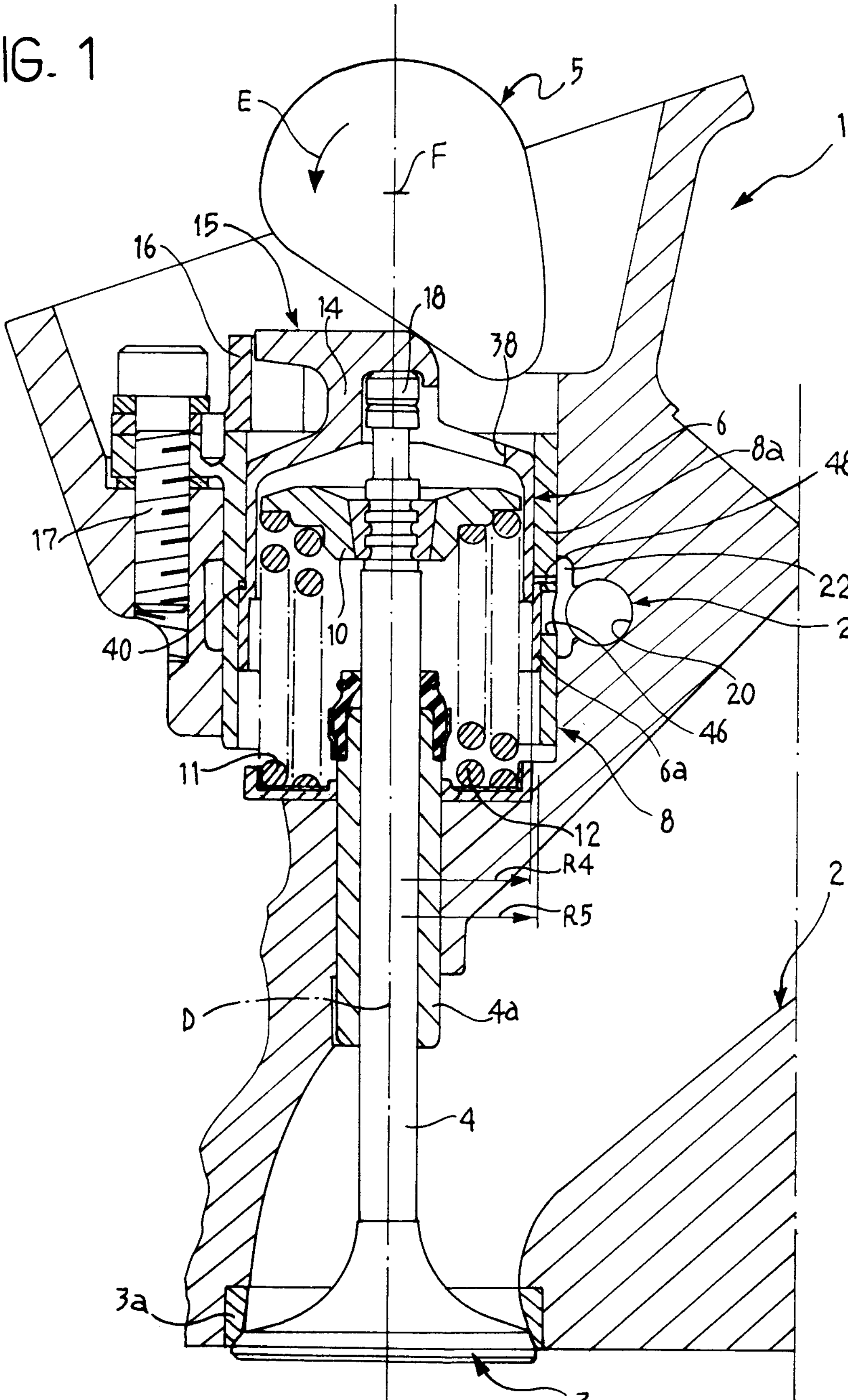
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FIG. 2

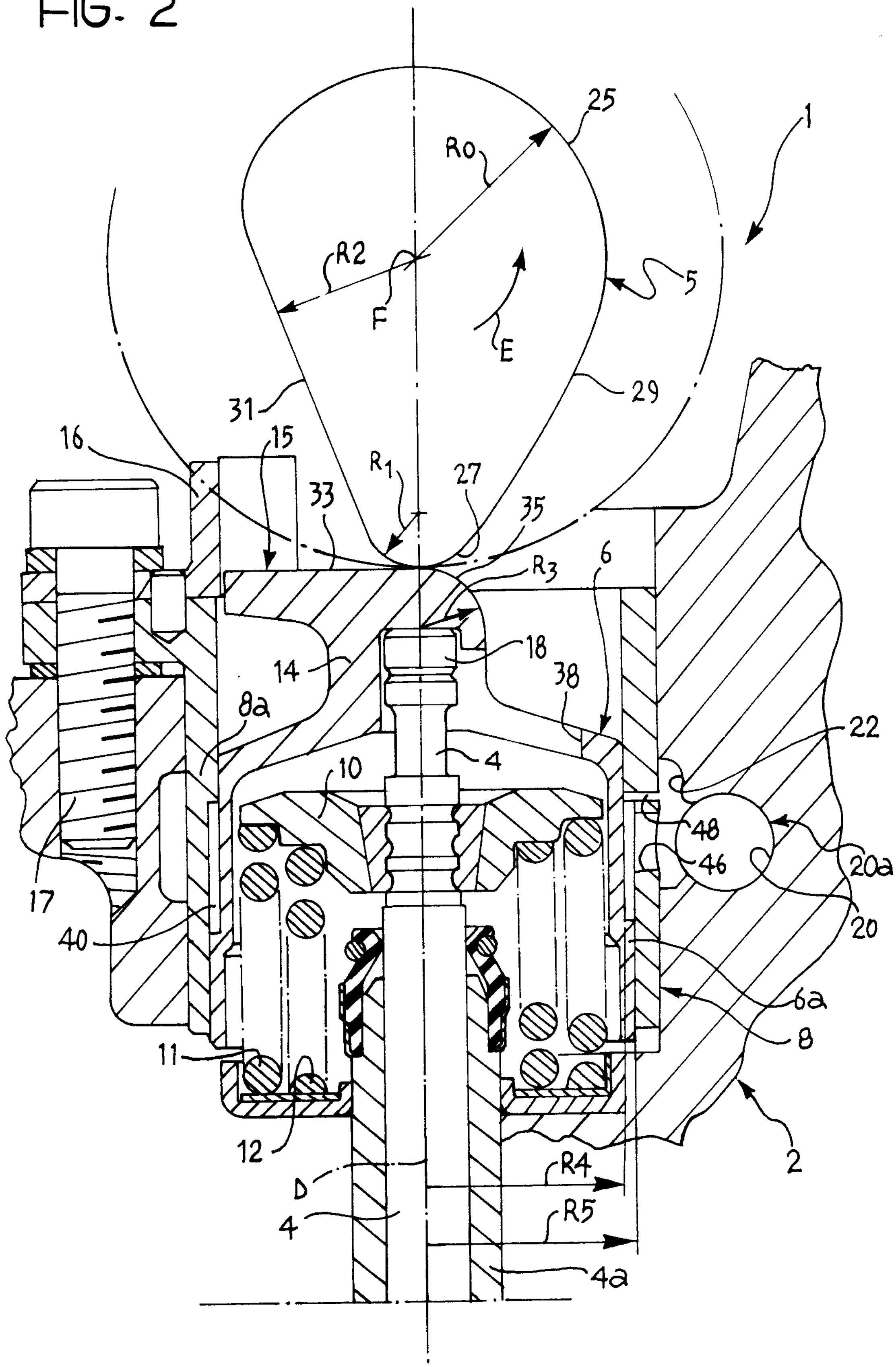
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FIG. 3

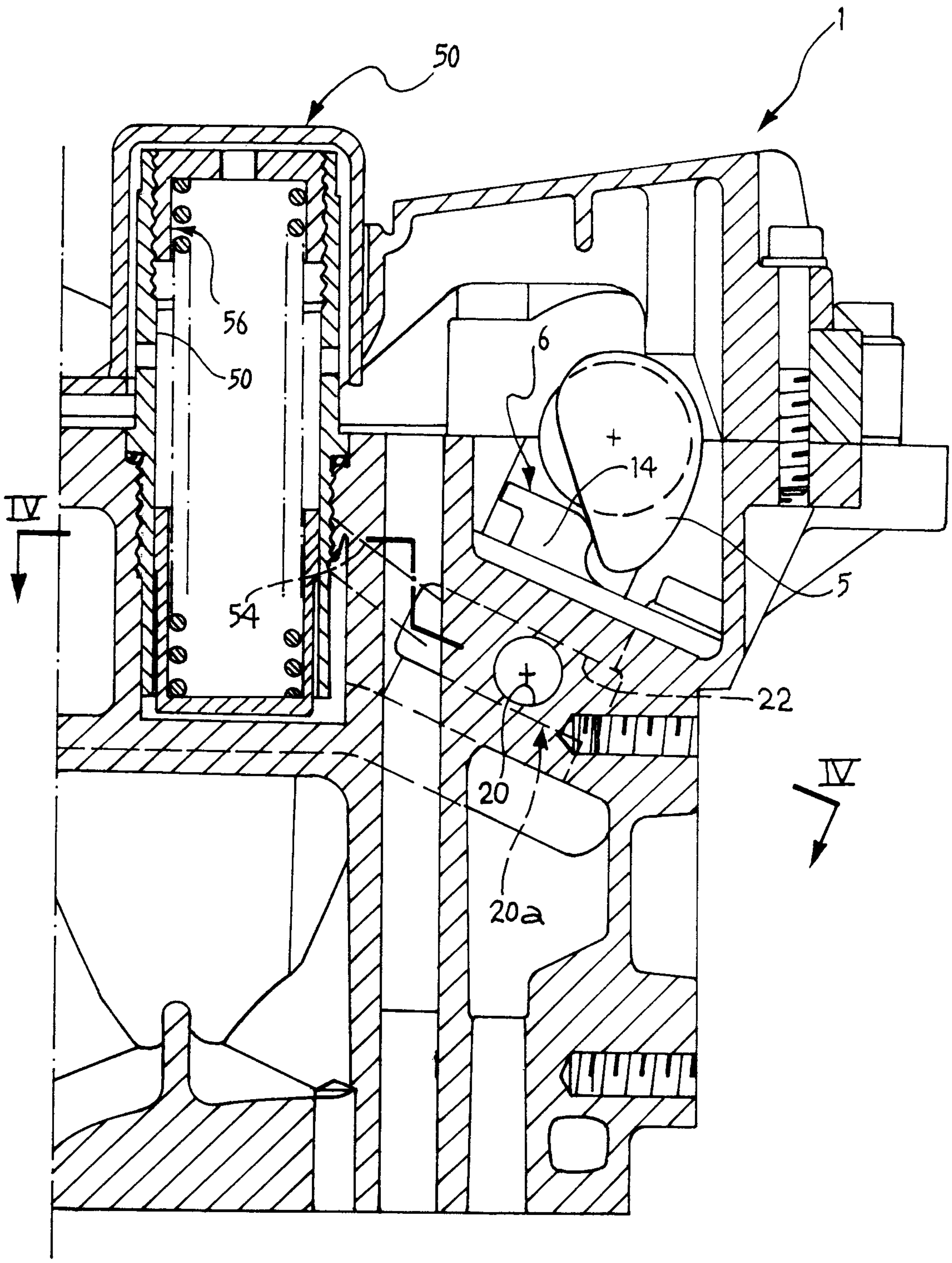
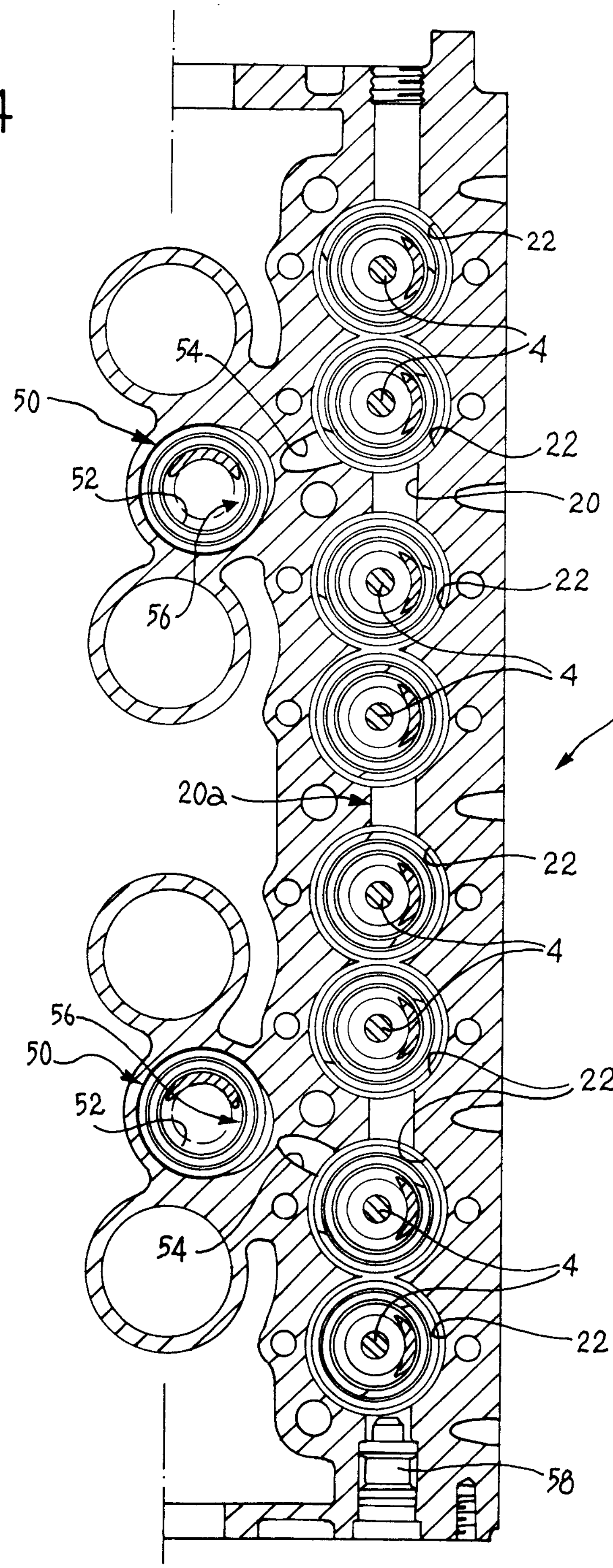


FIG. 4



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EUROPEAN SEARCH REPORT

Application Number
EP 95 11 9946

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X,D	EP-A-0 574 867 (CENTRO RICERCHE FIAT) 22 December 1993 * claims; figures *	1-8	F01L1/00 F01L1/08 F01L1/16 F01L13/00
A,P	EP-A-0 681 092 (FIAT AUTO SPA) 8 November 1995 * column 7, line 3-42; figure 7 *	9-11	
A	DE-A-39 13 530 (VOLKSWAGEN AG) * claim 1; figure 1 *	1	
A	FR-A-1 602 356 (LE MOTEUR MODERNE) * the whole document *	8	
A	FR-A-2 359 967 (CHRYSLER FRANCE)		

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
F01L

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	9 April 1996	Klinger, T

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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