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(54) IMPROVEMENTS IN OR RELATING TO VALVE TIMING MECHANISMS

(71) I, STEPHEN WILLIAM MITCHELL, a British Subject of 93 Lowercroft Road, Starling, Bury, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:-

The present invention relates to valve timing mechanisms, and in particular to variable valve timing mechanisms for internal combustion engines.

It is known that the volumetric efficiency of for example a four stroke poppet valve internal combustion engine is a function of the valve timing. An engine with a valve timing such that the inlet valve opens slightly before the piston is at the top dead centre (TDC) position and closes slightly after the piston is at the bottom dead centre (BDC) position will result in good volumetric efficiency and hence good torque characteristics at low engine speeds. In contrast if good volumetric efficiency and hence high power is to be obtained at high engine speeds it is necessary for the inlet valve to open substantially before the piston is at the TDC position and close substantially after the piston is at the BDC position.

A further problem met when considering valve timing mechanisms is that of inlet and exhaust valve overlap, that is the condition in which both the inlet and exhaust valves are open when the piston is approaching and departing from the TDC position. The reduction of this overlap at low engine speeds results in reduced exhaust emissions by preventing a proportion of the incoming air/fuel charge from going into the exhaust system. It is also known that benefit can be obtained by retarding the opening of the exhaust valve at low engine speeds, so as to obtain more work out of the expansion stroke and thereby reduce fuel consumption, and by advancing the opening of the exhaust valve at the high engine speeds, so as to avoid work in scavenging the exhaust gases.

In view of the above, engines with fixed valve timing must be a compromise.

It is an object of the present invention to provide a valve timing mechanism which avoids the problems associated with known valve timing mechanisms.

According to the present invention, there is provided a valve timing mechanism for an internal combustion engine including at least one valve-actuating camshaft driven from a crankshaft, the mechanism comprising a member which is arranged in use to be rotatable by the crankshaft and movable in translation relative to the camshaft in dependence upon an engine operating condition, the movable member being connected to the camshaft by an eccentric linkage such that movement of the movable member relative to the camshaft varies the angular position of the camshaft about its axis of rotation relative to the angular position of the crankshaft about its axis of rotation and also varies the angular velocity of the camshaft relative to the angular velocity of the crankshaft, thereby varying the valve timing.

The present invention also provides an internal combustion engine comprising at least one cylinder, at least one inlet and at least one exhaust valve for the or each cylinder, and inlet valve camshaft and an outlet valve camshaft for the or each cylinder, a crankshaft, and means for driving the camshafts from the crankshaft, the driving means comprising a movable member rotatably driven by the crankshaft and connected to at least one camshaft by an eccentric linkage, and the movable member being movable in translation relative to said at least one camshaft in dependence upon an operating condition of the engine, such movement of the movable member causing the eccentric linkage to vary the angular position of the camshaft about its axis of rotation relative to the angular position of the crankshaft about its axis of rotation and also to vary the angular velocity of the camshaft relative to the angular velocity of the crankshaft, thereby varying the valve timing.

Preferably the movement in translation of the movable member is in dependence upon engine load and/or speed. The movable member may be supported on a pivotal arm. The movable member may alternatively be mounted on a slide for example.

The movable member is advantageously in the form of a rotatable plate provided with a slot extending radially with respect to the axis of rotation of the plate, the slot receiving a follower supported eccentrically by the camshaft. Alternatively, the movable member may support an eccentrically mounted follower which is slidably received in a slot in a plate supported by the camshaft.

An embodiment of the present invention will now be described by way of example with reference to the accompanying drawings, in which:-

Figure 1 is a sectional view through the cylinder of an engine embodying the present invention;

Figure 2 is a sectional view on the line 2-2 of Figure 1;

Figure 3 is a sectional view on the line 3-3 of Figure 1;

Figure 4 is a sectional view of a portion of Figure 2 to an enlarged scale and showing more detail;

Figures 5, 6 and 7 show the same view as Figure 4 with components of the engine being relatively displaced;

Figure 8 illustrates the angular movement of an inlet camshaft under varying conditions; and

Figure 9 is a detailed sectional view of a piston and cylinder arrangement shown in Figure 3.

Referring to Figure 1, the illustrated engine has many conventional features which it is considered do not need detailed description. The engine has four cylinders 1, 2, 3 and 4 each having two inlet valves 5 and two exhaust valves 6. Four inlet valve camshafts 7, 8, 9 and 10 are shown, four exhaust valve camshafts also being provided but not shown in Figure 1. Camshafts 7 and 10 are driven by respective shafts extending through camshafts 8 and 9. A crankshaft 11 rotatably drives a movable member 12 via gears 13 and 14, the member 12 being supported on an arm 15 pivotal about a journal 16 on which the gear 14 is supported.

Referring now to Figures 2 and 3, an exhaust camshaft 17 for the exhaust valves of cylinder 1 is shown in Figure 2, and an exhaust cam 18 for the exhaust valves of cylinder 3 is shown in Figure 3. It may be seen from Figure 3 that the member 12 is driven by the gear 14 via a further member 19, the members 12 and 19 being meshed together and movable together with the arm 15 pivoted about the journal 16. The position of the arm 15 is determined by a device described in more detail with reference to Figure 9.

Referring now to Figure 4, the inlet and exhaust valves 5, 6, inlet and exhaust ports 21, 22 and a sparking plug 23 are shown. In addition, the timing mechanisms driving the inlet and exhaust camshafts 7, 17 are shown in more detail.

The inlet valve camshaft 7 supports a cam 24 and an eccentric 25 supporting a follower 26. The follower 26 is located in a slot 27 provided in the movable member 12 (hereinafter referred to as the eccentric plate). The exhaust camshaft 17 which operates the exhaust valves of the cylinder 1 is also provided with a cam 28, an eccentric 29, a follower 30 and a slot 31 in the movable member or eccentric plate 19. The eccentric plates 12 and 19 are mounted in bearings and are driven at half engine speed by the gears 13, 14 from the engine crankshaft 11. The bearings are housed in the arm 15 which can pivot such that the axes of rotation of the eccentric plates 12 and 17 can coincide with the axes of the camshafts 7 and 17 or can be moved eccentric to these camshafts.

Figure 4 shows the high engine speed condition where the position of the arm 15 is against a suitable stop and is such that the axis of the inlet eccentric plate 12 coincides with the axis of the inlet camshaft 7. The timing of the inlet valve opening and closing is as determined by the design of the inlet cam 24. The axis of the exhaust valve eccentric plate 19 also coincides with the axis of the exhaust camshaft 17 and the timing of the exhaust valve opening and closing is as determined by the design of the exhaust cam 28. The cams are designed so that the inlet valve 5 will open substantially before the piston is at the top dead centre position and close substantially after the piston is at the bottom dead centre position. The exhaust valve 6 will be opened substantially before the piston is at the BDC position and close substantially after the piston is at the TDC position.

Figure 4 shows the inlet valve 5 about to open and the exhaust valve about to open, and Figure 5 shows the inlet valve 5 about to close and the exhaust valve 6 about to close. It can be seen that this is in the high engine speed condition where the axes of the camshaft 7 and 17 coincide with the eccentric plates 12 and 19. Figures 4 and 5 do not show the correct relative positions of the inlet and exhaust camshafts but simply the opening and closing positions. The reference numerals of Figure 4 are not shown but are referred to in the description of Figures 5 to 7.

Figure 6 shows the inlet and exhaust valves about to open, and Figure 7 shows the inlet and exhaust valves about to close when the position of the arm 15 is moved so that the axes of the eccentric plates 12 and 19 are at their maximum eccentricity relative to the axes of the camshafts 7 and 17 respectively. The arm 15 is moved to the described position by the piston and cylinder arrangement 20 in low engine speed conditions, as will be apparent from the following description of Figure 9. The angular positions of the followers 26 and 30 relative to the direction of movement of the arm 15 is such that the eccentric plates 12 and 19 have to turn through a greater angle in order for the cams 24 and 28 to be in position to start opening the valves. In the case of the inlet valve 5 which in the high engine speed position normally opens substantially before the piston is at the TDC position, the engine crankshaft will have to rotate twice the angular movement that the eccentric plate 4 has to turn through and consequently the inlet valve will open later than it would in the high engine speed condition. Thus it now opens slightly before the piston is at the TDC position, although it can be arranged to open slightly after the TDC position if necessary. In the case of the exhaust valve 6 which in the high engine speed condition normally opens substantially before the piston is at the BDC position, the engine crankshaft will have to rotate twice the angular movement that the eccentric plate 19 has to turn through, and consequently the exhaust valve 6 will open later than it would in the high engine speed condition. It thus opens slightly before the piston is at the BDC position.

When the arm 15 is in the low engine speed position, the effect of the eccentricity of the eccentric plates 12 and 19 relative to the centres of the inlet and exhaust camshafts 7, 17 not only alters the opening positions of the inlet and exhaust cams relative to the engine crankshaft (Figure 6) but the eccentric plates 12 and 19 have to rotate the camshafts through a reduced angular movement in order to close the inlet and exhaust valves (Figure 7). The reduced angular movements of the eccentric plates which are driven at half crankshaft speed results in both the inlet valves and the exhaust valves not only opening later but closing earlier, that is at full eccentricity the inlet valve will open slightly before the piston is at the TDC position and close slightly after the piston is at the BDC position, and the exhaust valve will open slightly before the piston is at the BDC position and will close slightly after the TDC position although it can be arranged to close slightly before the TDC position.

Figure 8 illustrates the angular movement of the inlet camshaft 7 relative to the constant angular movement of the eccentric plate 12 during the full period of inlet valve opening and closing. the movement of the camshaft 7 whilst the cam 24 is on its base circle not being relevant. It can be seen that the direction of the eccentricity is along an axis such that the reduction in angular movement as a result of the eccentricity is equally divided between the opening and closing of the cam, but this need not necessarily be so and another axis could be chosen that would alter the rate of opening relative to the rate of closing so that the angular difference between the opening of the inlet valve at high engine speeds to the opening of the inlet valve at low engine speeds would be different to the angular difference between the inlet valve closing at high engine speeds and the inlet valve closing at low engine speeds. These differences can also apply to the operation of the exhaust valve.

The rate of change of angular velocity of the camshaft when the eccentric plate 12 is eccentric to the camshaft substantially follows a smooth curve and the minimum rate of change of angular velocity is advantageously arranged when the cam starts to open or just close the valve and considering the modifying effect of the rate of change of valve opening velocity as determined by the cam profile, the resulting opening and closing trajectory of the valve at low speeds is ideally suited to the valve opening requirements as demanded by the engine due to the fact that cam profiles which are designed for high engine speeds apply negative acceleration to the valve gear when the valve has opened less that half of its total movement in order that the valve return spring can provide the force to keep the valve assembly in contact with the cam profile. At low engine speeds when the inertia loadings of the valve gear are small, the effect of the eccentric mechanism delays the negative acceleration acting on the valve so that the inlet valve for example will be open further at the position of maximum piston velocity which would suggest a further improvement in volumetric efficiency.

Figure 9 illustrates the piston and cylinder arrangement 20 which controls the position of the arm 15.

The arrangement 30 comprises a piston 32 movable in a cylinder 33 under the influence of oil pressure derived from the engine oil pump. A control piston 34 is positioned by a unit (not shown) such as a centrifugal device to sense engine speed which can be used in conjunction with a diaphragm in the inlet manifold to sense manifold pressure which is indicative of engine load, or it can be positioned by means of an electrical unit controlled from suitable transducers. the actuating force on the control piston 34 being small. The control piston 34 is carried by the body of the piston 32 so that the relative movement of the two pistons 32 and 34 can provide a regulating action.

Oil from the engine oil pump is directed into the cylinder 33 on the annulus side (left hand side as shown in Figure 9) of the piston 32 the area of this side of the piston being 50% of the full area of the piston. A feed from the annulus side of the piston 32 is taken into a recess 35 where it is directed through holes into a recess between lands 36, 37 of the control piston 34.

The width of the land 36 of the control piston 34 is less than that of a recess 38 in the body of the piston 32 such that oil will flow from the recess between lands 36 and 37 into recess 38 and escape into the recess between a land 39 and land 36 of the control piston 34 and then leak away into the engine through the holes provided around the left hand end (as shown in Figure 9) of the control piston 34. The rate at which the oil flows away from the recess 38 into the recess between lands 36 and 39 of the control piston 34 is a function of the difference between the width of the land 36 and the width of the recess 38 in the body of the piston 32, and the flow is such that the pressure in the recess 38 is half the supply pressure. The oil in recess 38 is fed to the right hand side (as shown in Figure 9) of the piston 32 so as to act on the full area of the piston. It can be seen that half the supply pressure acting on the full area of the piston will equal the thrust produced by the full supply pressure acting on the annulus area of the piston which is 50% of the full area.

Any small movement of the control piston 34 will alter the pressure acting on the full side of the piston 32 which will cause the piston to move. This movement relative to the control piston will continue until a state of equilibrium is once again reached.

The piston 32 now positions the arm 15 in accordance with the position of the control piston 34 which in turn is positioned as a function of the speed, load or speed and load of the engine. The position of the piston 32 and the arm 15 will result in the correct amount of eccentricity of the eccentric plates 12 and 19 to give the desired valve timing for the particular engine speed, load or speed and load condition.

It will be appreciated that the described engine can use conventional cams designed in accordance with mathematical formulae to give the highest possible valve operating speeds with no additional mass at high engine speeds other than valve, valve tappet buckets, valve collets and collet head and a proportion of the mass of the valve spring as with a conventional overhead camshaft arrangement. It is also appreciated that the variable eccentric plate mechanism can control the angular velocity of a fixed eccentric which can be used in place of a cam to open and close the valves.

It will also be appreciated that when the axes of the camshafts and eccentric plates coincide the followers will not move radially within the slots in the eccentric plates, but radial movement will occur when the axes do not coincide. In the described embodiment the axes are arranged to coincide at high engine speeds so that the maximum radial sliding movement of the followers which does occur is at a relatively low speed, but if desired the axes could be arranged to coincide at or intermediate a low speed.

The pivotal arm could be replaced by for example a sliding support if desired, and the drive between the crankshaft and eccentric plates can be other than via gears, for example via chains.

WHAT I CLAIM IS:-

1. A valve timing mechanism for an internal combustion engine including at least one valve-actuating camshaft driven from a crankshaft, the mechanism comprising a member which is arranged in use to be rotatable by the crankshaft and movable in translation relative to the camshaft in dependence upon an engine operating condition, the movable member being connected to the camshaft by an eccentric linkage such that movement of the movable member relative to the camshaft varies the angular position of the camshaft about its axis of rotation relative to the angular position of the crankshaft about its axis of rotation, and also varies the angular velocity of the camshaft relative to the angular velocity of the crankshaft, thereby varying the valve timing.

2. An internal combustion engine comprising at least one cylinder, at least one inlet and at least one exhaust valve for the or each cylinder, an inlet valve camshaft and an outlet valve camshaft for the or each cylinder, a crankshaft, and means for driving the camshafts from the crankshaft, the driving means comprising a movable member rotatably driven by the crankshaft and connected to at least one camshaft by an eccentric linkage, and the movable member being movable in translation relative to said at least one camshaft in dependence upon an operating condition of the engine, such movement of the movable member causing the eccentric linkage to vary the angular position of the camshaft about its axis of rotation relative to the angular position of the crankshaft about its axis of rotation and also to vary the angular velocity of the camshaft relative to the angular velocity of the crankshaft, thereby varying the valve timing.

3. An internal combustion engine according to claim 2, comprising means for moving the movable member in dependence upon engine load.

4. An internal combustion engine according to claim 2, comprising means for moving the movable member in dependence upon engine speed.

5. An internal combustion engine according to claim 2, comprising means for moving the movable member in dependence upon engine load and speed.

6. An internal combustion engine according to any one of claims 2 to 5, wherein the movable member is supported on a pivotal arm.

7. An internal combustion engine according to claim 6, wherein the orientation of the pivotal arm about its pivot is controlled by a piston and cylinder arrangement comprising a control piston the position of which is dependent upon an engine operating condition, a main piston in which the control piston is slidably received, and a cylinder in which the main piston is received, a linkage being provided between the main piston and the pivotal arm.

8. An internal combustion engine

according to claim 6 or 7, wherein the pivotal arm supports two movable members one of which drives the inlet valve camshaft and the other of which drives the outlet valve camshaft, the two movable members rotating at the same rate as each other.

9. An internal combustion engine according to claim 8, wherein the movable members are meshed together and one of them is driven by a gear train from the crankshaft, one gear in the train being pivotal about the same axis as the pivotal arm.

10. An internal combustion engine according to any one of claims 2 to 9, wherein the or each movable member comprises a rotatable plate provided with a slot extending radially with respect to the axis of rotation of the plate, the slot slidably receiving a follower supported eccentrically by the camshaft.

11. An internal combustion engine according to any one of claims 2 to 10, comprising four in-line cylinders, four in-line inlet valve camshafts, and four in-line exhaust valve camshafts, the camshafts being arranged in pairs on opposite side of a single drive means with one camshaft of each pair being driven by a shaft extending through the other camshaft of the pair.

12. An internal combustion engine according to any one of claims 2 to 11, the timing mechanism being such that the inlet valve timing may be altered to retard the opening of an inlet valve while simultaneously the closing of the inlet valve is advanced, and the exhaust valve timing may be altered to retard the opening of an exhaust valve while simultaneously the closing of the exhaust valve is advanced.

13. An internal combustion engine according to claim 12, wherein the degree of alteration to the position of inlet valve opening at any particular instant cam be different to the degree of alteration to the position of inlet valve closing at the same particular instant.

14. An internal combustion engine according to claim 12 or 13, wherein the degree of alteration to the position of exhaust valve opening at any particular instant can be different to the degree of alteration to the position of exhaust valve closing at the same particular instant.

15. An internal combustion engine according to claim 12, 13 or 14, wherein the degree of alteration to the inlet valve opening and closing at any particular instant can be different to the degree of alteration to the exhaust valve opening and closing at the same particular instant.

16. An internal combustion engine substantially as hereinbefore described with reference to the accompanying drawings.

17. A valve timing mechanism substantially as hereinbefore described with reference to the accompanying drawings.

MARKS & CLERK,
Chartered Patent Agents,
Agents for the Applicant

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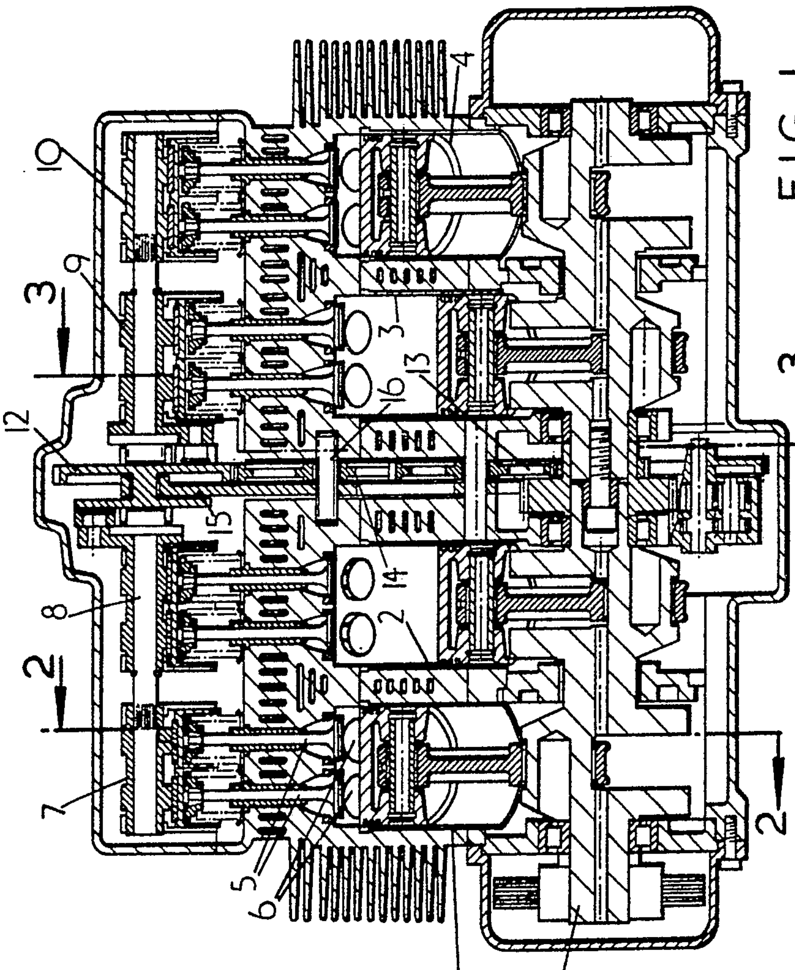

FIG. 1

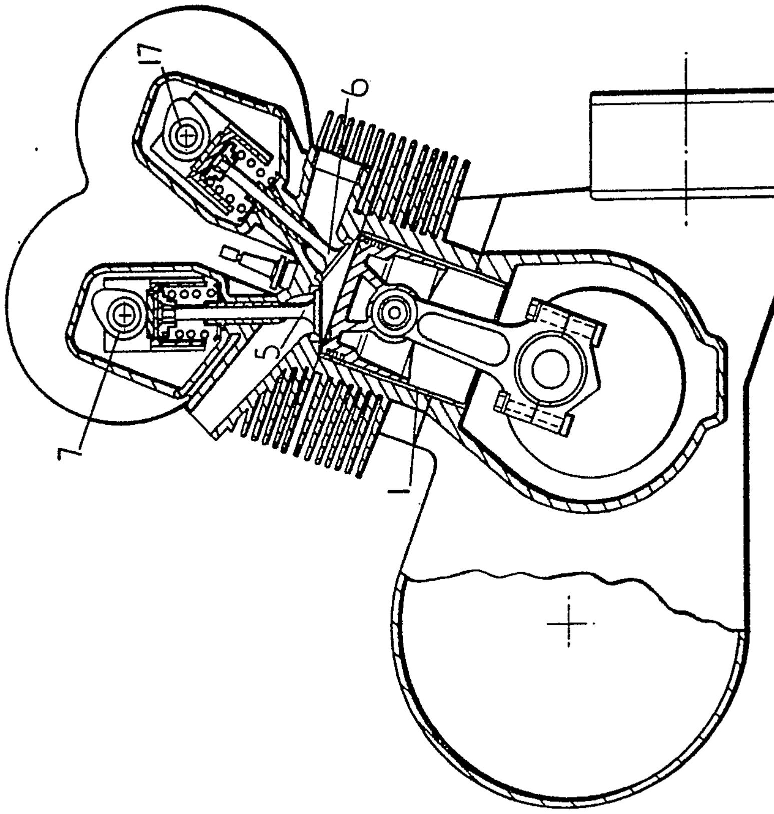

FIG. 2

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Sheet 3

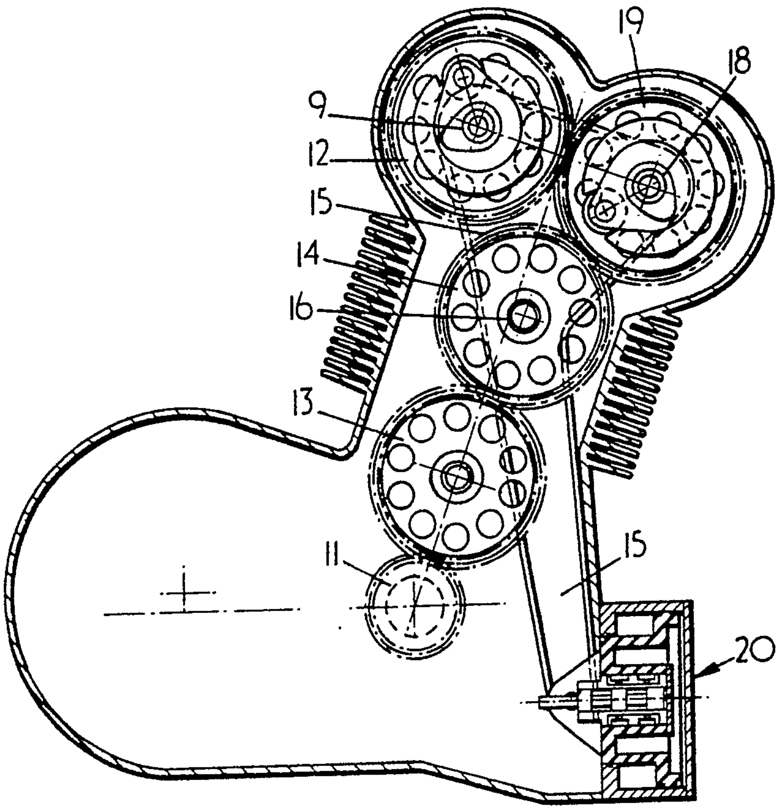

FIG. 3

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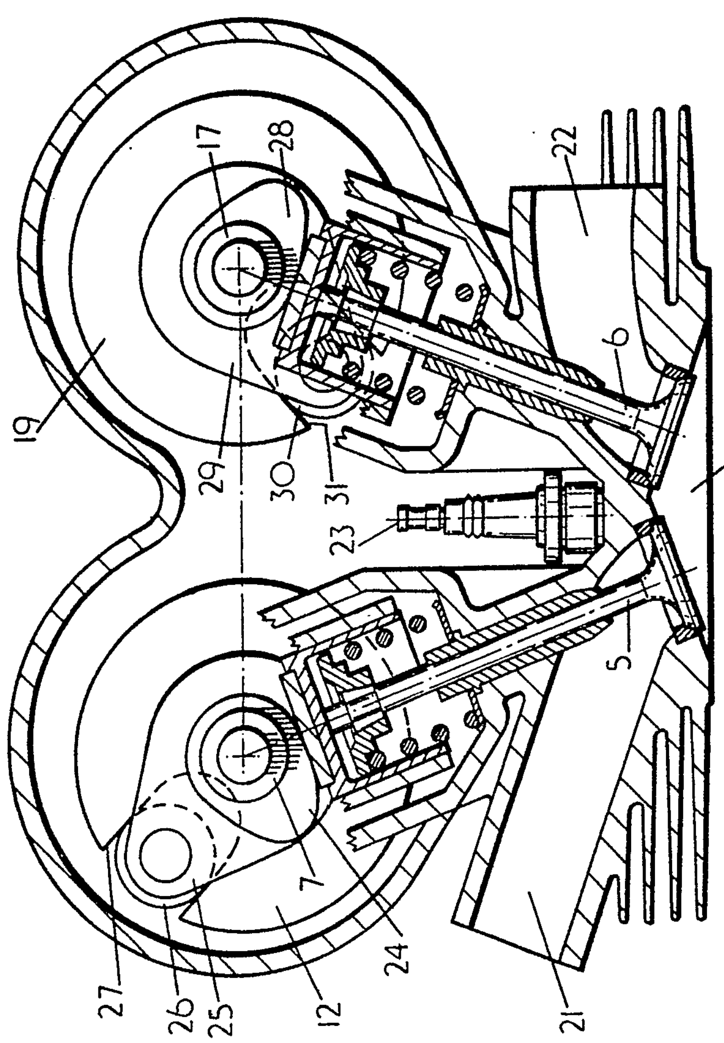

FIG. 4

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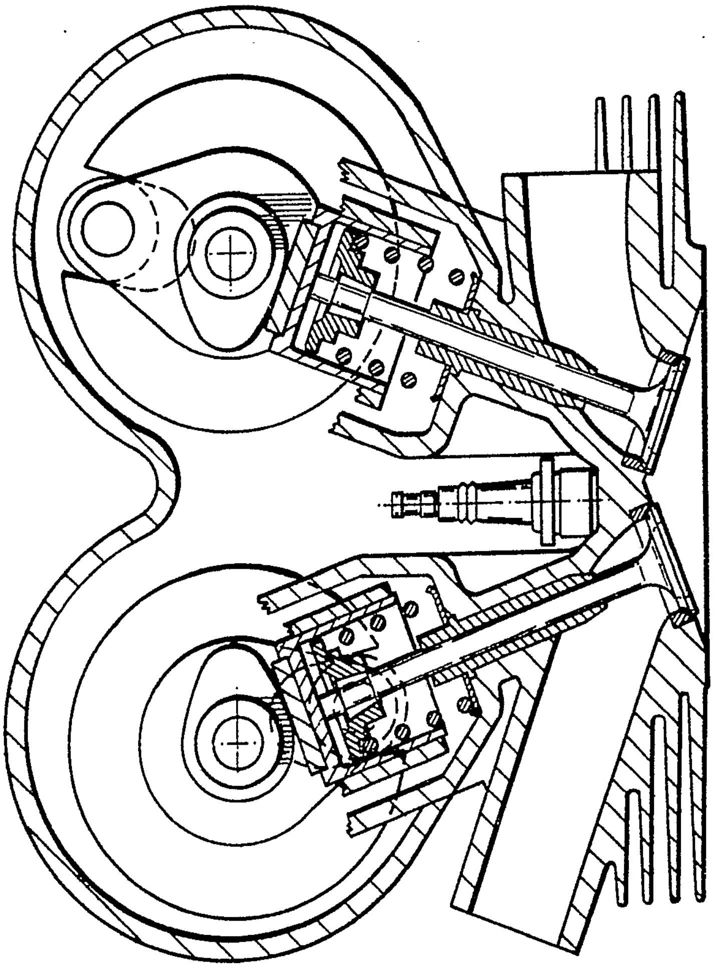

FIG. 5

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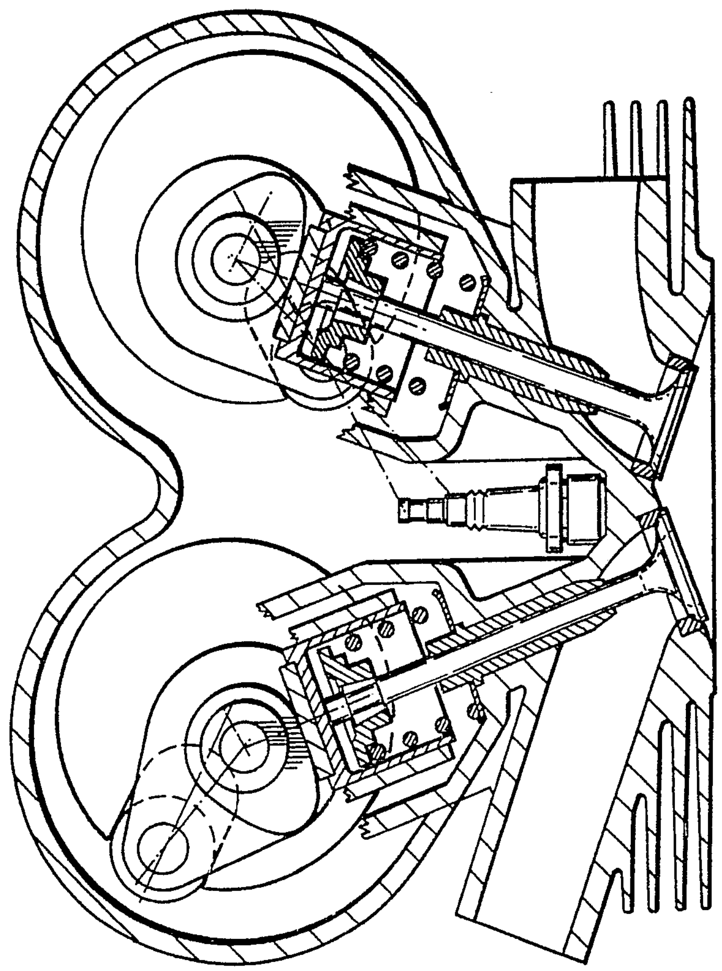

FIG. 6

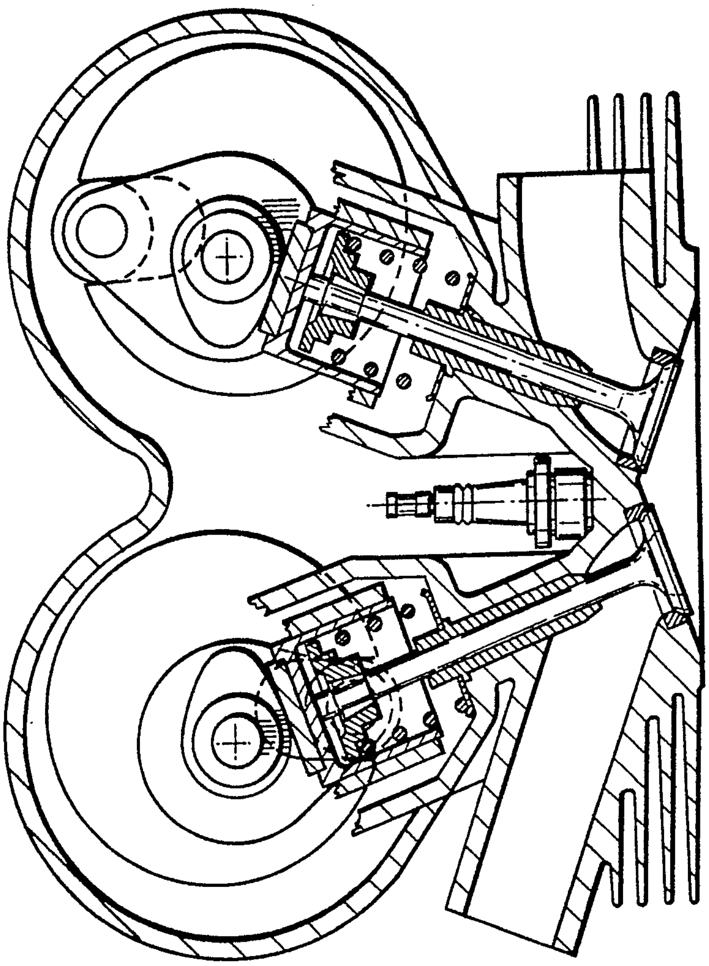

FIG. 7

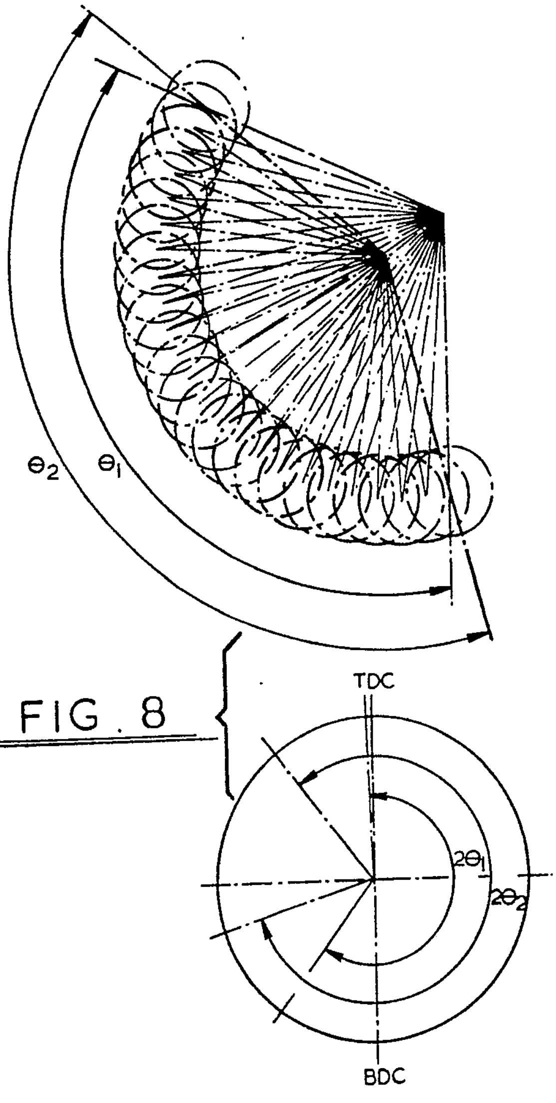

FIG. 8

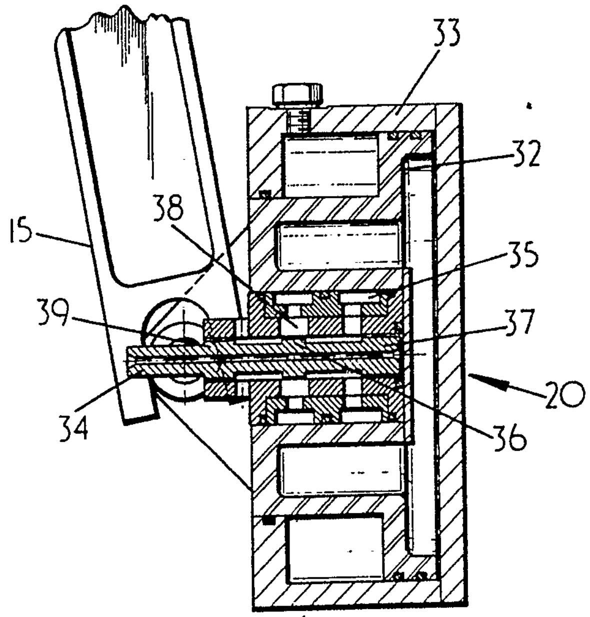

FIG. 9