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Torazza et al.

[54] VALVE-ACTUATING MECHANISM FOR AN INTERNAL COMBUSTION ENGINE

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Feb. 3, 1969 Italy 50581 A/69

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[51] Int. Cl. F01l 1/34, F01l 1/04

[58] Field of Search 123/90.15, 90.16, 90.17, 90.18, 123/90.24, 90.27, 90.31, 90.39, 90.48, 90.6

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Primary Examiner—Al Lawrence Smith
Attorney—Sughrue, Rothwell, Mion, Zinn & Macpeak

[57] ABSTRACT

A valve-actuating mechanism for an internal combustion engine has a number of rocker arms for operating the respective valves, each rocker arm having a profiled cam surface for engaging the respective valve, and means, preferably hydraulically operated, for varying the valve movement produced by the rocker arm in dependence upon the engine speed and load to vary the valve timing for optimum efficiency.

16 Claims, 10 Drawing Figures

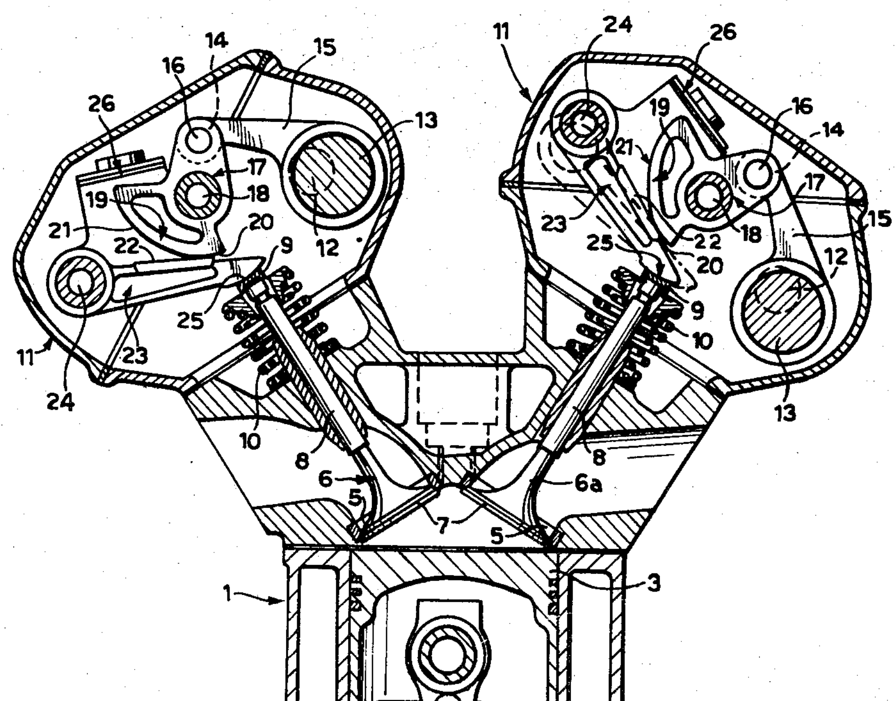
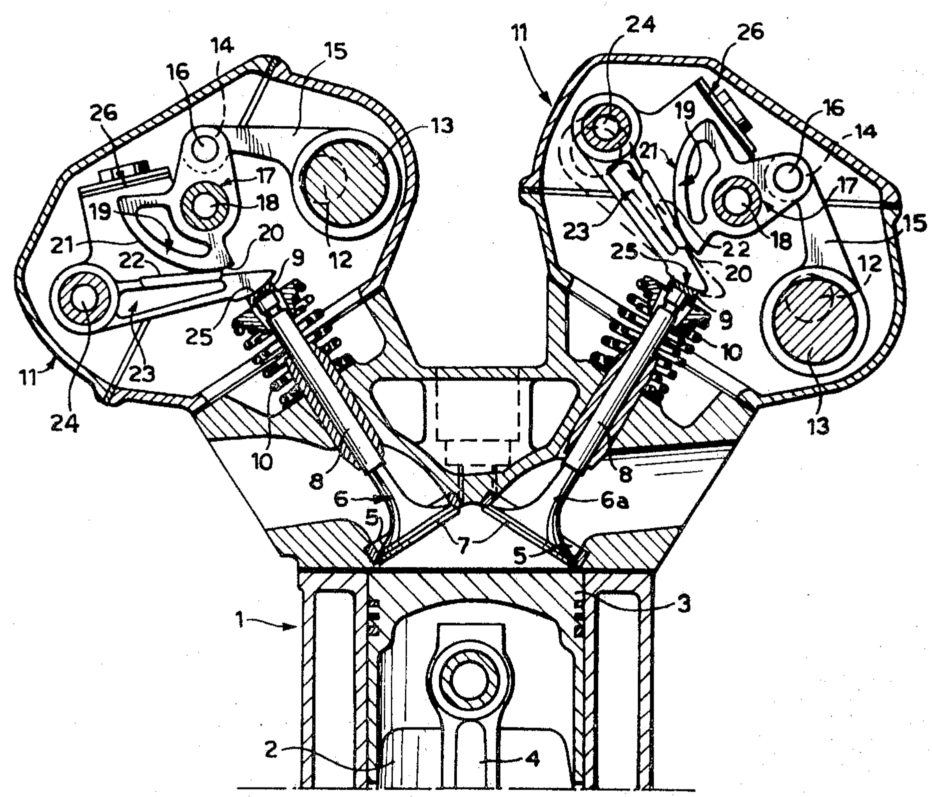


Fig_1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6a
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3
1
2
4

Fig. 2

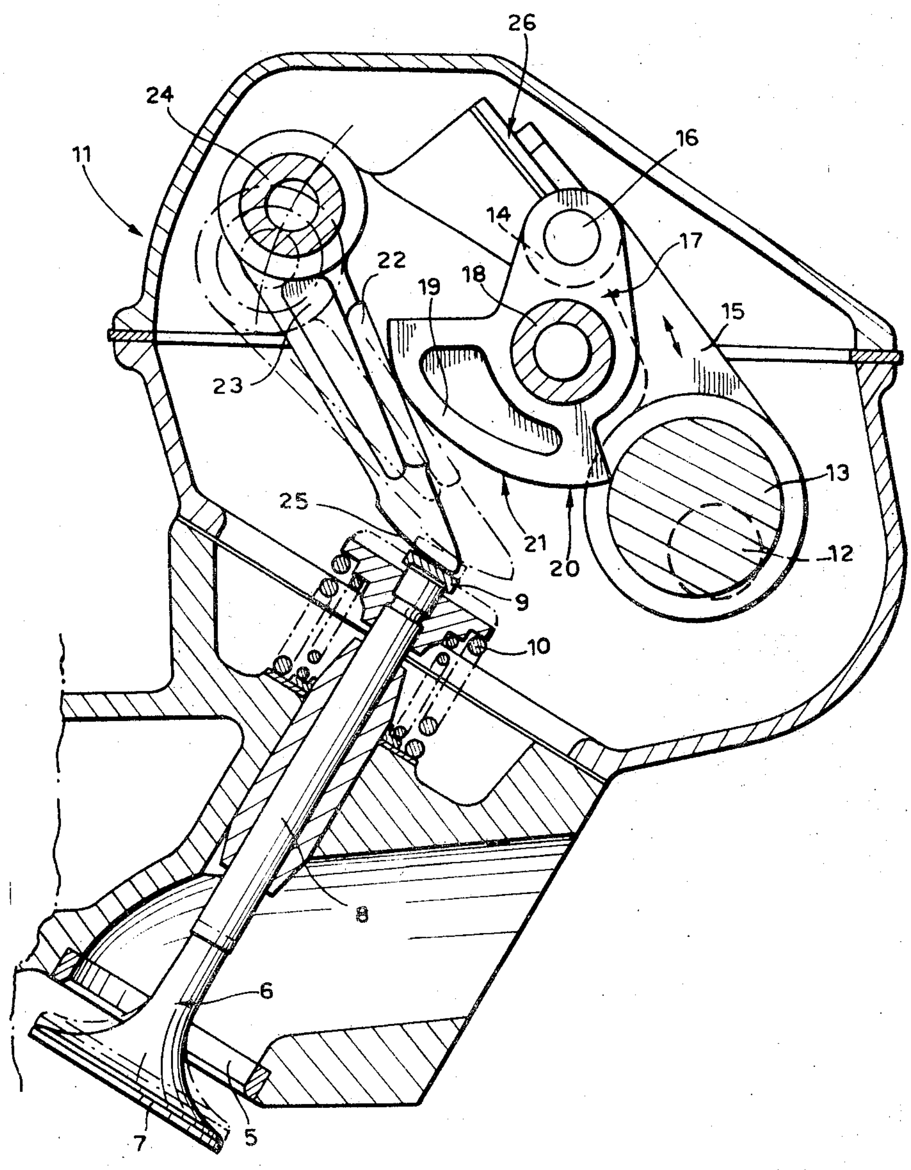

Fig. 3

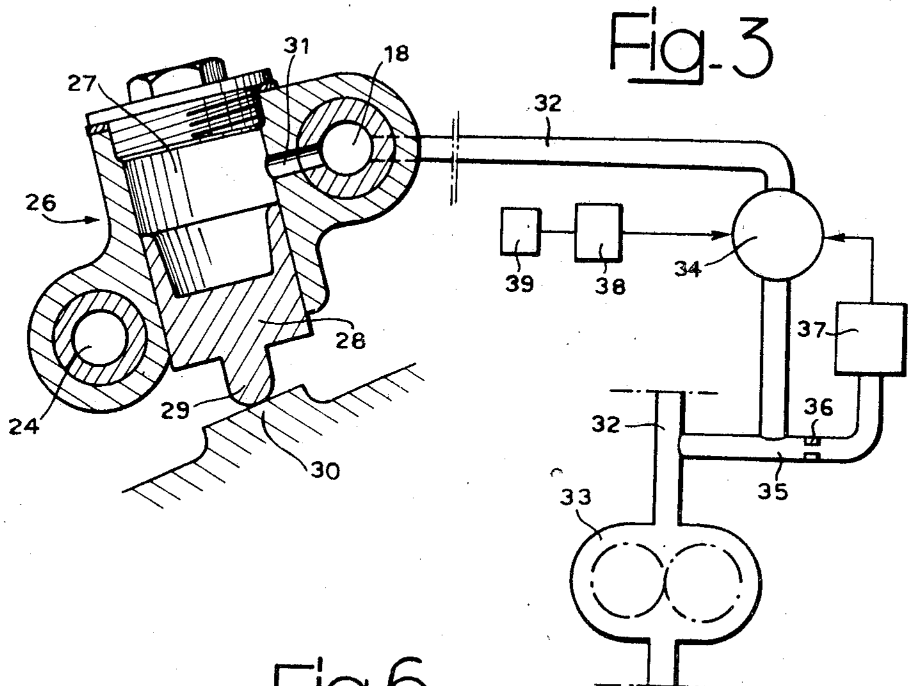
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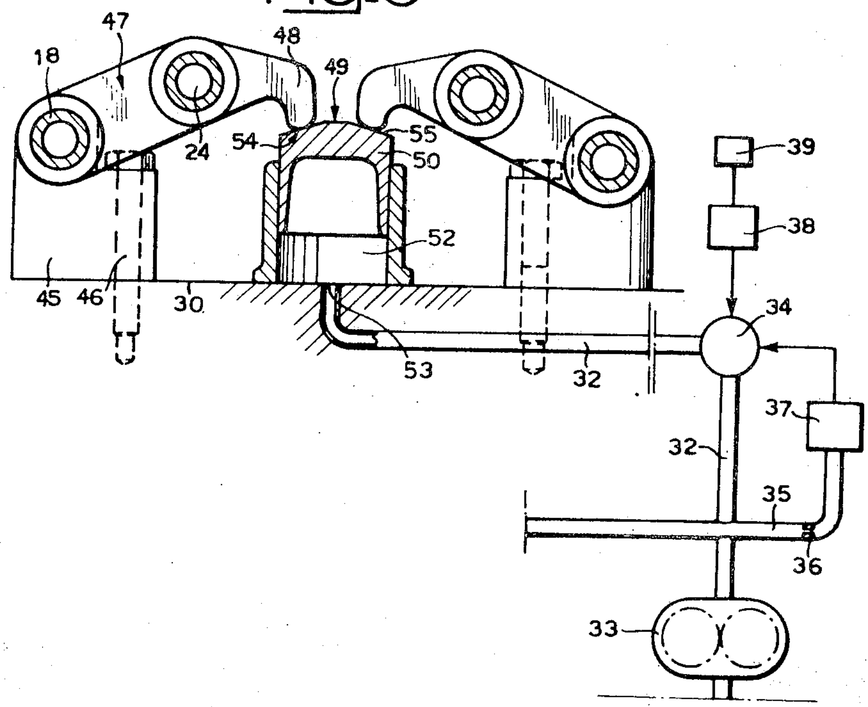
Fig. 6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48
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33

Fig. 4

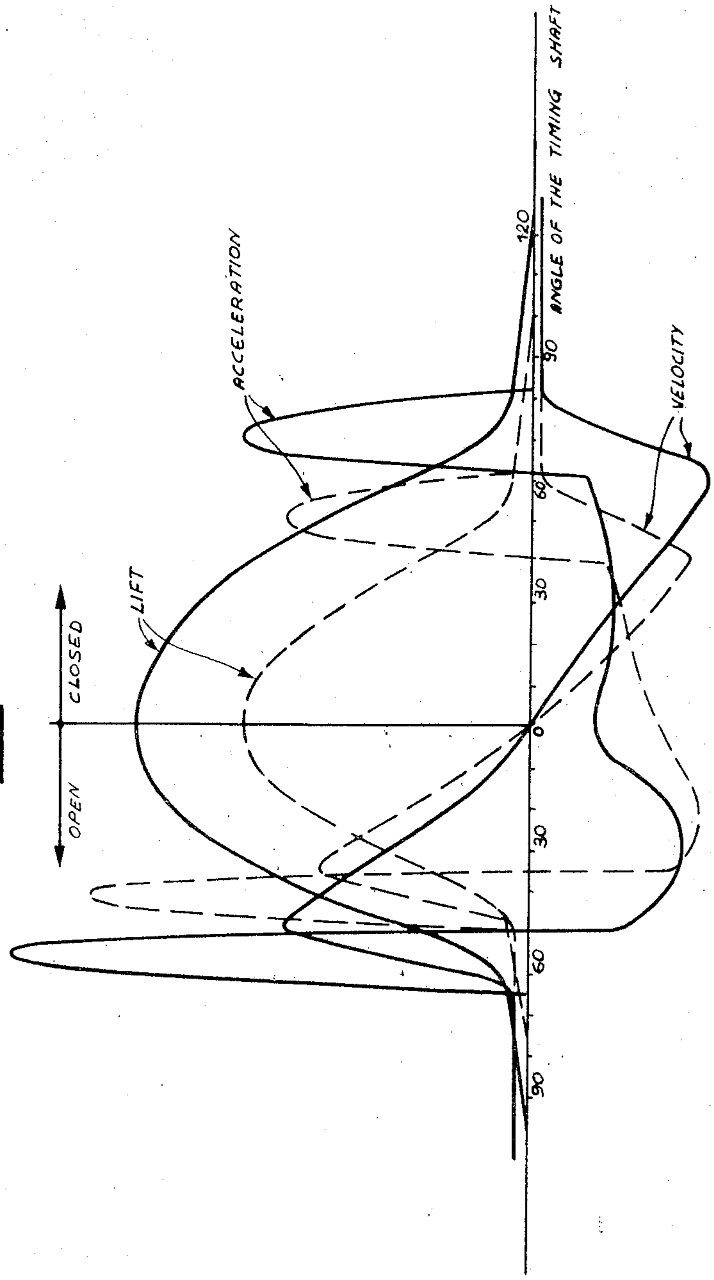
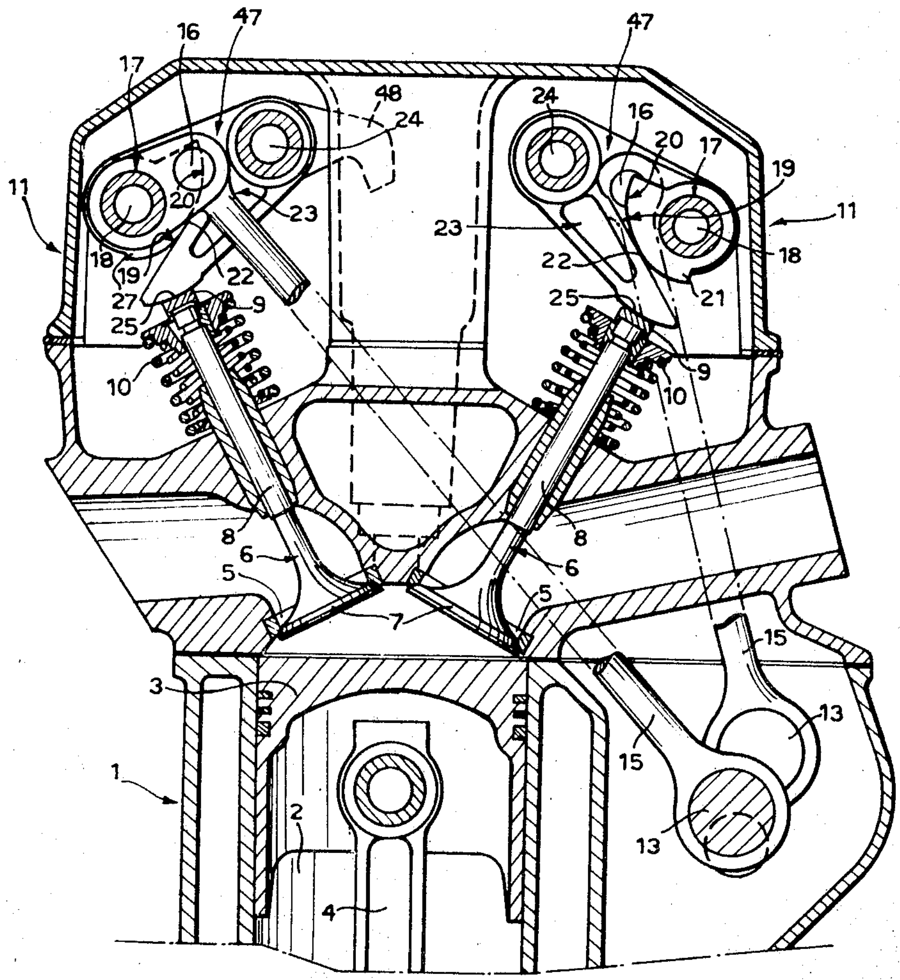
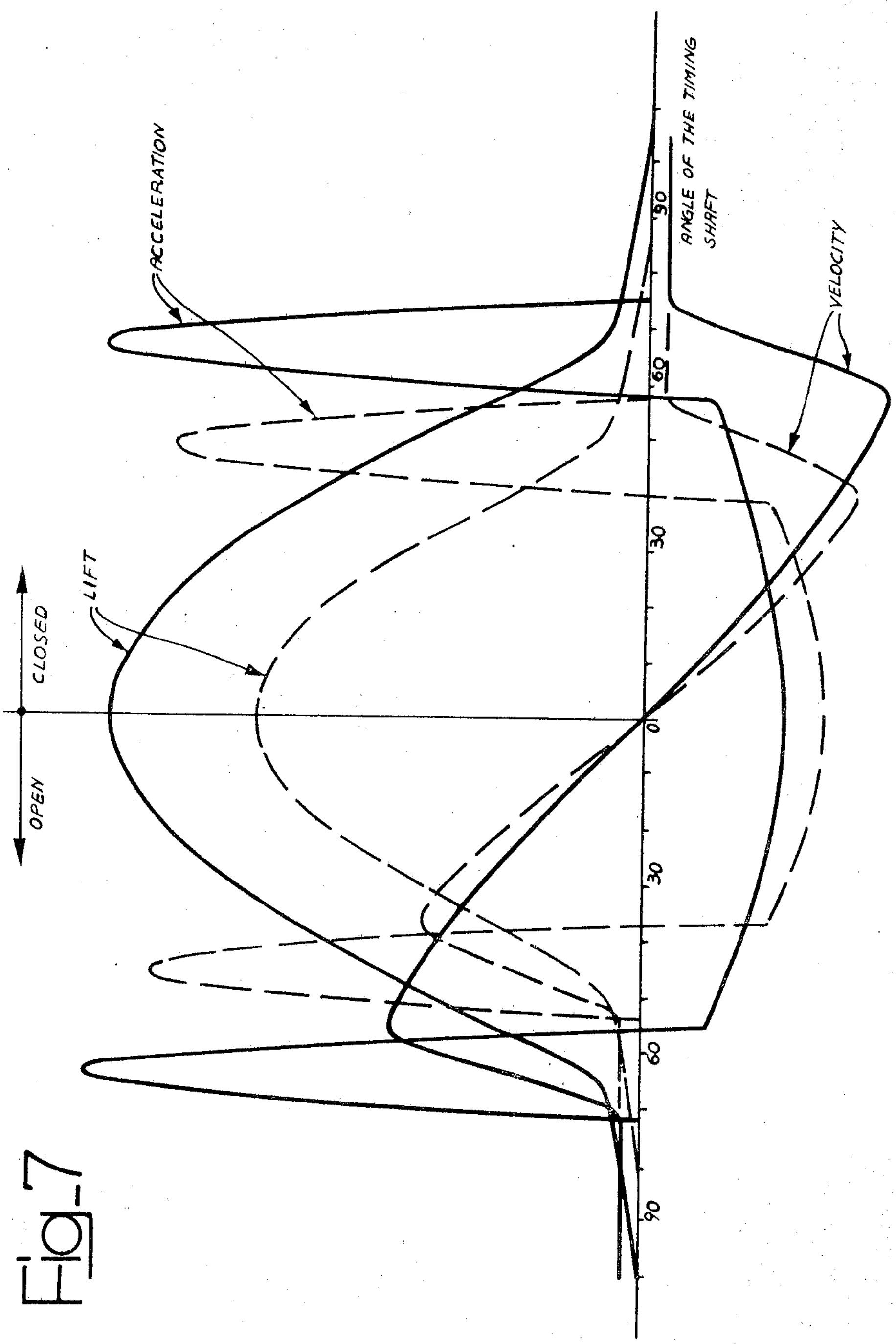

ACCELERATION
VELOCITY
LIFT
CLOSED
OPEN
ANGLE OF THE TIMING SHAFT
120
90
60
30
0
30
60
90

Fig.5




ACCELERATION
LIFT
CLOSED
OPEN
VELOCITY
ANGLE OF THE TIMING SHAFT
90
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30
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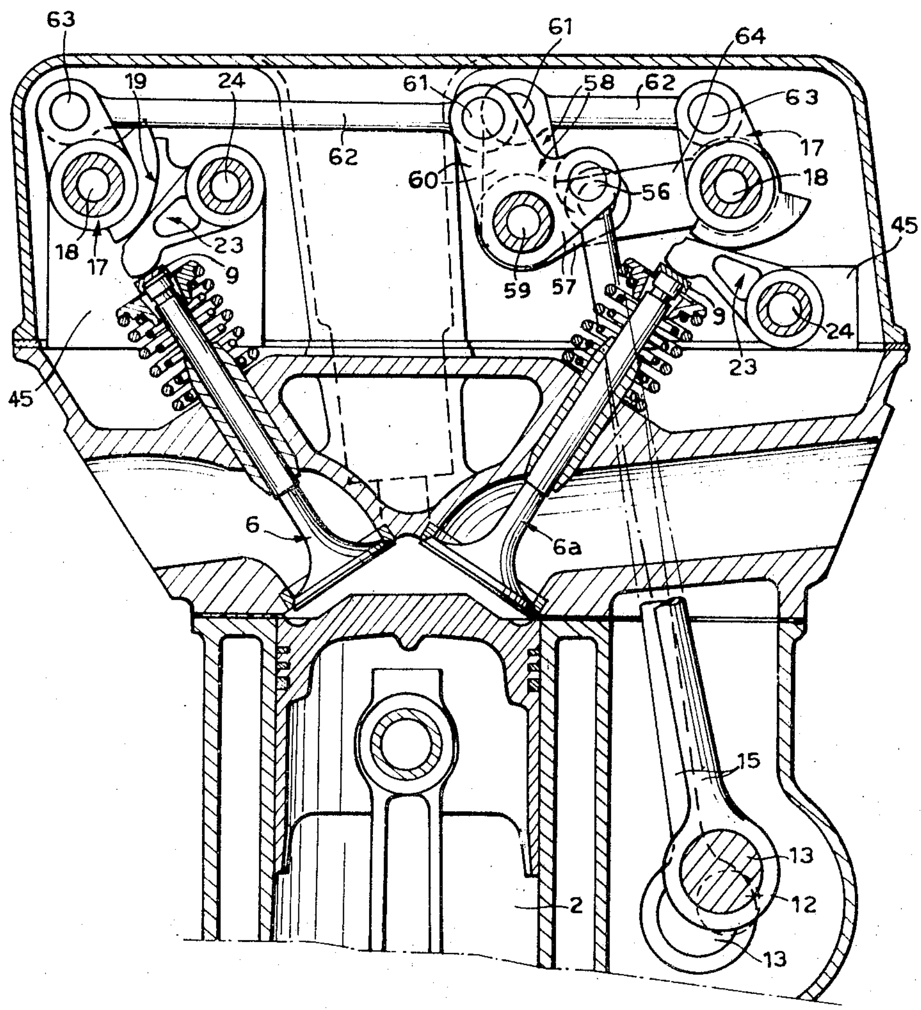
Fig_7

Fig.8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19
24
61
61
58
62
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62
60
17
18
56
45
18
17
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57
9
9
24
23
45
6
6a
15
13
12
2
13

Fig. 9

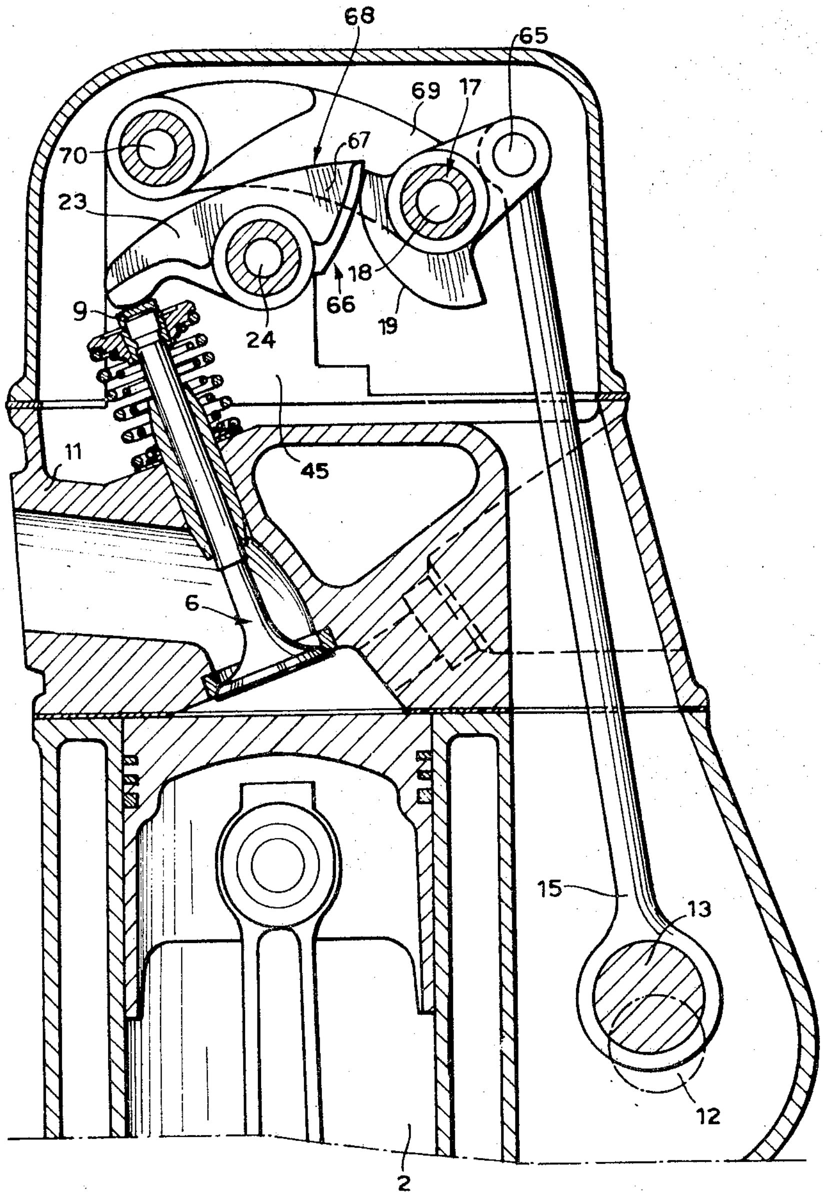
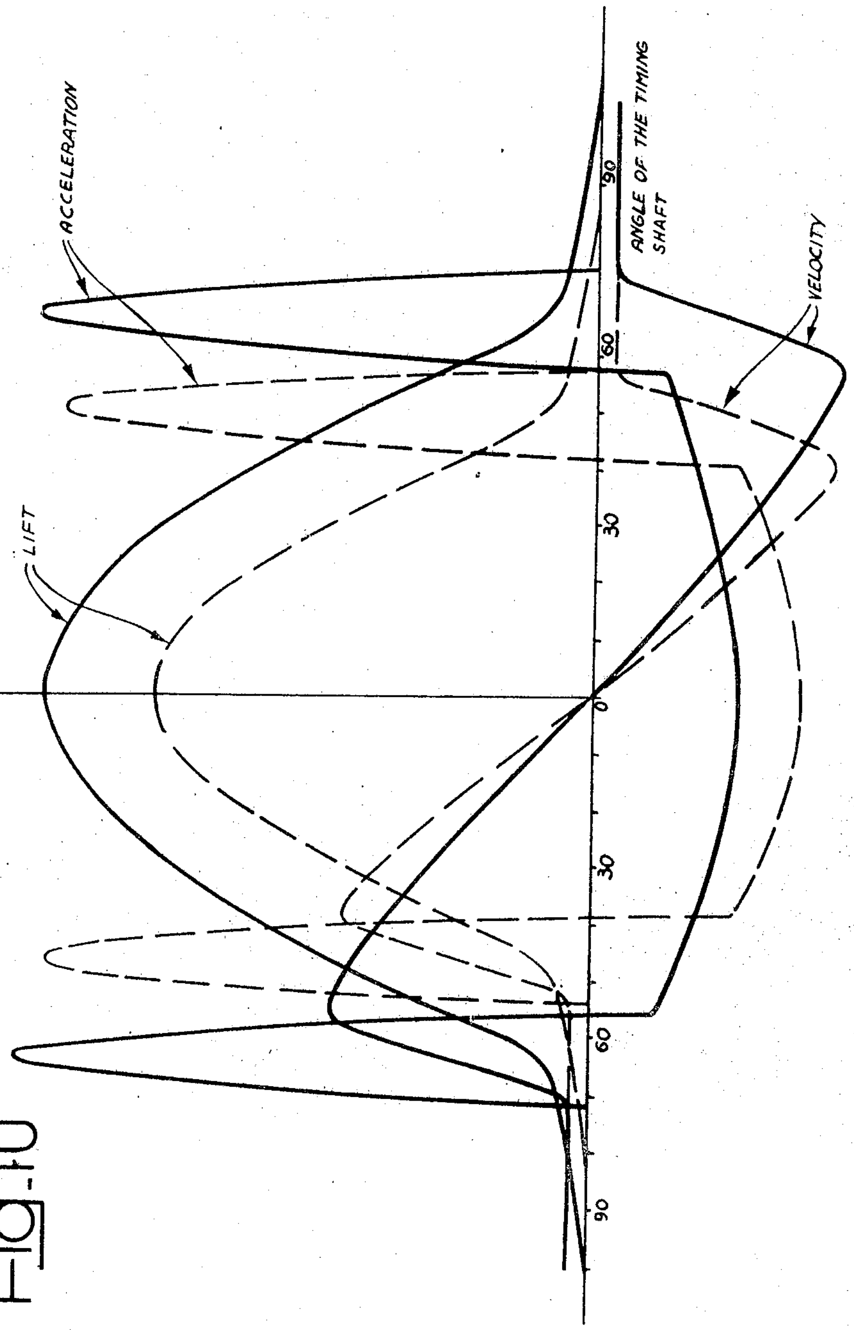


Fig.10


ACCELERATION
LIFT
CLOSED
OPEN
VELOCITY
ANGLE OF THE TIMING SHAFT
90
60
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60
90

VALVE-ACTUATING MECHANISM FOR AN INTERNAL COMBUSTION ENGINE

This invention concerns a device for driving the valve gear of internal combustion engines, from a rotary drive shaft.

It is known that in internal combustion engines with a four-stroke cycle the induction and exhaust strokes, which theoretically should last only for the time taken by the piston to make the relative strokes, are generally augmented, by causing each valve to open before the bottom dead center position of the piston, and to close after the top dead center position: that is to say, an advance and a delay are given to the induction and exhaust strokes, so selected as to allow for delays inherent in filling and exhausting the cylinder, due to the resistance to the passage of gas through the inlet and exhaust valves.

From this it follows that the total period during which each valve is open has to be large (that is, with a large advance and a large delay in valve opening and closing respectively) with large engine speeds, while this period has to be much smaller at low speeds if the engine power is to be maintained at low speed.

Since the engine, especially if it is used for motor vehicles, is required to operate over a fairly wide range of speeds, the timing used in practice is usually a compromise, giving moderate engine powers at both low and high speeds; that is to say, the performance of the engine will only be good within a fairly limited speed range.

Similar considerations also apply in the case of two-stroke engines which have operated valves.

There also arises the problem of restricting the outflow from the engine cylinders of poisonous gases made up of unburnt hydrocarbon, carbon monoxide and oxides of nitrogen; this outflow diminishes with reduction of the changeover period—that is, the period during which both the exhaust valve and the inlet valve of any one cylinder are open.

From the foregoing considerations it will be apparent that, both from the point of view of good performance (that is, high torque and low fuel consumption at all speeds), and from the point of view of low atmospheric contamination (that is, small amounts of poisonous gases in the engine exhaust), the timing of the engine valve gear ought to be variable in accordance with the speed of and the load on the engine.

To this end various mechanisms have been patented, none of which, however, either through constructional difficulties, or because of their inherent disadvantages and limitations, proved satisfactory in practice.

Among the best known of such mechanisms are the following:

2. Mechanisms which allow variation of the clearance between each cam and the respective valve tappet, or which in order to reduce timing increase the play: such arrangements have a serious disadvantage in that they lead to unacceptable cam impact velocities as soon as the tappet clearance becomes noticeable;
b. Mechanisms with frustoconical cams: each cam has a variable lift law along its width so that by shifting the cam shaft axially, different valve timings are achieved. From this it necessarily arises that the tappets engaged by the cams must be spherical, and in consequence there is punctiform contact between each cam and tappet, with very high contact pressures, if the valve springs are so proportioned as to allow high valve operating speeds;
c. Mechanisms which consist in controlling separately the opening and the closing of each valve with two cams mounted on different shafts: by varying the relative angular settings of the two shafts the timing can be varied. With this system the laws of valve opening and of closing remain constant and one has to accept a valve lift which is constant in relation to the maximum valve lift through a certain angle of rotation of the cam shafts, the constant lift section being the more extensive the greater is the variation of timing which is required. Mechanisms of this sort, whose operation seems more reliable than mechanisms (a) and (b) are, however, complicated in construction and expensive;
d. Mechanisms which produce a variation of the keying between the engine shaft and each valve cam shaft: such mechanisms are obviously applicable only to engines having twin cam shafts for the induction and for the exhaust valves, driven by a chain or a toothed belt. The aforesaid variation of the keying can be obtained, for example, by rocking arms carrying sprocket wheels which engage parts of the chain or the belt, to change the chain or belt tension. Systems of this sort only make possible a scaling down of the timing, whilst the duration of the strokes remains unchanged, and
e. Mechanisms in which there is interposed, between each cam and the respective valve tappet, a drum which shifts angularly on a radius the center of which coincides with the axis of the cam: with this system there is achieved only the same effect as in the previous case, that is, a scaling down of the timing with strokes of constant duration.

With the aim of obviating the disadvantages of the various mechanisms listed above, the present invention provides a valve actuating mechanism for an internal combustion reciprocating engine, including a rotary drive shaft, and characterized in that the mechanism includes means for converting the rotary movement of the drive shaft into rocking movement of a cam having a cam surface the contour of which includes a profiled valve operating portion, and means for varying the amplitude of the valve movement caused by said profiled cam portion in dependence upon the speed of and load upon the engine.

Further characteristic features and advantages of this invention will be apparent from the description which follows, with reference to the accompanying drawings which illustrate by way of example some practical embodiments, and in which:

FIG. 1 is a sectional elevation of a first embodiment of a valve actuating mechanism according to the invention with induction and exhaust valves closed;

FIG. 2 is a part-sectional elevation of part of the mechanism illustrated in FIG. 1, on a larger scale, with the valve open;

FIG. 3 is a diagrammatic illustration of an hydraulic device which is part of the mechanism illustrated in FIG. 1;

FIG. 4 consists of a number of diagrams illustrating graphically the variations of the lift, speed and acceleration of the valves plotted against the angular position of the rotating shaft of the mechanism illustrated in FIG. 1, for two different positions of the hydraulic device illustrated in FIG. 3;

FIG. 5 is a view in elevation of a variant of the embodiment illustrated in FIG. 1;

FIG. 6 is a diagrammatic representation of an hydraulic device forming part of the mechanism illustrated in FIG. 5;

FIG. 7 consists of a number of diagrams illustrating graphically the variations of the lift, speed and acceleration of the valves plotted against the angular position of the rotating shaft of the mechanism illustrated in FIG. 5 for two different positions of the device illustrated in FIG. 6;

FIG. 8 is a sectional elevation of a second embodiment of a valve actuating mechanism according to the invention;

FIG. 9 is a sectional elevation of part of a third embodiment of the invention, and

FIG. 10 comprises a number of diagrams illustrating graphically the variations of the lift, speed and acceleration of the valves plotted against the angular position of the rotating shaft of the mechanism illustrated in FIG. 9 for two different positions of an hydraulic device similar to that illustrated in FIG. 6.

Throughout the drawings, the same reference numerals indicate the same or corresponding parts.

FIGS. 1 and 2 illustrate the valve gear of a four-stroke internal combustion engine of the kind having overhead induction and exhaust valves arranged in a V configuration and driven by two respective overhead camshafts driven by the engine.

In FIG. 1, 1 indicates part of the internal combustion engine, showing the upper part of a cylinder 2 housing a slidable piston 3 driven by a connecting rod 4.

The cylinder 2 has at its upper end two ports **5** for induction and exhaust respectively, said ports **5** cooperating respectively with an induction valve **6** and an exhaust valve **6a**.

Each valve **6, 6a** comprises a head **7** which is shaped in the conventional manner to make sealing contact with seats ground on the respective ports **5**. Each vale **6, 6a** further has a stem **8** formed integrally with the head **7** and surmounted by a cotter **9**.

On each valve **6**, between the cotter **9** and the port **5**, and surrounding the stem **8**, there are provided a number of springs **10** (two in the example illustrated) to urge the valves **6–6a** towards their closed positions (FIG. **1**).

In order to effect opening of the valves **6, 6a** against the biasing action of the springs **10**, there are housed in the cylinder head **11** of the engine **1**, above the valves, two cam shafts **12**, shown in broken outline in FIG. **1**, one for each valve **6–6a**. The cam shafts **12** rotate at a speed which is directly related to the speed of the engine. In this particular case, where the engine is of the four-stroke type, the shafts **12** rotate at a speed equal to half that of the engine shaft.

Each of the two cam shafts **12** supports, at axially spaced positions, a number of circular eccentrics **13** the number of which is equal to the number of valves on the shafts **12**.

To each eccentric **13** there is keyed or secured a connecting rod **15** having a small end **14** to which is pivotally connected, by means of a pin **16**, a cam in the form of a rocker arm **17**. The rocker arms **17** associated with each cam shaft **12** are pivotally supported on a shaft **18**.

On the side of the shaft **18** opposite to the pivot pin **16** each rocker arm **17** has a shaped cam surface **19** having a first section **20** concentric with the axis of the shaft **18**, and a second section, hereinafter referred to as the gauge portion **21**, the shape of which determines the law of operation of the respective valves **6–6a**.

Each rocker arm **17** acts with its cam surface **19** on the upper surface **22** of an oscillating lever **23** pivotally mounted at one end on a shaft **24**. At its other end, on its lower surface, each oscillating lever **23** has a rounded surface **25**, the profile of which is an arc of a circle, which rests upon the cotter **9** of the respective valve stem **8**.

The shaft **24**, which acts as a fulcrum for the oscillating levers **23**, is carried by a support **26** which is able to rotate around the shaft **18** on which the rocker arms **17** are supported. Each support **26** (FIG. **3**) has an internal cylindrical bore **27** in which a piston **28** is slidingly mounted. The piston **28** has at its lower end an integral projection **29** which rests upon a bearing face **30** formed in the cylinder head **11**.

The upper part of the cylindrical bore **27** communicates, through a port **31**, with a duct **32a** through which flows oil under pressure from the lubrication system of the engine, supplied by a pump **33**. The flow or the pressure of the oil, or both, are regulated by a valve **34** in dependence upon the speed of the engine and the load on the latter, that is, the torque delivered thereby.

A branch duct **35** including a flow restrictor **36** branches from the duct **32a**, downstream of the pump **33**. The branch duct **35** supplies oil to a centrifugal governor **37** which in turn controls, in accordance with the speed of the engine, the oil pressure upstream of the restrictor **36**, that is, the input oil pressure at the valve **34**. The oil pressure, determined by the centrifugal governor **37**, controls the opening of the valve **34**.

The opening of the valve **34** is at the same time controlled by a vacuum-operated capsule **38**, communicating with the induction manifold of the engine, shown diagrammatically at **39**. The capsule **38** controls the valve **34** in dependence on the load on the engine.

In the mechanism thus far described, given the considerable amplitude of angular oscillation of the rocker arms **17**, which is necessary to be able to accommodate a cam surface **19** which gives the necessary movement, the oscillation of each rocker arm **17** is effected by means of a desmodromic drive system comprising an articulated quadrilateral linkage formed by the respective eccentric **13** which acts as a crank, the connecting rod **15** and the rocker arm **17**.

By this means, the reactions of the valve springs **10** in the negative acceleration parts of each piston stroke can be reduced compared with valve-actuating mechanisms generally used. Moreover a more rigorous law of movement for the valves is obtained insofar as it is influenced by errors in construction of a single cam only.

To justify the introduction of the oscillating lever **23** between the rocker arm **17** and the cotter **9** of the respective valve **6**, it should be noted that drive of the valve **6** directly via the rocker arm **17** would require excessive dimensions for the rocker arm, if one wished to obtain the law of acceleration usually adopted for the valves of reciprocating engines.

In order to obtain the most favorable law of movement for each valve **6**, while keeping the dimensions of each rocker arm cam **17** and of its cam surface **19** within acceptable limits, it has been found preferable for the connection between the rocker arm **17** and the valve **6** to be made through the intermediary of oscillating lever **23**, placed in such a way that the area of contact of the cam surface **19** with the upper surface **22** of the oscillating lever **23**, conjugate with it, is shifted towards the axis of rotation of the oscillating lever **23** in order to increase of the lift. This leads to angular velocities of the oscillating lever **23** contrary to those of the rocker arm **17** during the whole valve-lifting movement, and to an obvious reduction in the overall dimensions of the device.

In the device described, since the shaft **24** is carried by the movable supports **26**, which in their turn are pivotable about the shaft **18**, the shaft **24** can shift through a circular arc having a radius equal to the distance between the axes of the shafts **18** and **24**, and it is precisely this shifting which achieves the required variation of the valve timing.

In fact, upon displacement of the shaft **24** towards the plane **30**, reduction in the amplitude of the timing is obtained, since the beginning of the gauge portion **21** of the cam surface **19** then comes into contact with the oscillating lever **23** at a delayed angular position of the drive shaft **12**, and, consequently the last part of the gauge portion **21** is not utilized.

In FIGS. **1** and **2** the oscillating lever **23** is shown as a continuous line in the position which gives the maximum timing, and in broken outline in the position which gives the minimum timing.

If the center of curvature of the rounded surface **25** of the oscillating lever **23**, when the associated valve **6** is closed, coincides with the axis of the shaft **18**, then the operating play or tappet clearance is not influenced by the position of the axis of the shaft **24**; conversely, it is possible to obtain an operating play or clearance which is linearly variable with the valve timing by suitably displacing the center of curvature of the rounded surface **25** from the axis of the shaft **18**. This can be useful if, for example, the engine is subject to thermal expansions of the kinematic fall of the valve **6** bringing about variations in the play which are proportional to the engine speed and load.

In the device illustrated in FIGS. **1** and **2** it should also be remarked that the position of the cam shafts **12** is of considerable importance. The placing which is shown permits advantageous exploitation of the low ratio between the length of the connecting rod **15** and the crank arm constituted by the radius of eccentricity of the eccentric **13**; in fact the lift of the valve is effected when the crank mechanism comprising the eccentric **13** and the connecting rod **15** is situated around the top dead center position which corresponds to the greatest angular accelerations of the rocker arm **17**.

Moreover, also as a result of the low value of the ratio between the length of the connecting rod **15** and of the crank arm, with clockwise rotation of the cam shaft **12**, the angle through which the shaft **12** has to rotate to effect the opening of the valve **6** is in fact less than that necessary to bring about closing of the valve; hence on closing of the valve there are lower speeds and accelerations than on opening of the valve.

In this manner it is possible to reduce the tendency towards valve bounce; moreover lower impact speeds are obtained between the head **7** of the valve **6** and the respective seat upon valve closure, such as one tries to get in normal valve drives to reduce noise to a minimum.

With regard to the hydraulic device for effecting rotation of the support **26**, the oil under pressure which enters the bore **27** is the same as is used to lubricate the engine. In fact the oil supply duct **32***a* connected to the bore **27** is branched from a duct **32** which supplies the lubricating oil to the engine; the quantity and the pressure of the oil, or both, are governed by the valve **34**, comprising a governor, which is attached to the device and which senses the speed and the load of the engine, as hereinbefore described.

With a drive of this kind one can regulate, in an entirely independent manner, the relative timing of the induction and the exhaust valves, since there are two separate drives for these valves.

In the practical embodiments just described and in those described hereafter, variation of the timing is achieved, as has been seen, by means of an hydraulic drive actuator. It will be appreciated, however, that the variation of the timing can be effected by means of mechanical, pneumatic or electrical actuators responsive to the speed and the load of the engine.

The diagrams illustrated in FIG. **4** show clearly the effect of the relationship which exists between the length of the connecting rod **15** and the aforesaid crank arm in the device illustrated in FIGS. **1** and **2**. As can be seen, the graphs showing the variations of the lift, velocity and acceleration of the valve are asymmetrical. Specifically, the absolute values of the velocity and the acceleration are greater when the value is open, with the advantage of obtaining, when the valve closes, low noise and efficient working of the cooperating members.

In FIG. **4** the continuous curves refer to a wide timing position, whereas the dashed curves refer to a position corresponding to a narrower timing.

In the variant of the FIG. **1** embodiment illustrated in FIG. **5**, a single rotating cam shaft **12** drives both the valves **6**; this cam shaft **12** is housed in the engine cylinder block on one side of the row of cylinders **2**.

Mounted on the rotating shaft **12**, and integral with it, are arranged, for each cylinder **2**, two circular eccentrics **13** each of which is connected to or integral with a respective connecting rod **15** which, in this case, is of considerable length. In fact the small end of the connecting rod **15** is linked, by means of a pin **16**, to a rocker arm cam **17** placed near the top of the cylinder head **11** of the engine **1**.

Each rocker arm **17** is mounted on a shaft **18** which is arranged to rotate within a number of supports **45** fixed to a bearing face **30** in the head **11**, by means of screws **46** (see FIG. **6**).

The rocker arm **17**, in contrast with that shown in FIGS. **1** and **2**, has its shaped cam surface on the same side of the shaft **18** as the pin **16**.

A support arm **47** is rotatably supported on the shaft **18** and in turn supports a shaft **24** which acts as a fulcrum for an oscillating lever **23**.

The support arm **47** is formed on the opposite side of the shaft **24** from the shaft **18** with a nose **48** which bears upon a trapezium-shaped upper surface **49** of a piston **50** of an hydraulic actuator device **51** illustrated in FIG. **6**. The piston **50** is mounted slidingly in a cylindrical bore **52** in the device **51**, the bore **52** being closed at its lower end by the bearing face **30**.

Oil under pressure is supplied to the lower part of the bore **52** through an aperture **53**. This oil is supplied, as in the embodiment illustrated in FIG. **3**, from the lubrication system via a duct **32***a*. In this case also the pressurized oil is circulated by a pump **33**, and a valve **34** governs the delivery and the pressure of the oil in accordance with the speed of the engine and the load to which the engine is subjected, that is, the torque delivered by the engine.

The upper surface **49** of the piston **50** has, on diametrically opposite sides with respect to the axis of the piston **50**, two inclined plane surfaces, shown as **54** and **55**, on which the noses **48** associated with the induction valve **6** and the exhaust valve **6***a* bear respectively.

In the valve mechanism just described variation of the timing in dependence upon the speed of and load on the engine is effected, as in the device illustrated in FIGS. **1** and **2**, by rotation of the shaft **24**, which acts as a fulcrum for the oscillating levers **23**, around the axis of the shaft **18**.

In this case, in view of the arrangement of the members, it is convenient to achieve the variation of timing with a single actuator device **51**, as illustrated in FIG. **6**.

It will be observed that in this case too, where a single cam shaft is provided, one can regulate independently the induction and exhaust timing. This can be effected by varying the inclination of the surfaces **54** and **55**, or by making the nose **48** associated with the induction valve **6** of a different length from the nose **48** associated with the exhaust valve **6***a*.

In the mechanism illustrated in FIG. **5** the cam shaft **12** is situated in the cylinder block, and in view of the consequent considerable length of the connecting rods **15**, difference in the ratios between the length of each connecting rod **15** and the effective crank arm connected thereto is scarcely detectable. As a result the laws of movement of the valve opening and closing operations will be almost symmetrical as can be seen from the diagrams illustrated in FIG. **7**.

In view of their length, it is more advantageous to make the connecting rods **15** work as tie rods; this can be done as illustrated in FIG. **5**, by making each rocker arm **17** operative to lift its respective valve when the crank gear consisting of the eccentric **13** and of the connecting rod **15** is situated around the bottom dead center.

The embodiment illustrated in FIG. **8** relates to a four-stroke internal combustion engine which has induction valves **6** and exhaust valves **6***a* arranged in a V formation. A single rotating cam shaft **12** is housed in the cylinder block to one side of the row of cylinders **2**, and governs the opening and the closing of said valves **6** and **6***a*.

Analogously to the embodiment of FIG. **5**, on the rotating cam shaft **12** and integral with it there are mounted or formed, for each cylinder **2**, two circular eccentrics **13**, on each of which a connecting rod **15** is mounted.

The small end of each connecting rod **15** remote from the eccentric **13** is joined by means of a pin **56** to one end of one arm **57** of a small bellcrank lever **58** acting as a rocker. The levers **58**, associated with the two connecting rods **15**, are mounted on a common shaft **59** which acts as a fulcrum.

To the end of the other arm **60** of each lever **58** there is pivotally joined, by means of a pin **61**, one end of a link **62** which is pivotally connected at its other end, by means of a pin **63**, to the rocker arm **17**. The rocker arm **17** has a cam surface **19** which acts on an oscillating lever **23** which in turn acts upon the cotter **9** of the respective valve **6**.

The shafts **18** and **24**, acting as respective fulcra for the rocker arm **17** and the oscillating lever **23**, are carried by supports **45** attached to the cylinder head **11** of the engine **1** by means of screws, not shown.

The shaft **59**, acting as a fulcrum for the rocker levers **58** is supported by a support **64** mounted rotatably upon one of the shafts **18**, which relates to the cam **17** driving one of the two valves, for example, that carrying the rocker arms **17** associated with the exhaust valves **6***a*.

The shaft **59** is supported by the piston of an hydraulic actuator device of the type illustrated in FIGS. **3** and **6**; by means of this device the shaft **59** can be displaced in dependence upon the speed of and load on the engine **1**.

The result of the displacement of the shaft **59** is a variation of the effective length of the connecting rod **15** and the link **62** and, therefore, a different profile of the cam surface **19** of the rocker arm **17**.

A disadvantage of this arrangement is that is comprises more moving parts than the embodiment described previously. Moreover the embodiment just described has a single actuator device, and from this it follows that, while the drive is achieved with greater ease, there is the disadvantage that the same variation of stroke for both the induction and exhaust valves. It is clear that to avoid this disadvantage it would be necessary to have the rocker levers **58** rotatable about fulcra on two separate shafts and to displace their axes of oscillation with independent actuators.

It should also be noted that in the embodiment described above, the oscillating lever 23, the pivotal axis of the shaft of which remains fixed, is not strictly indispensible and could be replaced by a flat or cylindrical cotter, but its use makes it possible to reduce the dimensions of the rocker arm 17.

Naturally the laws of opening and closing of the valves are almost symmetrical, in view of the length of the connecting rod 15; as a close approximation one can say that the diagram illustrated in FIG. 7 is valid for this case as well.

The embodiment illustrated in FIG. 9 relates to a four-stroke internal combustion engine 1 with induction and exhaust valves 6, 6*a* in line. In this case, also, the single rotating shaft 12 arranged in the cylinder block on one side of the cylinder 2, drives the opening and closing of the valves referred to above.

The shaft 12 carries, for each valve, the integral circular eccentric 13 on which the lower end of the connecting rod 15 is rotatably mounted. The rocker arm 17 is pivotably connected to the upper end of the connecting rod 15 by means of a pivot pin 65, the cam 17 being mounted for rocking movement on a shaft 18.

The cam 17 has a cam surface 19 which acts on a contoured surface 66 made on a face of one arm 67 of a rocker 68 pivotally mounted on a shaft 24. The other arm of the rocker 68 consists of the lever 23 which acts upon the cotter 9 of the respective valve 6.

The shaft 18 on which the rocker arms 17 are rotatably mounted is supported by an oscillating arm 69 rotatably mounted upon a shaft 70. The shaft 70 and the shaft 24 are in turn rotatably supported upon the support 45 which is fixed to the cylinder head 11 of the engine 1 by means by screws (not shown).

The shaft 18 on which the rocker arms 17 are mounted bears upon an hydraulic actuator device of the same type as those illustrated in FIGS. 3 and 6 and is thus displaceable in accordance with the speed of and load on the engine 1.

In FIG. 9 is illustrated the position of the shaft 18 corresponding to the maximum cam lift; for a displacement of the shaft 18 downwardly a reduction in timing results, so that the said displacement causes a rotation of the rocker arm 17 around the pivot pin 65 and hence a given portion of the cam surface 19 comes into operation with a delay angle equal to the angle through which the rocker arm 17 has rotated around the pin 65.

The contoured surface 66 made upon the face of the arm 67 of the rocker 68 must have its center of curvature coincident, when the respective valve 6 is closed, with the axis of the shaft 70, if the valve operating clearance or play is to be constant with variations in the timing. Analogously with the mechanism illustrated in FIG. 1, a displacement of the center of curvature of the contoured surface 66 of the axis of the shaft 70 causes the valve operating play to vary linearly with valve timing.

It should be noted that the displacement of the axis of oscillation of the rocker arm 17, apart from inducing rotation of the rocker arm 17 itself, with consequent change in the angle of opening of the valve 6, also causes a variation in the effective length of the arm 67, as a result of which the loss of maximum valve lift, corresponding to reduction of the valve opening angle, is less than it is in the other devices previously described herein.

The diagrams illustrated in FIG. 10 show the laws of movement of the valve in two different positions of the shaft 18. In view of the length of the connecting rod 15 it is evident that the diagrams are practically symmetrical.

It will be understood that details of practical embodiments of the invention can be widely varied from those described and illustrated herein by way of example, without departing from the scope of the appended claims.

What is claimed is:

1. In a valve-actuating mechanism for an internal combustion engine, of the kind having a rotary drive shaft, a plurality of rocker arms, means converting rotary movement of the shaft into rocking movement of the rocker arms, and respective valves operable by the rocker arms, the improvement comprising the rocker arm having a cam surface the contour of which includes a profiled valve operating portion, means varying the amplitude of the valve movement caused by said profiled valve operating portion, means varying the amplitude of the valve movement caused by said profiled cam portion in dependence upon the speed of and load upon the engine including an oscillating lever interposed between the cam surface and the valve effective to amplify the valve displacement caused by the valve operating portion of the said cam surface, and means effecting a relative rotation between the rocker arm and the oscillating lever, said rotation having a magnitude determined by the speed of and load on the engine, wherein the means effecting the relative rotation between the rocker arm and the oscillating lever displace the fulcrum of the oscillating lever along an arcuate path which is centered upon the axis of rocking movement of the rocker arm.

2. Mechanism according to claim 1, wherein the fulcrum of the oscillating lever is carried by a movable support which is in turn rotatably mounted on the shaft bearing the rocker arms, and including means rotating the movable support in dependence upon the speed of and the load on the engine.

3. Mechanism according to claim 2, including an hydraulic actuator device arranged to cause rotation of the support in dependence upon the speed of and load on the engine.

4. Mechanism according to claim 3, wherein the hydraulic actuator device comprises means defining a bore in the movable support, a piston slidably mounted in the bore, a fixed reaction surface engaging the piston, and means supplying said bore with oil the pressure of which is dependent upon the speed of and load on the engine.

5. Mechanism according to claim 4, wherein said means supplying pressurized oil comprises a pump in the lubrication system of the engine.

6. Mechanism according to claim 4, including a valve regulating the pressure of the oil supplied to said bore, a centrifugal governor controlling the operation of said valve in dependence on the engine speed and a vacuum-operated capsule, connected to the induction system of the engine, controlling the operation of said valve in dependence on the engine load.

7. Mechanism according to claim 1, wherein the point of contact between the cam surface and the oscillating lever shifts towards the fulcrum of the oscillating lever upon increase of the amplitude of valve displacement.

8. Mechanism according to claim 1, wherein the angular speeds of the rocker arm and of the oscillating lever are in opposite directions throughout the whole duration of the displacement of the valve (6).

9. Mechanism according to claim 1, wherein the oscillating lever has a rounded surface comprising an arc of a circle by which said lever contact the respective valve.

10. Mechanism according to claim 9, wherein the center of said rounded surface coincides with the axis of oscillation of the rocker arm.

11. Mechanism according to claim 9, wherein the center of the said rounded surface is displaced with respect to the axis of oscillation of the rocker arm.

12. Mechanism according to claim 1, including respective shafts mounted in the cylinder head of the engine, and respective crank means, for example an eccentric and a connecting rod connecting the induction and exhaust valves to the respective shafts, opening of the respective valves being effected when the crank means are in top dead center positions.

13. Mechanism according to claim 2, including a cam shaft on which the rocker arms are rotatably mounted, the movable support comprising an oscillating arm mounted for rotation about the cam shaft, and further including a shaft carried by the oscillating arm and acting as a fulcrum for the oscillating lever, means being provided for rotating said arm in dependence upon the speed of and load on the engine.