United States Patent

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[33] **Italy**
[31] **52410 A/69**

[54] **VARIABLE CAMSHAFT MECHANISM**
5 Claims, 5 Drawing Figs.

[52] **U.S. Cl.** **123/90.17,** 64/25, 74/568
[51] **Int. Cl.** **F01l 1/34**
[50] **Field of Search** 123/90.15, 90.17; 64/25; 74/568

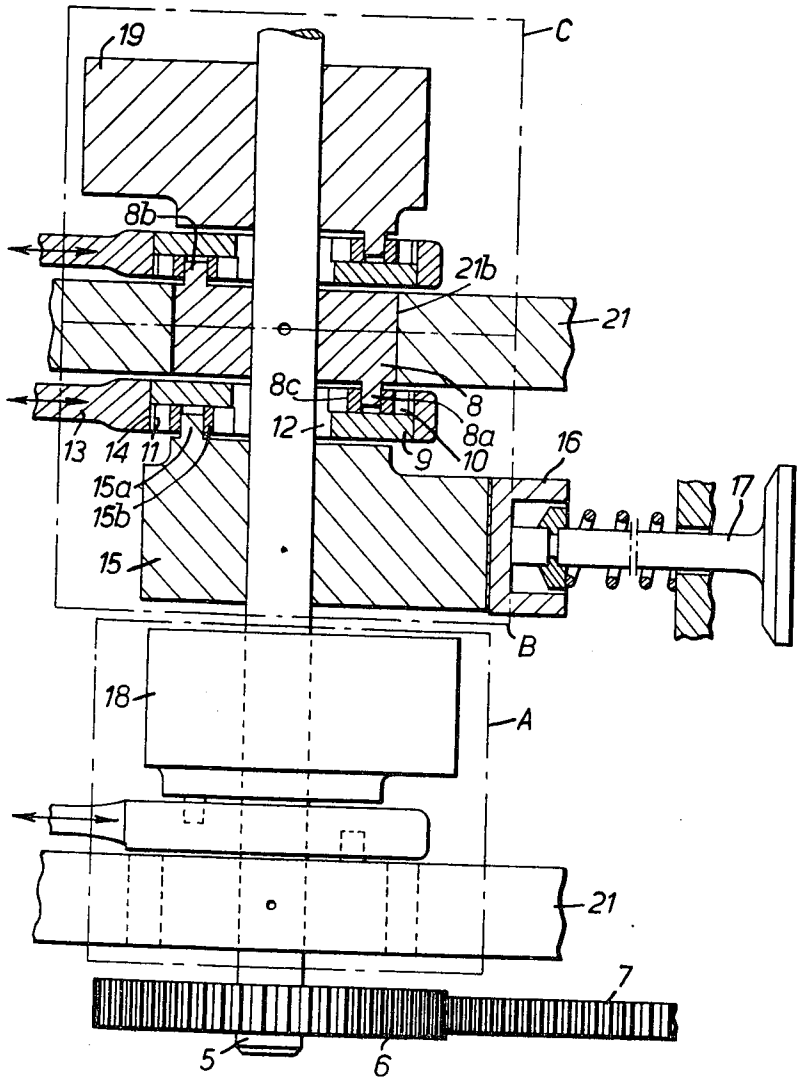
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ABSTRACT: A device, particularly for controlling the opening and closing of valves of an internal combustion engine, comprises a camshaft on which a plurality of cams are mounted for rotation relative to the shaft. Each cam is driven by the shaft through an intermediate member which rotates about an axis eccentric to the shaft axis. A drive member rotatable with the shaft has a pin which engages in a radial slot on one side of the intermediate member, and likewise the cam has a pin engaging in a radial slot on the other side of the intermediate member, these slots being preferably 180° spaced apart.



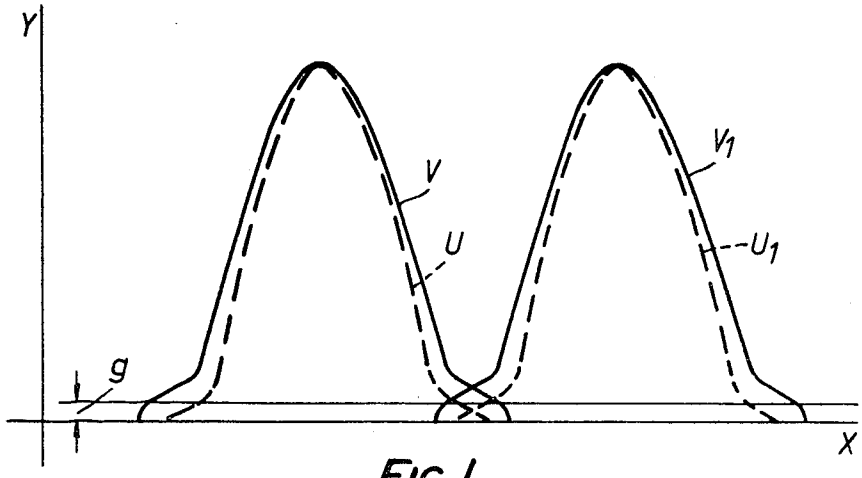


FIG. 1.

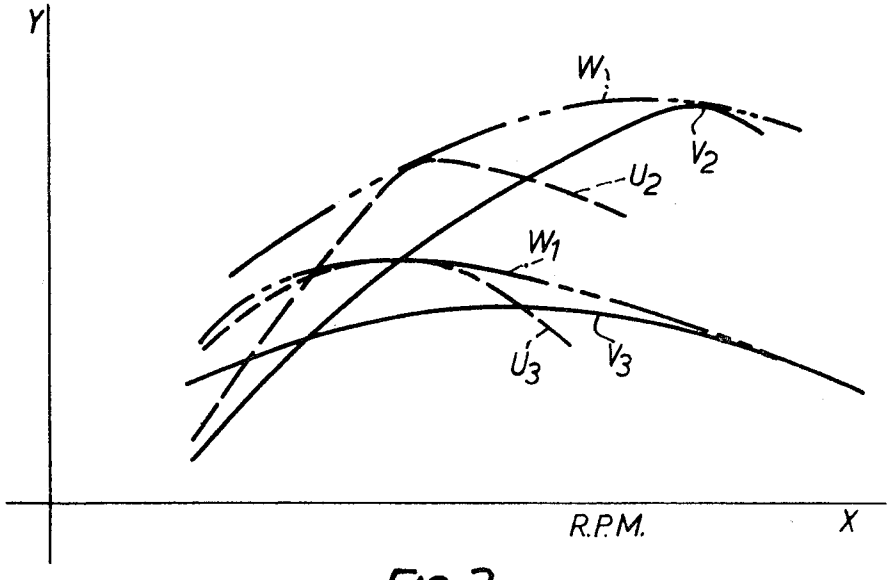


FIG. 2.

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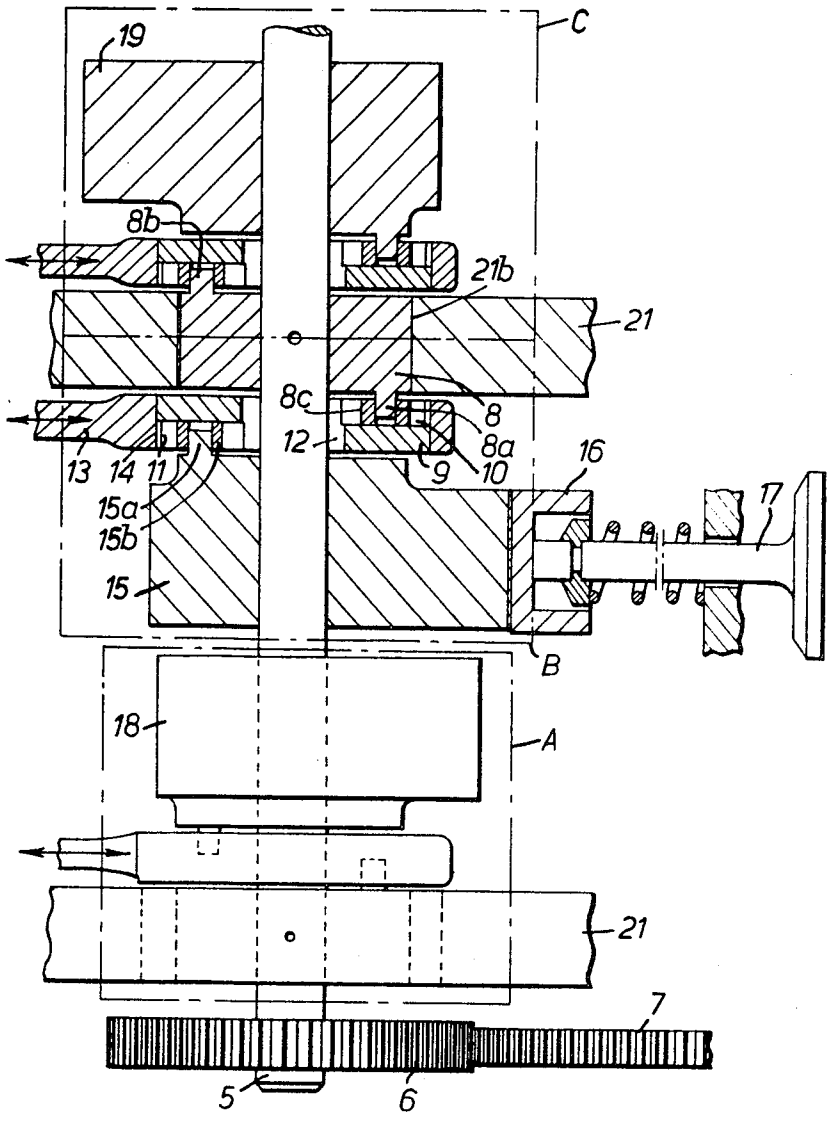


FIG. 3.

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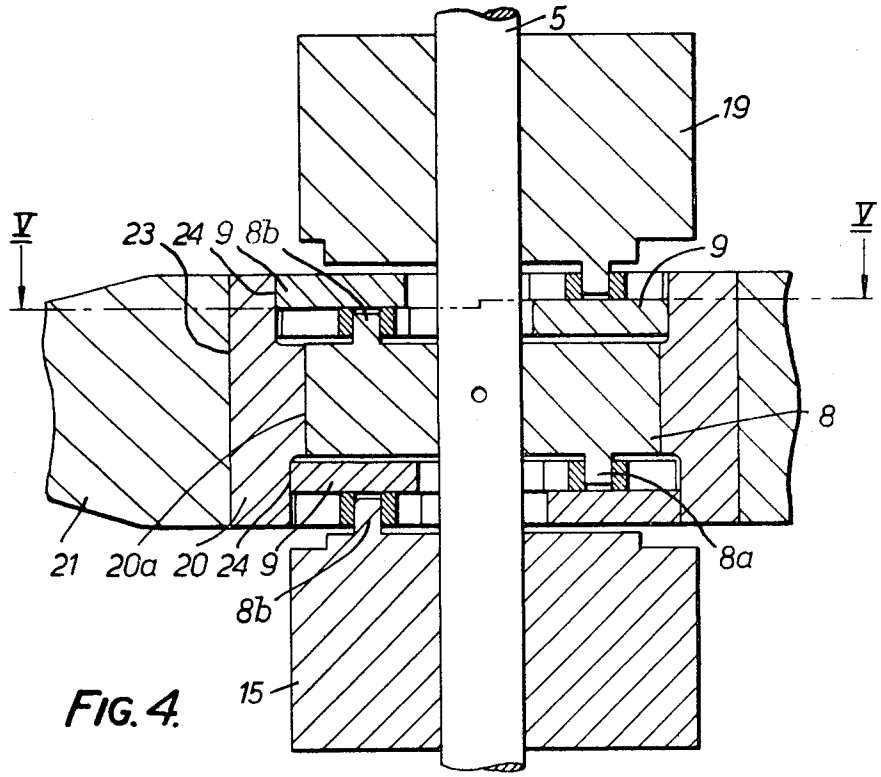


FIG. 4.

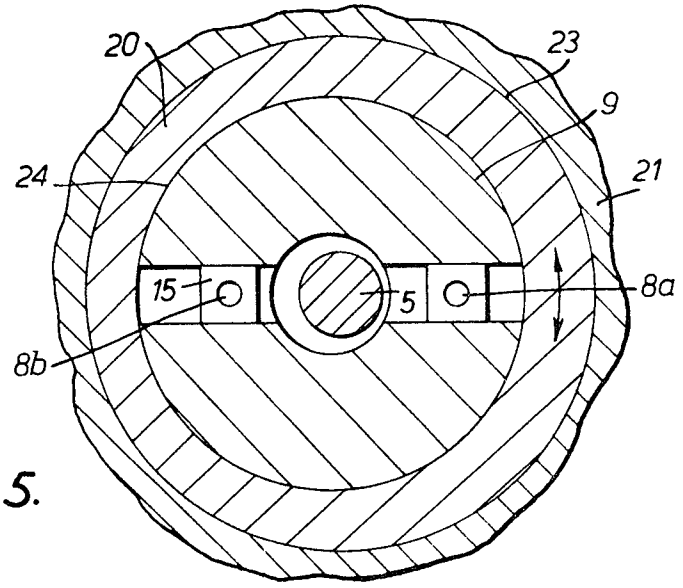


FIG. 5.

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VARIABLE CAMSHAFT MECHANISM

This invention relates to a device for moving a cam relative to its driving shaft, and is particularly applicable to an internal combustion engine to vary the movement of the cams which control the inlet and exhaust valves of the engine.

It is known that the lift diagram for internal combustion engine valves is of special importance when determining engine performance requirements at various rotation speeds. Experience has shown, for example, that taking the maximum lift of the intake valve as a constant—the initial and final phases and the gradient of the lift diagram that will produce optimum performance will vary in dependence on the number of revolutions per minute. For instance, the time at which the intake valve begins to open must be calculated so that the intake valve will be sufficiently wide open when a pressure drop occurs in the cylinder and so that it will be fully closed in time to prevent the "rejection" phenomenon that occurs when part of the mixture drawn onto the cylinder is ejected. In practice it has been shown that since optimum phasing is linked with the dynamic properties of gases within the suction manifold and cylinder, it will vary depending on engine speed. Similar considerations apply to the exhaust valve in that the correct phasing of this valve is associated with problems both of performance and of reducing atmospheric pollution caused by the discharge of partially unburnt products.

In addition, the laws governing the varying rates of valve lift and fall have differing effects on the rate at which the cylinder is filled and on evenness of combustion; these laws should therefore be varied in the light of the desired engine speed to achieve optimum performance.

The valve lift diagram is usually determined by the design of the camshaft cams, which may vary in different engines. These cams impart motion either indirectly, through rods and valve rockers or directly, in the case case of an overhead camshaft.

Since the cam profiles are constant, they will produce valve lift diagrams that are consistently identical, both in shape and in phase, irrespective of the speeds at which the engine rotates. This means that the timing system obtained with traditional mechanisms cannot be equally effective at different engine speeds.

One object of the present invention is to provide a device which will enable a cam to be moved relative to its camshaft in such a way that the shape or phase, or both shape and phase, of the curve of cam follower displacement against shaft rotation can be varied. As applied to an internal combustion engine, it is a further object of the invention to vary the movement of the valve-control cams in such a way as to obtain improved torque and power values at certain engine speeds.

Yet another object of the invention is to produce a device for dynamic, automatic shape or phase regulation of cams controlling internal combustion valves which will considerably reduce the percentage of unburnt waste gases, thus reducing atmospheric pollution.

According to the present invention there is provided a device for moving a cam relative to its driving shaft comprising a drive member rotatable with the shaft, an intermediate member mounted for rotation eccentrically with respect to the shaft, the drive member being connected to the intermediate member at a position on the drive member spaced a predetermined distance from the axis of the shaft, and a cam rotatable about the axis of the shaft but having a drive connection to the intermediate member at a position thereon spaced about the shaft axis from the connection between the intermediate member and the drive member.

The present invention also provides a device for moving a cam relative to its driving shaft comprising a drive member rotatable with the shaft, an intermediate member mounted in an external bearing which is eccentric with respect to the shaft, the shaft extending through an opening in said intermediate member dimensioned to allow limited movement of said bearing to vary a parameter of the eccentricity, a cam coaxial with the shaft and rotatable relative thereto, a first coupling between the drive member and the intermediate

member at a first position spaced from the shaft axis, a second coupling between the intermediate member and the cam at a second position angularly spaced from the first position with respect to the shaft axis, the two couplings being so spaced from the shaft axis that they are at varying distances from the axis of the intermediate member during operation, each said coupling having a movable connection with the intermediate member to permit the variation in its distance from the axis of the intermediate member.

According to another aspect, the present invention provides a device for use in an internal combustion engine to control the opening and closing of the inlet and exhaust valves thereof, the device comprising a camshaft, a plurality of cams mounted on said shaft for rotation relative to the shaft, each cam having a drive connection from the shaft which includes means operative over one part of the cam revolution to drive the cam faster, and over another part of the same revolution to drive the cam slower, than the rotational speed of the camshaft.

The invention will now be particularly described with reference to the accompanying drawings in which:

FIG. 1 illustrates typical valve lift curves of an internal combustion engine, the abcissa X representing crankshaft angular rotation and the ordinate Y representing valve lift in mm. (g representing phasing clearance);

FIG. 2 illustrates typical curves of torque (in kg.)/engine speed (in r.p.m.) and power (in hp.)/ engine speed;

FIG. 3 is an axial section through one embodiment of device according to the invention;

FIG. 4 is an axial section through a modification of the embodiment of FIG. 3;

FIG. 5 is a section on the line V—V of FIG. 4.

As seen in FIG. 1, curve u represents the intake valve where opening is not very advanced, and ul represents the exhaust valve whose closure is not greatly retarded. These curves relate to V and VI, which represent the same valves opening earlier and closing later. The values of power and torque resulting from the characteristics of FIG. 1 are shown, in relation to engine speed, by the curves of FIG. 2. It will be noted that curves u and ul correspond qualitatively to powers and torques $u2$ and $u3$, which are higher at a lower r.p.m., whereas curves V and V1 produce the best performance, V2 and V3 respectively, at a higher r.p.m.

When dealing with the problem of atmospheric pollution, a shift from one type of curve to another may affect the quantity of unburnt products in exhaust gases emitted at low engine speeds.

To achieve optimum engine performance, therefore, as the engine speed increases it should be possible to move continuously from diagrams u and ul to diagrams V and V1, producing power and torque curves similar to W and W1 as a result of varying the curves at different engine speeds.

This aim obviously cannot be achieved with the use of traditional timing mechanisms since, as is known, motion is imparted to the valves solely by the fixed cam profile, and this cannot be altered while the engine is running.

The aforegoing analysis is of course merely an outline of the principles involved and is not a comprehensive survey of the problem. For reasons of brevity, certain important factors have not been mentioned or have merely been touched upon, e.g., inertia of fluid masses moving along the pipes; mechanical properties of return springs, etc.

In the present state of the art, various mechanisms are used to overcome the valve lift problems in an attempt to optimize the laws of variation according to engine rotation speed. These mechanisms incorporate variable arm levers, special valve rockers, etc., each of which has known advantages and disadvantages which it would be superfluous to enumerate.

The device of the present invention, on the other hand, can be incorporated in the traditional timing mechanisms of internal combustion engines and can achieve continuous variations in cam phase and in valve lift in accordance with engine requirements.

Basically, the device comprises a collar, forming an integral part of a camshaft, from which collar a pin extends into a radial groove cut in one face of a disc. The disc is fitted on to the camshaft, with ample radial clearance, and has on its opposite face a second groove, at an angle of 180° to the first groove, in which engages a pin fixed to the cam whose movement is being controlled. This cam is mounted on the camshaft for rotation relative thereto. The disc is supported in a bearing which itself can be moved to vary the eccentricity of the disc with respect to the camshaft. By reason of the differing positions in which this disc can be placed within its plane of rotation and thus by reason of the different positions assumed by the groove engaged by the pin of the collar and the groove engaged by the pin of the cam the angular velocities imparted to the cam at different rotational positions can be varied.

The mechanism will now be considered in more detail with reference to FIGS. 3 to 5.

In FIG. 3, the mechanism is shown three times. For ease of reference, each of the three different versions is encircled by a broken line rectangle marked A, B or C. In rectangle A is shown the mechanism applied to a single cam. It will be noted that rectangles B and C have one side in common. They represent, in cross section, two mechanisms, each applied to one cam. In this case, the two mechanisms have certain components in common, as will be seen in the illustration and as described below, with a view to reducing cost.

A camshaft 5 which drives all the cams, is rotatable by conventional means such as the toothed wheel 6 and belt (or chain) 7. The cams are not integral with shaft 5, whose section is constant, but rotatable with respect to shaft 5. Their movement is restricted in a longitudinal direction.

The mechanism shown inside rectangle B will be particularly considered since those in rectangles A and C operate in the same way.

Collar 8 is rigidly attached to shaft 5, and it incorporates two pins, 8*a*, 8*b*. In precisely the same way as pin 8*a*, 8*b* helps to operate the device shown in rectangle C. It will therefore not be discussed in this description of the identical mechanism illustrated in rectangle B. Pin 8*a* slides and rotates in the radial groove 10 of a disc or intermediate member 9, facilitated by a slider 8*c* made of antifriction material. On the other side of disc 9 is a second radial groove 11, which is symmetrical to and preferably at an angle of 180° to the first groove.

The central hole 12 of disc 9 is wide and the disc does not touch the surface of camshaft 5, but is free to move into positions eccentric with respect to the camshaft. An arm 13 forms a bearing 14 which supports the disc 9 for rotation, the arm being movable to move the disc perpendicularly to the plane of the drawing. The component of this movement, in the plane of the drawing, is indicated by arrow *y*, but of course a component of movement can exist perpendicular to the plane of the drawing. Pin 15*a* of cam 15 can rotate in and slide along radial groove 11 in disc 9, and cam 15 is freely rotatable on shaft 5 but cannot slide in a longitudinal direction. Cam 15 controls tappet 16 of valve 17. Here again, a slide 15*b* is placed between pin 15*a* and groove 11 to reduce wear.

The shaft 5 is supported, through collar 8 and the collars associated with other cams, by the engine block 21 or other stationary part of the engine.

The device functions as follows. The shaft 5 is rotated around its axis by the drive mechanism, i.e., the toothed gear 6 and belt (or chain) 7. The shaft rotates the collar 8, which in turn rotates in and is supported by a bearing 21. The pin 8*a*, projecting from the collar 8, engages in the radial groove 10 of disc 9, and rotates the disc. Through the groove 11 and pin 15*a*, the disc 9 rotates cam 15 which controls tappet 16 of valve 17. If the axis of shaft 5 coincides with the axis of disc 9, whose position can be varied by displacement of arm 13, there is no difference in the angular velocity of the two units, and groove 11 therefore causes pin 15*a* of cam 15 to rotate at the same angular velocity as shaft 5.

Let us now suppose that disc 9 in FIG. 3 is moved to the right by arm 13, thus producing an eccentricity P, between shaft 5 and disc 9. If shaft 5 rotates at a constant speed, the angular velocity of disc 9 will no longer be equal but, in the angular position shown in FIG. 3, will be higher than that of shaft 5. In fact: $W_9=W_5\ r/r-p$ when:

r = distance of the axis of pin 8*a* from the rotation axis of collar 8.

p = extent of eccentricity (in FIG. 3) of disc 9 with respect to shaft 5.

W_5 = angular velocity of shaft 5.

W_9 = angular velocity of disc 9.

The equation shows that by increasing the eccentricity p, the difference in angular velocity between disc 9 and shaft 5 can be increased (referring to the relative positions of the components as shown in FIG. 3) in terms of an eccentricity p determined by a shift to the right in the position of arm 13 carrying disc 9. In other words, disc 9 is at the end of an acceleration phase which has increased its angular velocity W_9 to a value higher than the angular velocity W_5 of the shaft 5, this value being adjustable at will within predetermined limits by varying the magnitude of the eccentricity p.

When the mechanism is rotated through 180°, the opposite situation occurs, i.e., the rotational speed of disc 9 is lower than that of shaft 5, since the following formula applies:

$$W_9=W_5\ r/r+p$$

From the above, it is apparent that there will be a moment in between the two situations described in which the angular velocity of the two members is equal. This moment will occur whenever the axes of radial grooves 10 and 11 are approximately perpendicular to the plane of the drawing.

It is evident that if shaft 5 and sleeve 8, with pin 8*a*, rotate at the same speed, disc 9 will accelerate or decelerate depending on the relative angular and instantaneous angular positions of the various interconnected components. In two relative angular positions disc 9 will rotate at a speed equal to that of shaft 5, while its rotation speed will be higher or lower than that of shaft 5 in intermediate angular positions.

These variations in relative speed are imparted by the transmission of motion by disc 9 through groove 11 and pin 15*a* to cam 15, with the result that cam 15 has maximum and minimum instantaneous velocities of $W5\ (r+p)/(r-p)$ and $W5\ (r-p)/(r+p)$ respectively.

In essence, with this device, the uniform motion of camshaft 5 can be used to make each cam rotate at different speeds, within the limits of the rotation speeds of shaft 5.

The degree of acceleration and deceleration can be adjusted continuously by varying the value of eccentricity p.

If now, the arm 13 is tilted about an axis parallel to the axis of the camshaft, or in some other way the disc 9 is moved in a direction perpendicular to the camshaft axis and direction y, then the phase angle or angular direction of the eccentricity will be changed, i.e., the plane containing the axes of the camshaft and disc will be rotated about the camshaft axis.

Basically therefore, the device is characterized by the fact that the cams can be moved at varying speeds, using the motion of the camshaft rotating at a constant speed. This speed variation can be regulated arbitrarily both in amplitude and in phase, and can also be inverted within predetermined limits by adjusting the extent and angular direction of eccentricity P.

The description of the device illustrated in rectangle B also applies to the two devices shown in rectangles A and C, except that the former, which controls cam 18, is completely separate, while the latter, which controls cam 19, has collar 8 in common with the mechanism described above, and is moved by collar 8 through pin 8*b*.

From the above, it is clear that the mechanism can alter the lift and fall times of the valves by directly determining the speed at which the cams rotate, and can modify the opening and closing phases of these valves as well as the law governing their motion. The intention is to shift continuously and gradually from the timing diagrams U and U1 in FIG. 1 (corresponding to the power and torque curves U2 and U3 in FIG. 2) to the curves V and V1 (corresponding to diagrams V2 and

V3 in FIG. 2). The basic aims are to obtain power and torque diagrams which are substantially similar to the diagrams indicated in FIG. 2, i.e., curves W and W1, to improve the overall performance of engines and to restrict pollution by exhaust gases.

The magnitude and direction of eccentricity p between shaft 5 and collar 9 can be regulated manually, but can also be adjusted automatically according to the engine speed or load.

FIG. 4 illustrates a modification of the device, corresponding to the illustrations in rectangles B and C in FIG. 3. In this modification the disc 9 is supported by a bearing 20 having a bearing surface 24 which supports disc 9. The bearing 20 also has a bearing surface 20*a* which supports collar 8, so that camshaft 5 is indirectly supported by bearing 20. This bearing 20 can also rotate in the bearing surface 23 of a block 21', the bearing 20 being displaceable angular about shaft 5 by an appropriate lever system (not illustrated). The bearing surface 24 of bearing 20 which supports disc 9 is eccentric with respect to the bearing surfaces 23 and 20*a*. When bearing 20 is rotated, the cam phase angle θ is rotated by the same amount (the magnitude of the eccentricity remaining constant). For example, when bearing 20 is rotated through 180° as shown in FIG. 5, the direction of eccentricity is inverted, so that the acceleration and deceleration effect on the cam is also inverted.

From the embodiments of FIG. 3 and FIG. 4, it has been seen that the eccentricity between the disc 9 and the shaft 5 can be varied in extent p or in cam phasing angle θ or both. This phase angle can be given a reference position in that $\theta = 0$ when the direction of the cam peak is aligned with the direction of the eccentricity displacement at the instant when the driving pins are also aligned with this displacement.

The range of properties obtainable by varying θ is:

θ	Properties (all maxima)
0°	Cam narrowing
90° approx.	Phase advance
180°	Cam broadening
270° approx.	Phase retard

By operating within the appropriate quadrant, the designer can obtain the optimum combination of the properties associated with the angular extremes of the quadrant. Thus for θ between 0° and 90° approx., various combinations of cam narrowing and phase advance can be obtained. Similarly from $\theta =$ 90° approx. to 180° there is a progressive shift from maximum phase advance with no change in cam width to zero phase change with maximum cam broadening.

A physical limit to the application of the variable camshaft may be set by the cam follower accelerations, which will be proportional the square of the instantaneous speed of rotation of the cam.

In applying the variable cam mechanism in an internal combustion engine, the variation of the selected parameter i.e., magnitude or phase) of the eccentricity can be effected in dependence on the engine load or engine speed or a combination of both load and speed. In the case of engine load, a sensor can sense the engine torque, or the gas pressure in the inlet manifold, and an electronic mechanism sensitive to this torque or gas pressure will displace the disc 9 by displacing the bearing 13 or 20 through a hydraulic or pneumatic actuator. In a similar way the electronic mechanism can be controlled by a member sensitive to engine speed.

I claim:

1. A device for use in controlling a valve of an internal combustion engine, said device comprising

a shaft,

a cam mounted for rotation on the shaft and

drive means for moving the cam relative to the driving shaft, said drive means in combination,

a drive member rotatable with the shaft,

an intermediate member mounted for rotation eccentrically with respect to the shaft,

a first drive coupling connecting the drive member to the intermediate member at a position on the drive member spaced a predetermined distance from the axis of the shaft, and

a second drive coupling connecting the cam to the intermediate member at a position thereon spaced about the shaft from the first coupling, and

means responsive to the pressure in the inlet manifold to vary a parameter of the eccentricity of the intermediate member.

2. A device, for use in controlling a valve of an internal combustion engine, said device comprising

a shaft,

a cam mounted for rotation on the shaft and

drive means for moving the cam relative to the driving shaft, said drive means comprising in combination,

a drive member rotatable with the shaft,

an intermediate member mounted for rotation eccentrically with respect to the shaft,

a first drive coupling connecting the drive member to the intermediate member at a position on the drive member spaced a predetermined distance from the axis of the shaft,

a second drive coupling connecting the cam to the intermediate member at a position thereon spaced about the shaft from the first coupling, and

means responsive to the speed of the engine to vary a parameter of the eccentricity of the intermediate member.

3. A device for use in an internal combustion engine to control the opening and closing of the inlet and exhaust valves thereof, the device comprising

a camshaft,

a plurality of cams mounted on said shaft for rotation relative to the shaft,

a drive connection between the shaft and each cam, each drive connection including drive means operative over one part of the cam revolution to drive the cam faster and over another part of the same revolution to drive the cam slower, than the rotational speed of the camshaft, the said drive means comprising

a drive member rotatable with the shaft,

an intermediate member rotatable eccentrically with respect to said shaft,

control means movable to move the intermediate member in a sense to vary the eccentricity of its rotation,

a first connection, at a point spaced from the shaft axis, from the drive member to the intermediate member,

a second connection, at a point spaced from the shaft axis but angularly spaced about the shaft axis from said first connection, from the intermediate member to said cam,

the two connections being on axially opposite sides of the intermediate member and having freedom to permit relative movement between said drive member, said cam and said intermediate member sufficient to accommodate the eccentricity of the intermediate member.

4. A device according to claim 3 wherein said second connection is at the same radial distance from the camshaft as but diametrically opposite to said first connection.

5. A device according to claim 3 including means responsive to variations in a parameter of the engine performance, such for example as engine speed or manifold pressure, to move said control means.

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