(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 963 508 B1

(12) **EUROPEAN PATENT SPECIFICATION**

(45) Date of publication and mention of the grant of the patent:
**03.11.2004 Bulletin 2004/45**

(21) Application number: **98902854.3**

(22) Date of filing: **13.02.1998**

(51) Int Cl.7: **F01L 1/12**, F01L 1/20, F01L 1/30, F01L 1/10, F01L 13/00

(86) International application number:
**PCT/AU1998/000090**

(87) International publication number:
**WO 1998/036157 (20.08.1998 Gazette 1998/33)**

(54) **ADJUSTMENT MECHANISM FOR VALVES**
STELLVORRICHTUNG FÜR VENTILE
MECANISME DE REGLAGE DE SOUPAPES

(84) Designated Contracting States:
**AT BE CH DE DK ES FI FR GB GR IE IT LI LU MC NL PT SE**

(30) Priority: **13.02.1997 AU PO508497**

(43) Date of publication of application:
**15.12.1999 Bulletin 1999/50**

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Printed by Jouve, 75001 PARIS (FR)

## Description

FIELD

**[0001]** The present invention relates to improvements in engines, such as internal combustion engines, particularly to the actuation of valves and most particularly, poppet valves for internal combustion engines.

**[0002]** The present invention also has application to engines or pumps which uses valves.

BACKGROUND

**[0003]** The available torque from an internal combustion engine is largely dependant on the volumetric efficiency of the engine. For reciprocating piston engines, this efficiency is a measure of the volume of atmospheric air drawn into the cylinder/s during an induction stroke, relative to the swept volume of the cylinder/s.

**[0004]** The valve timing of the reciprocating internal combustion engine has a significant effect on the volumetric efficiency of the engine at particular engines speeds. An engine having fixed valve timing, ie a fixed crankshaft angle for valve opening before piston at top dead centre and a fixed crankshaft angle for valve closing beyond top dead centre, will have a particular engine speed where it operates most efficiently. At this speed, the fixed synchronisation of the inlet and exhaust valves opening and closing relative to the piston position create the combination giving the most torque.

**[0005]** Obviously it is desirable to have the most torque possible available over a wide range of engine speeds. To achieve the maximum torque at high engine speeds, it is desirable to have the valves (inlet and exhaust) open for as much piston travel as possible. This gives air more time to enter and exhaust gases more time to exit the cylinder and therefore increases volumetric efficiency. However, there are limiting factors for how much piston travel, or how much of an angle (of crankshaft rotation) the inlet and exhaust valves can be kept open. For example, increasing the angle that the valves are open increases the angle that both the inlet and exhaust valves are open at the same time, which is called valve overlap. Valve overlap is desirable at high engine speeds as it increases torque output. However, the same amount of valve overlap that produces good torque at high engine speeds will cause the engine to run poorly and reduces torque output at low engine speeds. Accordingly, in general, opening the valves earlier and closing them later improves volumetric efficiency at high engine speed at the expense of torque at low engine speed. Conversely decreasing valve overlap increases engine torque at low engine speeds but does not give the best efficiency at high engine speeds.

**[0006]** It is therefore desirable to have a mechanism whereby the timing of the valve opening and closing can be adjusted according to parameters such as engine speed, in order to optimise the torque across a range of engine speeds. Further, other parameters, such as for example, throttle position and which gear is engaged, may be used to vary the timing of the opening and closing of the valves.

**[0007]** Apart from valve timing, there are other factors which are important in the operation of reciprocating engine poppet valves. Firstly, just before the valve is opened, the valve actuator should accelerate slowly towards the valve, in order to reduce then eliminate the clearance between the valve and the actuator or between any intervening tappet arrangement and the actuator. This is to ensure that the valve and actuator do not impact on each other with large velocities or forces. The valve then needs to be opened as quickly as possible in order to facilitate the filling of the cylinder with fresh air and fuel in the case of an intake valve, or empty the cylinder of exhaust gas in the case of an exhaust valve. Once opened, the valve should be held open for as long as possible before closing rapidly. The valve should then reseat as gently as possible and then stay closed until the cycle repeats. As there should be no radical changes in motion, (excessive acceleration) of the valve, a substantially sinusoidal motion has been found to be acceptable in providing a path for valve movement.

**[0008]** The actuation of valves and the control of their motion has been accomplished in the past by the use of camshafts. Camshafts have an eccentric cam lobe that actuates a valve, wherein the profile of the cam lobe determines the motion characteristic of the valve. A problem with this arrangement is that the camshafts spin rapidly and the valves rely on valve springs to keep them in contact with the outer surface of the cam lobe. As the camshafts spin more rapidly, the valves can leave the surface of the cam lobe due to inertia. This problem has been addressed in part by increasing the strength of the valve spring, however this makes opening the valve harder and increases wear on the cam lobe surface.

**[0009]** Another major problem with camshafts is the inability to change the lobe shape, making modification of the motion characteristics of the valve difficult. To modify the valve timing, the camshaft needs to be replaced or machined, with the result that torque is only optimised over a narrow speed range for a particular cam lobe profile. This is one of the reasons that engines that perform well at high speed usually lack torque in the lower range of engine speeds. Further, as the valve springs push against the cam lobes as the camshaft rotates, significant twisting forces are generated along the camshaft, which can result in camshaft breakage.

**[0010]** There are existing devices that attempt to solve some of the above problems, however, none are completely satisfactory. One device is a cam shaft having two standard cam lobes for the two inlet valves, and a third cam lobe between the two standard inlet lobes. When the engine is spinning below a certain engine speed, the inlet valves are actuated by the standard cam lobes. When the engine accelerates over a predetermined engine speed, a pin engages with the valve's actuators, which allows both the valves to be actuated by the third cam lobe, which has a different profile suited to high engine speeds. wherein the inlet valves open earlier and stay open longer. A similar mechanism operates in the exhaust valve camshaft. This system has the disadvantage that it is not possible to vary the valve opening and closing times between the two predetermined valve motion characteristics, ie there are only two valve opening durations available. This results in a marked "step" in torque output from lower rpm to higher rpm and fails to achieve the maximum torque output across the whole range of engine speeds, as effectively only two specific engine speeds are optimised.

**[0011]** Another method of varying the valve opening and closing angle is where the camshaft speed is not always half the crankshaft speed over parts of a single revolution, but varies according to the engine speed. For example, at low rpm the camshaft may spin at the standard rate of half the crankshaft speed. At higher engine speeds, a mechanism mounted on the camshaft causes the camshaft to spin at lower than half crank speed while opening the valves and keeping them open, thus ensuring that the valves are open over a wider angle than at a lower speed. In order to make up lost time (as the crankshaft must average one revolution for every two crankshaft revolutions), the camshaft must then spin faster than half crank speed for the remainder of the revolution to ensure that it is in the correct position when it is time for the valves to open again. This system is obviously less than ideal as a complex mechanism is used to vary the speed of the camshaft with respect to the crankshaft over a single revolution. Further, valve lift cannot be modified as the cam lobe profiles cannot be modified.

**[0012]** Another disadvantage of most valve actuation means is that they comprise a cam shaft which opens the valves. Camshafts are difficult to manufacture, and are subject to wear and breakage.

**[0013]** It is has also been found that the method of adjusting valve clearances between the top of the valve and the valve actuator, for example rocker arm or cam shaft lobe, has disadvantages, such as the need for the clearance adjusting mechanism to be on the rocker arm, thereby adding inertia to the rocker arm, or the use of shims which are difficult to get at under the cam lobes, and require buckets to locate them, which add to the overall length of the valve assembly and therefore add to the dimensions of the engine.

**[0014]** EP 0311282 discloses a valve mechanism for varying the duration and magnitude of opening a valve, including a crank on a drive shaft which reciprocates a link and in turn rotates an intermediate lever about a pivot. The rotation of the intermediate lever actuates the valve via the engagement between a drive pin in the intermediate lever and the curved track of a pivotally mounted valve lever. The actuation of the valve is modified by changing the location of the drive shaft via rotation of a support disk, which in turn changes the reciprocating direction of the link and changes the portion of the track that is engaged by the drive pin.

SUMMARY OF INVENTION

**[0015]** It is an object of the present invention to alleviate at lest one disadvantage associated with the prior art.

**[0016]** According to the present invention, there is provided an apparatus for adjusting the motion characteristics of a valve for an internal combustion engine having a head, said apparatus being as set out in accompanying claim 1.

**[0017]** Accordingly, the present invention provides means for adjusting the motion characteristics of a valve. The motion characteristics of the valve include timing, such as the crankshaft's angular location before the top dead centre reference angle where the valve opens, duration, such as the angle of the crankshaft rotation for which the valve will stay open, lift or travel the amount of lift of the valve for a given crankshaft angular location, rate of travel and/or force. In one form, the adjustment is actuated mechanically. In another form, the adjustment means is located between a valve actuation means and the valve. Advantageously, adjusting the motion characteristics of the valve by way of the present invention, enables selection of engine performance criteria from a range of predetermined characteristics, together with a selection of the degree to which the criteria is to be performed. For example, the adjustment of valve motion characteristics may be selected in a manner which accentuates engine torque. Or, selection may be made to accentuate engine fuel economy.

**[0018]** It may also be desirable to produce a valve actuation means which produces an approximate sinusoidal motion of valve lift in relation to crankshaft rotation and/or which also allows the motion characteristics of the valve to be varied.

**[0019]** Usually the valve actuation means includes a rotating member.

**[0020]** Typically the adjustment means varies the valve opening angle, and/or the valve closing angle and/or the valve lift, either individually or collectively. It has been found that it is advantageous to vary the valve lift and duration, and that while these may be done separately, it has been found that it is beneficial to increase valve lift and valve spring duration as engine speed rises.

**[0021]** Accordingly, it is desirable that the adjustment means varies the valve opening and closing angle and the valve lift collectively.

**[0022]** In another form, the invention provides an apparatus for adjusting the motion characteristics of a valve, including adjustment means to adjust the valve motion in accordance with the adjustment means travel along a non-straight path.

**[0023]** In another form, the invention provides an adjustment means for use in an apparatus for adjusting a motion characteristics of a valve comprising a plate having a guide path.

**[0024]** In another form, the invention provides an apparatus for adjusting the motion characteristics of a valve including a first guide path and a second guide path wherein the motion characteristics of the valve are determined by differences in shape and/or alignment between the first and second guide paths.

**[0025]** In another form, the invention provides an apparatus for adjusting the clearance of a valve actuated by a desmodromic valve actuation means including a valve having a threaded end portion.

**[0026]** In a preferred embodiment, the means for adjusting the motion characteristics of the valve include an adjustment member having a guide path and a pivotally mounted valve actuation member having at least one guide surface, wherein a pin moves along both the guide path and the guide surface, causing the pivotally mounted actuation member to pivot and move the valve.

**[0027]** Typically, the pin is driven in a substantially cyclic motion. Desirably the guide path of the adjustment member and the guide surface of the valve actuation member are not collateral over their entire length, ie there is a difference in the paths such that they deviate from each other at least over part of their length. This difference in paths produces the movement of the actuation member as the pin travels along both paths. Also, a kinematic inversion of pin and guide is contemplated as an alternative embodiment.

PREFERRED EMBODIMENT

**[0028]** One or more of the preferred embodiments of the present invention will now be described, with reference to the accompanying drawings, wherein:

Figures 1a-1d show a schematic representation of an adjustment mechanism in accordance with the present invention in various states of assembly;

Figure 2a shows a schematic side view of the adjustment mechanism of the present invention and a prior art valve actuation mechanism;

Figure 2b shows a schematic view of a non-desmodromic adjustment mechanism of the present invention and a prior art valve actuation mechanism.

Figure 3 shows an isometric view of part of a first embodiment of the adjustment mechanism of the present invention;

Figure 4 shows an isometric view of all of the first embodiment of the adjustment mechanism of the present invention;

Figures 5a and 5b show end views of a second embodiment of the adjustment mechanism of the present invention;

Figures 6a and 6b show end views of the adjustment mechanism shown in figures 5a and 5b;

Figure 7 is a graph of typical extremes of variation in lift and duration of the valves compared with the position of the crankshaft, as varied by the adjustment mechanism of the present invention;

Figure 8 is a first embodiment of a guide plate of the adjustment mechanism of the present invention;

Figure 9 is a second embodiment of the guide plate of the adjustment mechanism of the present invention;

Figures 10a-10d are embodiments of rocker arms of adjustment mechanism of the present invention;

Figure 11 is a schematic side view of a first embodiment of a slot of a guide plate of the adjustment mechanism in accordance with the present invention;

Figure 12 is a schematic side view of a profiled surface of a guide plate of the adjustment mechanism in accordance with the present invention;

Figure 13 is a schematic side view of a second embodiment of the slot of the guide plate of the adjustment mechanism of the present invention;

Figure 14 is a schematic representation of the slot of the guide plate of the adjustment mechanism of the present invention;

Figures 15a-15d are embodiments of a sliding pin of the adjustment mechanism of the present invention;

Figures 16a-16d are embodiments of the guide plates of the adjustment mechanism of the present invention;

Figure 17a is a first embodiment of a guide plate adjustment means of the adjustment mechanism of the present invention;

Figure 17b is a second embodiment of the guide plate adjustment means of the adjustment mechanism of the present invention;

Figure 18a is a perspective view of a valve clearance adjustment mechanism in accordance with the present invention.

Figure 18b is an exploded perspective view of the valve clearance adjustment mechanism shown in figure 18a.

**[0029]** Referring to Figures 2a, 2b, 4, 5a and 5b, a mechanism 10 is shown for adjusting the motion characteristics of a poppet valve 1. The mechanism 10 includes a valve actuation means, for example a valve crankshaft 12 having a crank pin 13, which is used to provide the cyclic displacement motion and base timing for actuation of the valve 1. The valve crankshaft 12 is normally driven by the crankshaft (not shown) by known means such as a belt or chain drive, or gears, at half the crankshaft rotation speed. The mechanism 10 is typically mounted in the head of a reciprocating four stroke engine (not shown) and further includes a pivot point 14 which is fixed to the head of the engine to pivotally locate a rocker arm 16 which actuates the motion of the valve 1. An adjustment member such as a guide plate 18 is mounted to the head such that it is able to be moved within a range of positions, for example a first position

20 and a second position 22 as shown in figures 1b, 5a, 5b, 6a, 6b and 9.

**[0030]** Crank pin 13 is attached to a conrod 24 through aperture 15 at one end and has a sliding pin 26 at the other end as shown in figures 1a-1d, 2 and 3. As the valve crankshaft 12 turns, the conrod 24 moves the sliding pin 26 along a path 25 in the guide plate 18. The guide plate 18 does not move in response to movement of the sliding pin 26, and the sliding pin 26 is constrained to move along path 25. The sliding pin 26 also travels along a path 28 on rocker arm 16. There is typically one rocker arm 16 per valve 1, and accordingly there may be two rocker arms 16 if two inlet (or exhaust) valves are used per cylinder. The two rocker arms 16 and guide plates 18 can be served by a single conrod 24 and sliding pin 26 as shown in figure 4, thus actuating two valves (inlet or exhaust) simultaneously as in a four valve per cylinder engine head. Obviously the number of valves that can be actuated by a single conrod and/or sliding pin is not limited to two per inlet / exhaust. This path 28 may be in the form of a slot having an upper and lower profiled surface, in the case of desmodromic valve actuation, as shown in figures 1a, 1b, 2a, 4, 5a, 5b, 6a, 6b, 10a, 10c, 13 and 14 or it may be a single profiled surface in the case of conventional valve actuation with a spring providing the valve closing force as shown in figures 2b 10b and 10d.

**[0031]** In either method of valve actuation described above, the path 25 in the guide plate 18 causes the sliding pin 26 to move in a way constrained by the profile of the path 25, which causes the rocker arm 16 to pivot about pivot point 14. The rocker arm 16, when pivoting, has a forked actuator arm 32 distal from the pivot point 14 which pushes on the valve 1 via two nuts tightened on the valve stem, as shown in figures 5, 18a and 18b and described later, thus causing valve 1 to open and close according to the differences in the profile of path 25 and path 28. It is the difference in path profiles as shown in figures 13 and 14 that causes the rocker arm 16 to pivot, and the profiles of the paths 28 and 25 may be varied according to the motion characteristics desired from the valve, which may vary from engine to engine, or with the purpose of the engine. Thus the individual path profiles of the guide plate 18 and rocker arm 16 are not intended to be limited to the various embodiments shown in this specification.

**[0032]** Further, the valve 1 can be replaced by valve system 100 as shown in figures 2a and 2b, wherein valve system 100 includes a known shim and bucket arrangement that allows the valve clearance to be adjusted, and a valve spring 101 ensures that the valve stays in contact with the actuator 32 when the valve is closing, which may be used in non-desmodromic or conventional valve actuation. The mechanism 10 may simply replace camshaft 102 as a valve actuation means.

**[0033]** The assembled parts of the mechanism 10 can be seen in figures 1 a to 1d, wherein the assembly of the conrod 24 to the crankshaft 12 by the crank pin 13, and the attachment of the guide plate 18 and rocker arm 16 to the sliding pin 26 is shown schematically. A partially assembled adjustment mechanism is shown in figure 3, wherein the assembled valve crankshaft 12, conrod 24 and sliding pin 26 are seen in relationship to the pivot point 14 which is normally in a fixed position, but can be rotated and includes in this embodiment eccentric section 11. The assembled mechanism 10 is shown in fig 4 wherein two guide plates 18 are slidingly attached to the eccentric section 11 of the rocker shaft 14 and two rocker arms 16 are pivotally attached to the pivot point 14, which would enable the mechanism 10 to operate two inlet or exhaust valves. The guide plates, in this case, are constrained to slide linearly by pins (not shown) which fit into guideways 9.

**[0034]** In operation, the valve crankshaft 12 rotates at half engine crankshaft speed. The conrod 24 is connected at one end to a crank pin 13 on the valve crankshaft 12 and at the other end to the sliding pin 26. The pin 26 is located in a guide path 25 of the guide plate 18. As the valve crankshaft 12 rotates, the pin 26 is constrained to move along the path 25 of the guide plate 18, however the guide plate 18 can move from a first position 20 to a second position 22, and any number of positions therebetween. The profile of the guide path 25, as shown in the figures, defines the trajectory of pin 26. The pin 26 also slides along path 28 of the rocker arm 16, and the different profile between the path 28 and path 25 causes the rocker arm 16 to pivot back and forth about pivot point 14. The actuator 32 attached to rocker arm 16 moves with the arm 16 and contacts the end of valve 1, pushing the valve open and pulling the valve closed. Where non-desmodromic valve actuation is desired, a valve spring may close the valve 1.

**[0035]** The position of the guide plate 18 can be varied, in the case of Figures 2a, 2b, 3 and 4 by rotation of the rocker shaft 14, in a second embodiment adjustment 1s due to eccentric adjusting shaft 30 as can be seen in figures 5a, 5b, 6a, 6b 17a and 17b. The shaft 30 has an eccentric off-centre lobe 31 which can be turned within aperture 34, thus causing the guide plate 18 to move from a first position 20 wherein the motion characteristics as shown by line 40a and 40b of figure 7 suit low engine speed, to a second position 22, wherein the motion characteristics of the valve suit high engine speeds shown by line 42a and 42b, also of in figure 7. The movement of the guide plate 18 can be seen in the comparison of open valve positions shown in figures 5a and 5b. In figure 5a, the adjusting shaft 30 and lobe 31 position the guide plate 18 in the first position 20. In figure 5b, the adjusting shaft 30 and lobe 31 position the guide plate 18 in the second position 22 and thus the maximum valve opening, as seen in figure 5b is greater than the maximum valve position seen in figure 5a. The operation of the shaft 30 and lobe 31 in the aperture 34 in the guide plate is shown in figures 17a and 17b and will be described in more detail below.

**[0036]** The motion characteristics of the valves can be seen in figure 7, wherein line 40a represents the valve lift of an inlet valve (vertical axis) versus crankshaft rotation angle (horizontal axis) for the valve 1 actuated by valve crankshaft 12 while the guide plate 18 is in the first position 20. The exhaust valve motion characteristics when the guide plate 18 is in the first position 20 are shown by line 40b. Line 42a represents the valve motion characteristics when the guide of an inlet valve when the guide plate 18 is in the second position 22. The exhaust valve motion characteristics when the guide plate 18 is in the second position 22, can be seen in line 42b. As can be seen from figure 7, there is a significant difference between valve lift, valve opening duration and valve overlap when the guide plate 18 moves from a first position 20 to a second position 22.

**[0037]** The reason for the difference in motion characteristics is that when guide plate 18 is in the first position 20, the profile of path 28 is positioned such that the differences in the profile between path 25 and path 28 are minimised, as can be seen in figures 5a, 5b and 8 and discussed below. This provides a lower valve lift as the pin 26 deflects less, as shown in the position of pin 26a in figure 8.

**[0038]** The first position 20 of guide plate 18 opens the valve the least amount, and over the shortest angle, and is therefore normally used for low engine speeds where excessive valve overlap is undesirable and increased turbulence is desirable. The second position 22 of the guide plate 18 is used to generate larger valve overlap and higher lift in the valves, as seen in the position of the pin 26b in figure 8, and the extension of the valve in figure 5b compared to the extension of the valve in figure 5a. This arrangement is used during high engine speeds where maximum gas flow is required. Figures 6a and 6b show the guide plate 18 in the first position 20 and second position 22 respectively, but the valves in both cases are closed fully, i.e. regardless of the position of guide plate 18, the valves still close effectively as shown by the equal positions of the valves in figures 6a and 6b. The difference in the positions of the plate 18 is clearly seen by the gap between pivot point 14 and guide plate 18 in figure 6b, whereas in figure 6a there is no gap. The mechanism 10 allows the guide plate 18 to be positioned at any point between the first position 20 and the second position 22, thus allowing the amount of valve overlap and/or the angle of valve opening to be adjusted to any point within the predetermined limits of the valve motion characteristics. This also provides the advantage of being able to modify the valve opening angle and/or lift according to any change in the conditions in order to maximise volumetric efficiency.

**[0039]** The differences between the profiles of path 25 in guide plate 18 and the path 28 in rocker arm 16 are designed to impart the desired valve motion characteristics to the valve. For example, when used with a rocker arm 16 having a straight path 28, the path 25 shown in figure 11, is made up of four portions, each with a specific function. Portion A is a portion whereby, when the sliding pin 26 is in this portion, the valve will be closed. As the pin 26 travels along the path 25, it moves to portion B, which is a ramp section designed to allow the pin 26 to begin to move at an angle to the direction of motion in portion A. This allows the actuator 32 to be brought into contact with the valve (or valve shim) relatively slowly, as there is usually a small gap between the top of the valve assembly and the valve actuator. Once the valve has contacted the top of the valve, sliding pin 26 enters portion C of path 25, where the slope of the path increases greatly and thus causes the actuator to push open the valve quickly. Once the maximum valve opening is approached the sliding pin 26 enters portion D whereby the velocity of the valve while opening is reduced, and the valve starts to decelerate. In portion D, the sliding pin 26 reaches the end of its travel and the valve crankshaft 12 begins to pull the sliding pin 26 back along the portion D in the reverse direction, thus starting to close the valve again.

**[0040]** Figure 12 shows the portions A-D of a profiled surface of a guide plate which uses a spring to return the valve to the closed position, and therefore does not require the lower portion of the path. The shape of the paths 50 and 51 in figures 11 and 12 respectively are designed to be used with a rocker arm having a substantially straight path 28.

**[0041]** If the path of the rocker arm had a shape the same as the shape of the path of the guide plate, then the rocker arm would not move relative to the guide plate and accordingly there would be no motion of the valve. Therefore there are numerous shapes that either the path of the guide plate, or the path of the rocker arm can take in order to produce the required motion of the valve providing that the other of the rocker arm or guide plate has a profile that is different. As an example, the shape of the path 52 of the guide plate shown in figure 13 can be used, provided the shape of the rocker arm path 53 differs in the correct areas to provide the motion in the rocker arm. This difference in path shapes is shown in figure 14 wherein the paths 25 and 28, have been overlapped in order to highlight the differences in the profiles which then cause the rocker arm to deflect and actuate the valve.

**[0042]** From figures 13 and 14, the differences in the paths in the guide plates and rocker arms can be seen, and the differences relate to valve lift. Figure 13 relates to path 52 in a guide plate that is adjusted rotatably, for example as shown in figure 9. It can be seen that this arrangement allows a far greater difference between paths 52 and 53, and accordingly, a far higher valve lift is achieved than in the linearly adjustable guide plate shown in figure 14. This increased valve lift shown in figure 13 is accomplished without a radical increase in path deviation, which would be necessary in a linearly adjustable guide plate, such as that shown in figure 8. It is undesirable to have too large a deviation in any of the paths as this may lead to increased wear on the path surfaces which will cause the valve motion characteristics to change.

**[0043]** An advantage of the present system is that by altering the differences in the profiles of the paths 25 and 28, it is possible to produce a valve motion with, for example, a more square top than that shown in figure 7.

**[0044]** In order to overcome or reduce wear due to high contact pressure between the path 28 and the sliding pin 26, it has been found that the sliding pin can be made with a non-circular cross section, called a wear portion, in the region where it travels along the path 28.

**[0045]** In figure 15a, a wear portion 60 is shown having a flat upper and lower surface where the pin contacts a straight path 28. The profile of the surfaces varies to match the facing surfaces of the paths. For example, the wear surfaces can be flat as in wear surface 60 when used with path 128 in rocker arm 116 as shown in figure 10a. Alternatively the wear surfaces can be curved with a common centre of curvature as shown by wear surfaces 360 in figure 15c to suit a similarly curved path 328 in figure 10c of constant radius. If conventional or non-desmodromic valve actuation mechanisms are used, then the sliding pin only needs one wear surface 260 or 460 as shown in figures 15c and 15d, as the valve spring will ensure continuous contact of the wear surface with the opposing path 28 surface.

**[0046]** It should be noted that the embodiments shown in the figures 10a-10d, 15a-15d, and 16a-16d, work together in respective sets. A rocker arm 116 having a path 128 as shown in figure 10a is used with a gudgeon pin 126 shown in figure 15a and a guide plate 118 having a path 125 shown in figure 16a. This arrangement forms an adjustment mechanism employing desmodromic valve actuation wherein the guide plate 118 is adjusted linearly.

**[0047]** Similarly, a rocker arm 216 having a path 228 (figure 10b) works with a gudgeon pin 226 (figure 15b) and a guide plate 218 having a path 225 (figure 16b) to form an adjustment mechanism employing a valve to close the valve, wherein the guide plate 218 is adjusted linearly.

**[0048]** A rocker arm 316 having a curved path 328 (figure 10c) works with a gudgeon pin 326 (figure 15c) and a guide plate 318 having a path 325 (figure 16c) to form an adjustment mechanism employing desmodromic valve actuation, wherein the guide plate 318 is adjusted pivotally.

**[0049]** A rocker arm 416 having a curved path 428 (figure 10d) works with a gudgeon pin 426 (figure 15d) and a guide plate 418 having a path 425 (figure 16d) to form an adjustment mechanism employing a valve to close the valve, wherein the guide plate 418 is adjusted pivotally.

**[0050]** In figure 17a embodiments of a mechanism for adjusting the position of the guide plate 18 is shown. The adjusting shaft 30 is situated in the aperture 34 in the guide plate 18. By rotating the adjusting shaft 30, the eccentric cam lobe 31 on the adjusting shaft 30 causes the guide plate 18 to move linearly, for example, as shown in figure 8. The amount of linear movement of the guide plate 18 is determined by the amount of rotation of the shaft 30. This allows the guide plate 18 to be adjusted to any point between and including the two extreme positions, being the first position 20 and the second position 22.

**[0051]** In figure 17b, the shaft 30 is rotatably received in to an aperture 134 in a guide plate 618 mounted so as to be pivotally adjustable about point 135. As the shaft is rotated, eccentric lobes 31 force the guide plate 618 to move. The guide plate is constrained to move pivotally and therefore, twisting the shaft 30 causes the guide plate 618 to move. As above, the amount of movement of the guide plate 618 can be controlled by the rotation of the shaft 30.

**[0052]** A control means (not shown) is used to control the rotation of the shaft 30 for each mechanism 10 which enables the guide plate to be positioned anywhere between the first position 20 and the second position 22. The control means may be a simple device for advancing the valve opening by twisting the shaft, or any other suitable means for moving the guide plate. Such mechanisms are commonly used to advance the ignition timing as engine speed rises. The valve timing in this case may be adjusted either with or independent of the ignition timing.

**[0053]** A further embodiment of a guide plate 518 is shown in figure 9 wherein the guide plate 518 is mounted to a rotatable pivot point 535, so that adjustment of the motion characteristics of the valve can be made by rotating pivot point 535 to which the guide plate 518 is attached, to any position between the two positions as shown by the arrow and dotted line, rather than linear motion as shown by the arrow in figure 8.

**[0054]** It should also be understood that the guide plates in any of the embodiments disclosed may be positioned in discrete locations between the first position 20 and the second position 22, for example by the use of a stepper motor. This would allow the position of the guide plates to be varied in steps according to data from various parameters such as engine speed, rate of change of engine speed, throttle position and gear position. Accordingly, a fuzzy logic table could be set up to position the guide plates in the optimum position for a set of predefined parameters.

**[0055]** Figures 16a to 16d show further alternative arrangements for the guide plates. Each guide plate is arranged to be mounted in such a way that its position is able to be controlled in order for the position of the path for the sliding pin to be controlled. In a non-desmodromic arrangement as shown in figure 16b and 16d, there is no requirement for the path to be a slot, and as such profiles 125 and 325 can be used, as a spring acting on the valve can be used in a conventional manner to close the valve and accordingly there will always be pressure on either profile 125 or profile 325 and the underside of the respective rocker arms 118 or 328. This arrangement has the advantage that there is a large body of knowledge regarding the use of valve springs to close a valve. Also, the reciprocating rocker arms may be made lighter.

**[0056]** Embodiments of the means for adjusting the position of the guide plate is shown in figure 17a and 17b. Figure 17a relates to a method of producing linear adjustment in the guide plate using a shaft 30 in an aperture 34 in the guide plate 18. The shaft has a lobe 31 which moves the guide plate to the desired position when the shaft 30 is turned. Aperture 34 is designed to move the guide plate 18 linearly, and therefore has substantially straight side walls. As many engines of the type that use poppet valves have numerous valves in alignment, a single shaft with multiple lobes 31 can be used to move all the guide plates 18 simultaneously.

**[0057]** A further embodiment is shown in figure 17b, wherein the shaft 30 is used to cause a rotational motion in the guide plate 18. The twisting of the shaft 30 with eccentric lobe 31 in aperture 134 causes the guide plate to pivot about fixed point 135. If the guide plate is mounted about a pivot point, as shown in figure 9, then the rotation of the shaft 30 will cause the guide plate to rotate, and thus increase or decrease the difference between the paths in the guide plate and rocker arm, which will effect the motion characteristics of the valve. As the aperture 134 is designed to move the guide plate 18 pivotally, side wall 136 is longer than side wall 137.

**[0058]** In the above embodiments, the rocker arm has pivoted while the guide plate has moved either linearly or pivotally. It can be readily determined that the rocker arm could also move linearly in response to the movement of the pin in the path of the guide plate. Further, the guide plate may be fixed in place, and all the adjustment movements can take place on the rocker arm, eg the rocker arm could have its pivot point moveable with respect to the guide plate. This arrangement has the advantage that the guide plate is then fixed, and all the movement is undertaken by the rocker arm, making the mounting of the guide plate greatly simplified.

**[0059]** It can be seen from the embodiments disclosed that the movement of the guide plate 18 from its first position to the second position causes the sliding pin 26 travelling along path 25 to not only increase the crank rotation angle across which the valves open, but also increases valve lift at the same time. These aspects in combination produce a result that is very desirable, as two of the valve characteristics change with only a change in one parameter, that being the movement of the guide plate. It is desirable to have the valves increase their lift at high engine speeds to ensure that the maximum amount of air enters the cylinder or exhaust gas exits from the cylinder in the time provided. However, at low engine speeds, it has been found that increased turbulence in the air entering the cylinder is desirable as it assists in the atomisation of the fuel in the air. When engines operate at low speeds, the velocity of the air entering the cylinder is also low, and therefore there is not as much turbulence in the air as it passes the inlet valves into the cylinder. It has been found that decreasing the valve lift and duration increases turbulence and therefore increases fuel atomisation, which increases torque. At higher engine speeds, the turbulence from the faster air flow provides sufficient energy for fuel atomisation, and the limiting factor becomes the amount of air able to be squeezed into the cylinder. The present invention allows for the adjustment of not only the valve opening duration, but also valve lift with only one parameter being adjusted.

**[0060]** The motion characteristics of the valve may also be varied in accordance with factors such as throttle position and also which gear is selected.

**[0061]** It should be noted that it is not essential to increase valve lift and duration with engine speed, and that it may be desirable under certain circumstances to decrease valve lift and/or durations of the inlet and/or exhaust valve as engine speed increases which the present invention is also able to accommodate.

**[0062]** In figure 18a there is shown a guide plate 418 used in desmodromic valve actuation, having two branches 420, each branch having an actuator 32. The actuators 32 sit between an upper flange member 422 and a lower flange member 424 at the upper end of a valve 1. The valve 1 includes a threaded portion 426, which has a lower nut 425 including the lower flange member 424 threadedly attached thereto, as shown in figure 18b. An upper nut 428, which is threadedly attached to the threaded portion 426 of the valve 1, includes the upper flange member 422. The gap between the upper flange member 422 and the lower flange member 424 may be set by an intermediate shim member (not shown) which would fit between the upper nut 428 and lower nut 425, whereby the size of the shim determines the gap between the upper flange member 422 and lower flange member 424.

**[0063]** In the embodiment shown, the upper flange member 422 includes a spacer 423 which contacts a corresponding spacer 427 on the lower flange member 424, thus providing the appropriate gap between the flanges. Typically, the size of the gap is slightly larger than the diameter of the portions of the actuator 32 that contact the upper and lower flanges, thereby allowing a clearance between the flanges and the actuators 32. The upper and/or lower nuts may be held in position by lock nuts (not shown). The valve clearance may then be adjusted by removing the upper lock nut (if provided), removing the upper nut 428 having upper flange member 422 and spacer 423, and replacing the upper flange member 422 with spacer with another flange member and spacer of suitable size, then reattaching the lock nut onto the threaded portion of the valve 1. In this way, the valve clearance can be adjusted to take into account any wear in the system, without having to replace the guide plate 418. The spacer 423 may be integral with or separate to the upper flange member 422, and the lower flange member 424 may also be replaced if desired.

**[0064]** Alternatively, the upper nut 428 and upper flange member 422 may be locked into position by the upper nut 428, and the gap between the upper and lower flanges can be set by the position in which the upper nut 428 and flange member 422 are set.

**[0065]** It is important that the contact surfaces of actuator 32 be of constant radius, as in this way the valve clearance will be constant as the valve is opened and closed, and the rocker arm 16 moves about pivot point 14.

## Claims

1. An apparatus for adjusting the motion characteristics of a valve for an internal combustion engine having a head, said apparatus being **characterized by** in combination:

   a guide plate (18) having a first non-straight guide path (25), said guide plate being adapted for movement of its position relative to the head;
   a guide member (26) and a valve actuator (16) in order to actuate the valve, said valve actuator being pivotally mounted and having a second guide path, wherein the first and second guide paths have differences in shape and/or alignment;
   a valve actuation means (12, 24) including a rotating member adapted to impart reciprocating motion to the guide member;

   **characterised in that**:

   the guide member being adapted to move along the first and second guide paths thereby causing the valve actuator to move relative to the guide plate due to the differences in the two paths;
   and the adjustment being determined by the position of the guide plate being varied to adjust the operative motion of the guide member.

2. The apparatus of claim 1, **characterised in that** the adjustment is determined by the position of the guide plate (18) being varied to adjust the motion characteristics of the valve (1) in response to operative motion of the guide member (26).

3. The apparatus of claim 1, **characterised in that** the adjustment is determined by the position of the guide plate (18) being varied to adjust the direction of the operative motion of the guide member (26).

4. An apparatus as claimed in claim 1, 2 or 3 **characterised in that** the valve includes at least one inlet and at least one exhaust valve of a cylinder of an intemal combustion engine, the adjustment being over a range of opening and closing angles for each valve, said apparatus further having:

   an inlet adjustment having an inlet guide plate (18) with a non-straight guide path (25) and adapted for movement relative to the head, an inlet guide member (26) and an inlet valve actuator (16) adapted for movement relative to the gulde plate in order to actuate the at least one inlet valve; and
   an outlet adjustment having an outlet guide plate (18) with a non-straight guide path (25) and adapted for movement relative to the head, an outlet guide member (26) and an outlet valve actuator (16) adapted for movement relative to the guide plate in order to actuate the at least one outlet valve;

   each respective member being in communication with their respective valve actuator and the respective guide plate, each member also being adapted to travel along their respective non-straight guide path of their respective guide plate wherein the adjustment is determined by the position of the respective guide plate being varied to adjust the operative motion of the respective member, and
   the motion characteristics of the inlet valve are variable independently of the motion characteristics of the outlet valve.

5. The apparatus of claim 1, 2, 3 or 4, **characterised in that** the valve actuator includes an arm (16), a first portion (28) of the arm being coupled to the guide plate (18), and a second portion (32) of the arm contacting the valve.

6. The apparatus of any one of claims 1 to 5, **characterised in that** the position of the guide plate (18) is variable during operation of the engine.

7. The apparatus of any one of claims 1 to 6, further **characterised in that** the valve actuator includes a desmodromic means for positively opening and closing the valve.

8. The apparatus of any one of claims 1 to 7, further **characterised in that** the rotating member includes a crankshaft (12) acting on a conrod (24) adapted to impart linear motion, wherein the conrod is connected at one end to the guide member and at the other end to a crank pin (13) on the crakshaft (12).

9. The apparatus of claim 8, **characterised in that** the crankshaft and conrod form a pilman rod assembly.

10. The apparatus of any one of claims 1 to 9, **characterised in that** the guide plate is coupled to the

head of the engine in a manner that enables movement of the guide plate relative to the head.

**11.** The apparatus of any one of claims 1 to 10, wherein the guide plate is coupled to the head to allow each guide plate to slide with respect to the head.

**12.** The apparatus of any one of claim 1 to 11, wherein the adjustment to the motion characteristics of the valve include one or a combination of valve opening angle, valve closing angle, valve rate of travel, valve lift, or valve opening duration.

**13.** The apparatus of claim 12, **characterized in that** the adjustment is over a range of one or a combination of valve opening angle, valve dosing angle, valve rate of travel, valve lift, or valve opening duration.

**14.** The apparatus as claimed in claim 1, wherein the guide plate (18) is movable between a first position and a second position, and any number of positions therebetween.

**15.** The apparatus as claimed in claim 14, wherein the differences In the paths are reduced when the guide plate is in the first position.

**16.** The apparatus as claimed in claim 15, wherein the position of the guide plate is varied by rotation of a rocker shaft (14) extending through an aperture in the guide plate.

**17.** The apparatus as claimed in claim 15, wherein the position of the guide plate is varied by rotation of an eccentric shaft (30) having an off-centre lobe (30) which is tumed within an aperture in the guide plate.

## Patentansprüche

**1.** Vorrichtung zum Einstellen der Bewegungsabläufe eines Ventils für einen Verbrennungsmotor, der einen Kopf aufweist, wobei die Vorrichtung durch die folgende Kombination **gekennzeichnet** ist:

eine Führungsplatte (18), welche eine nicht-gerade Führungsbahn aufweist (25), wobei die Führungsplatte so ausgelegt ist, dass sie ihre Lage relativ zum Kopf bewegt;
ein Führungselement (26) und ein Ventilstellglied (16) zur Betätigung des Ventils, welches Ventilstellglied drehbar befestigt ist und eine zweite Führungsbahn aufweist, wobei die erste und die zweite Führungsbahn in Form und/oder Ausrichtung verschieden sind;
Ventilbetätigungsmittel (12, 24), die ein drehbares Element aufweisen, welches das Führungselement in eine Hin- und Herbewegung versetzt;

**dadurch gekennzeichnet, dass**:

das Führungselement so ausgelegt ist, dass es sich längs der ersten und der zweiten Führungsbahn bewegt, wodurch sich das Ventilstellglied, auf Grund der Unterschiede der beiden Bahnen, relativ zur Führungsplatte bewegt;
und die Regelung durch die Lage der Führungsplatte bestimmt wird, welche Lage verändert wird, um die. Betätigungsbewegung des Führungselements einzustellen.

**2.** Vorrichtung nach Anspruch 1, **dadurch gekennzeichnet, dass** die Regelung durch die Lage der Führungsplatte (18) bestimmt ist, welche Lage zur Regelung der Bewegungscharakteristika des Ventils (1) in Abhängigkeit von der Betätigungsbewegung des Führungselements (26) geändert wird.

**3.** Vorrichtung nach Anspruch 1, **dadurch gekennzeichnet, dass** die Regelung durch die Lage der Führungsplatte (18) bestimmt ist, welche Lage zum Einstellen der Richtung der Betätigungsbewegung des Führungselements (26) verändert wird.

**4.** Vorrichtung nach einen der Ansprüche 1, 2 oder 3, **dadurch gekennzeichnet, dass** das Ventil zumindest ein Einlass- und ein Auslassventil eines Zylinders eines Verbrennungsmotors umfasst, wobei die Regelung für jedes Ventil über einen Öffnungs- und Schließwinkelbereich erfolgt, wobei die genannte Vorrichtung weiters aufweist:

eine Einlassregelung, die eine Einlassführungsplatte (18) mit einer nicht-geraden Führungsbahn (25), wobei die Platte für eine Bewegung relativ zum Kopf ausgebildet ist, sowie ein Einlassführungselement (26) und ein Einlassventilstellglied (16) aufweist, welches für eine Bewegung relativ zur Führungsplatte ausgebildet ist, um das zumindest eine Einlassventil zu betätigen; und
eine Auslassregelung, die eine Auslassführungsplatte (18) mit einer nicht-geraden Führungsbahn (25), wobei die Platte für eine Bewegung relativ zum Kopf ausgebildet ist, sowie ein Auslassführungselement (26) und ein Auslassventilstellglied (16) aufweist, welches für eine Bewegung relativ zur Führungsplatte ausgebildet ist, um das zumindest eine Auslassventil zu betätigen;

jedes zugehörige Element steht mit dem zugehörigen Ventilstellglied und der zugehörigen

Führungsplatte in Verbindung, weiters ist jedes Element so ausgelegt, dass es sich längs der zugehörigen nicht-geraden Führungsbahn der jeweiligen Führungsplatte bewegbar ist, wobei die Regelung durch die Lage der jeweiligen Führungsplatte bestimmt ist, welche Lage zur Regelung der operativen Bewegung des zugehörigen Elements geändert wird, und wobei

die Bewegungscharakteristika des Einlassventils unabhängig von den Bewegungscharakteristika des Auslassventils änderbar ist.

**5.** Vorrichtung nach Anspruch 1, 2, 3 oder 4, **dadurch gekennzeichnet, dass** das Ventilstellglied einen Arm (16) aufweist, wobei ein erster Abschnitt (28) des Arms mit der Führungsplatte (18) verbunden ist und ein zweiter Abschnitt (32) des Arms das Ventil kontaktiert.

**6.** Vorrichtung nach einem der Ansprüche von 1 bis 5, **dadurch gekennzeichnet, dass** die Lage der Führungsplatte (18) während des Betriebs des Verbrennungsmotors änderbar ist.

**7.** Vorrichtung nach einem der Ansprüche von 1 bis 6, weiters **dadurch gekennzeichnet, dass** das Ventilstellglied ein desmodromisches Mittel zum zwangsläufigen Öffnen und Schließen des Ventils aufweist.

**8.** Vorrichtung nach einem der Ansprüche von 1 bis 6, weiters **dadurch gekennzeichnet, dass** das Drehelement eine Kurbelwelle (12) aufweist, welche eine Pleuelstange (24) antreibt, die für die Erzeugung einer linearen Bewegung ausgebildet ist, wobei ein Ende der Pleuelstange mit dem Führungselement und das andere Ende mit einem Zapfen (13) auf der Kurbelwelle (12) verbunden ist.

**9.** Vorrichtung nach Anspruch 8, **dadurch gekennzeichnet, dass** die Kurbelwelle und die Pleuelstange ein "pilman"-Gestänge bilden.

**10.** Vorrichtung nach einem der Ansprüche von 1 bis 9, **dadurch gekennzeichnet, dass** die Führungsplatte mit dem Zylinderkopf so verbunden ist, dass eine Bewegung der Führungsplatte relativ zum Kopf ermöglicht ist.

**11.** Vorrichtung nach einem der Ansprüche von 1 bis 10, **dadurch gekennzeichnet, dass** die Führungsplatte so mit dem Kopf verbunden ist, dass jeder Führungsplatte ein Gleiten relativ zum Kopf ermöglicht ist.

**12.** Vorrichtung nach einem der Ansprüche von 1 bis 11, **dadurch gekennzeichnet, dass** das Einstellen für die Bewegungscharakteristika des Ventils eines von bzw. eine Kombination von Ventilöffnungswinkel, Ventilschließwinkel, Ventilverstellgeschwindigkeit, Ventilhub oder Ventilöffnungsdauer aufweist.

**13.** Vorrichtung nach Anspruch 12, **dadurch gekennzeichnet, dass** das Einstellen sich über einen Bereich eines von bzw. einer Kombination von Ventilöffnungswinkel, Ventilschließwinkel, Ventilverstellgeschwindigkeit, Ventilhub oder Ventilöffnungsdauer erstreckt.

**14.** Vorrichtung nach Anspruch 1, **dadurch gekennzeichnet, dass** die Führungsplatte (18) zwischen einer ersten Stellung und einer zweiten Stellung, mit einer beliebigen Anzahl von dazwischen liegenden Stellungen, verstellbar ist.

**15.** Vorrichtung nach Anspruch 14, **dadurch gekennzeichnet, dass** die Unterschiede zwischen den Bahnen verringert sind, wenn sich die Führungsplatte in der ersten Stellung befindet.

**16.** Vorrichtung nach Anspruch 15, **dadurch gekennzeichnet, dass** die Lage der Führungsplatte durch die Drehung eines, durch eine Bohrung in der Führungsplatte hindurchgeführten Schwingzapfens (14) verstellbar ist.

**17.** Vorrichtung nach Anspruch 15, **dadurch gekennzeichnet, dass** die Stellung der Führungsplatte durch die Drehung einer Exzenterwelle (30) mit einem Nocken (30), welche in einer Bohrung der Führungsplatte sitzt, veränderbar ist.

**Revendications**

**1.** Appareil pour régler les caractéristiques de mouvement d'une soupape pour un moteur à combustion interne qui a une culasse, ledit appareil étant **caractérisé par**, en combinaison :

une plaque guide (18) ayant un premier chemin guide non rectiligne (25), ladite plaque guide étant adaptée pour un mouvement de sa position par rapport à la culasse;

un élément guide (26) et un actionneur de soupape (16) pour actionner la soupape, ledit actionneur de soupape étant monté de façon à pouvoir pivoter et ayant un second chemin guide, où les premier et second chemins guides ont des différences en forme et/ou en alignement;

un moyen d'actionnement de soupape (12, 24) comprenant un élément tournant adapté pour transmettre un mouvement alternatif à l'élément guide;

**caractérisé en ce que** :

l'élément guide étant adapté à bouger le long des premier et second chemins guides, faisant de ce fait que l'actionneur de soupape se déplace par rapport à la plaque guide du fait de la différence entre les deux chemins;
et le réglage étant déterminé par le fait que la position de la plaque guide est modifiée pour régler le mouvement d'actionnement de l'élément guide.

**2.** Appareil selon la revendication 1, **caractérisé en ce que** le réglage est déterminé par le fait que la position de la plaque guide (18) est modifiée pour régler les caractéristiques de mouvement de la soupape (1) en réponse au mouvement d'actionnement de l'élément guide (26).

**3.** Appareil selon la revendication 1, **caractérisé en ce que** le réglage est déterminé par le fait que la position de la plaque guide (18) est modifiée pour régler la direction du mouvement d'actionnement de l'élément guide (26).

**4.** Appareil selon la revendication 1, 2 ou 3, **caractérisé en ce que** la soupape comprend au moins une soupape d'admission et au moins une soupape d'échappement d'un cylindre d'un moteur à combustion interne, le réglage étant dans une plage d'angles d'ouverture et de fermeture pour chaque soupape, ledit appareil ayant en outre :

un réglage d'admission ayant une plaque guide d'admission (18) avec un chemin guide (25) non rectiligne, adapté pour un mouvement par rapport à la culasse, un élément guide d'admission (26) et un actionneur de soupape d'admission (16) adapté pour un mouvement par rapport à la plaque guide, de façon à actionner ladite au moins une soupape d'admission; et
un réglage d'échappement ayant une plaque guide d'échappement (18) avec un chemin guide (25) non rectiligne, adapté pour un mouvement par rapport à la culasse, un élément _guide d'échappement (26) et un actionneur de soupape d'échappement (16) adapté pour un mouvement par rapport à la plaque guide, de façon à actionner ladite au moins une soupape d'échappement;

chaque élément respectif étant en communication avec son actionneur de soupape respectif et la plaque guide respective, chaque élément étant également adapté à se déplacer le long de son chemin guide non rectiligne respectif de sa plaque guide respective, où le réglage est déterminé par le fait que la position de la plaque guide respective est modifiée pour régler le mouvement d'actionnement de l'élément respectif, et

les caractéristiques de mouvement de la soupape d'admission sont modifiables indépendamment des caractéristiques de mouvement de la soupape d'échappement.

**5.** Appareil selon la revendication 1, 2, 3 ou 4, **caractérisé en ce que** l'actionneur de soupape comprend un bras (16), une première partie (28) du bras étant raccordée à la plaque guide (18), et une seconde partie (32) du bras étant en contact avec la soupape.

**6.** Appareil selon une quelconque des revendications 1 à 5, **caractérisé en ce que** la position de la plaque guide (18) est modifiable pendant le fonctionnement du moteur.

**7.** Appareil selon une quelconque des revendications 1 à 6, **caractérisé en outre en ce que** l'actionneur de soupape comprend un moyen desmodromique pour positivement ouvrir et fermer la soupape.

**8.** Appareil selon une quelconque des revendications 1 à 7, **caractérisé en outre en ce que** l'élément tournant comprend un vilebrequin (12) qui agit sur une bielle (24) adaptée pour transmettre un mouvement linéaire, dans lequel la bielle est raccordée à une extrémité à l'élément guide et à l'autre extrémité à un maneton (13) sur le vilebrequin (12).

**9.** Appareil selon la revendication 8, **caractérisé en ce que** le vilebrequin et la bielle forment un ensemble bielle - manivelle.

**10.** Appareil selon une quelconque des revendications 1 à 9, **caractérisé en ce que** la plaque guide est raccordée à la culasse du moteur d'une façon qui permet un mouvement de la plaque guide par rapport à la culasse.

**11.** Appareil selon une quelconque des revendications 1 à 10, dans lequel la plaque guide est raccordée à la culasse pour permettre à chaque plaque guide de glisser par rapport à la culasse.

**12.** Appareil selon une quelconque des revendications 1 à 11, dans lequel le réglage des caractéristiques de mouvement de la soupape comprend un ou une combinaison de : angle d'ouverture de soupape, angle de fermeture de soupape, vitesse de déplacement de soupape, levée de soupape ou durée d'ouverture de soupape.

**13.** Appareil selon la revendication 12, **caractérisé en ce que** le réglage couvre une plage d'un ou d'une combinaison de : angle d'ouverture de soupape, angle de fermeture de soupape, vitesse de déplacement de soupape, levée de soupape ou durée d'ouverture.

**14.** Appareil selon la revendication 1, dans lequel la plaque guide (18) est déplaçable entre une première position et une seconde position, et n'importe quel nombre de positions entre celles-ci.

**15.** Appareil selon la revendication 14, dans lequel les différences entre les chemins sont réduites quand la plaque guide est dans la première position.

**16.** Appareil selon la revendication 15, dans lequel la position de la plaque guide est .modifiée par rotation d'un axe de culbuteur (14) qui s'étend au travers d'une ouverture dans la plaque guide.

**17.** Appareil selon la revendication 15, dans lequel la position de la plaque guide est modifiée par rotation d'un axe excentrique (30) qui a un lobe excentré (30) qui est tourné dans une ouverture dans la plaque guide.

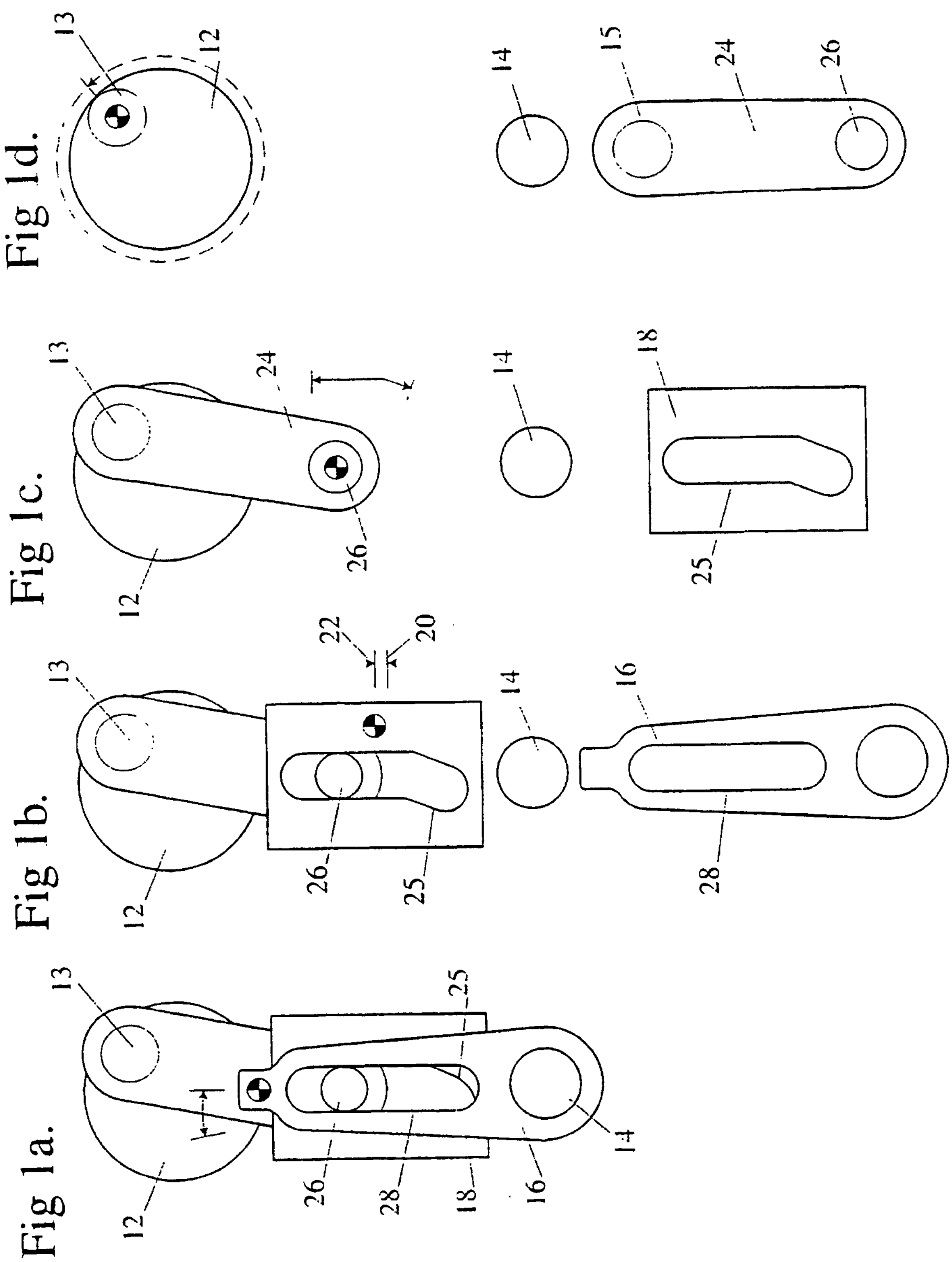
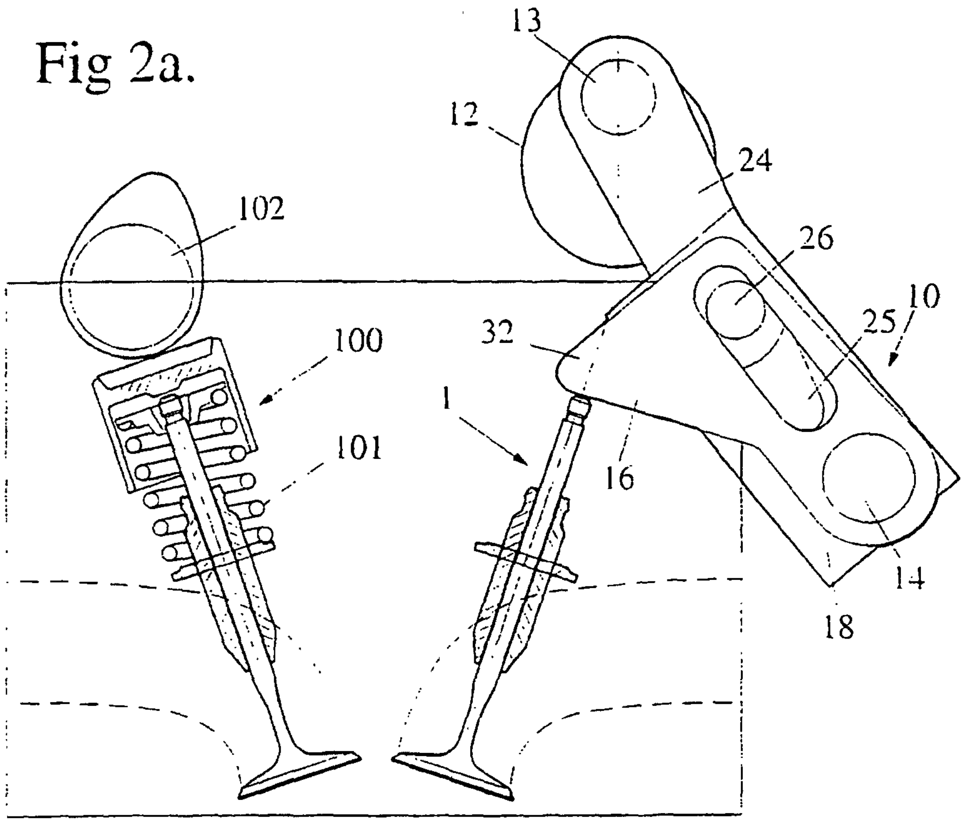


Fig 1a.
Fig 1b.
Fig 1c.
Fig 1d.
12
13
14
15
16
18
20
22
24
25
26
28


13
12
24
102
26
10
25
32
100
1
101
16
14
18
Fig 2a.

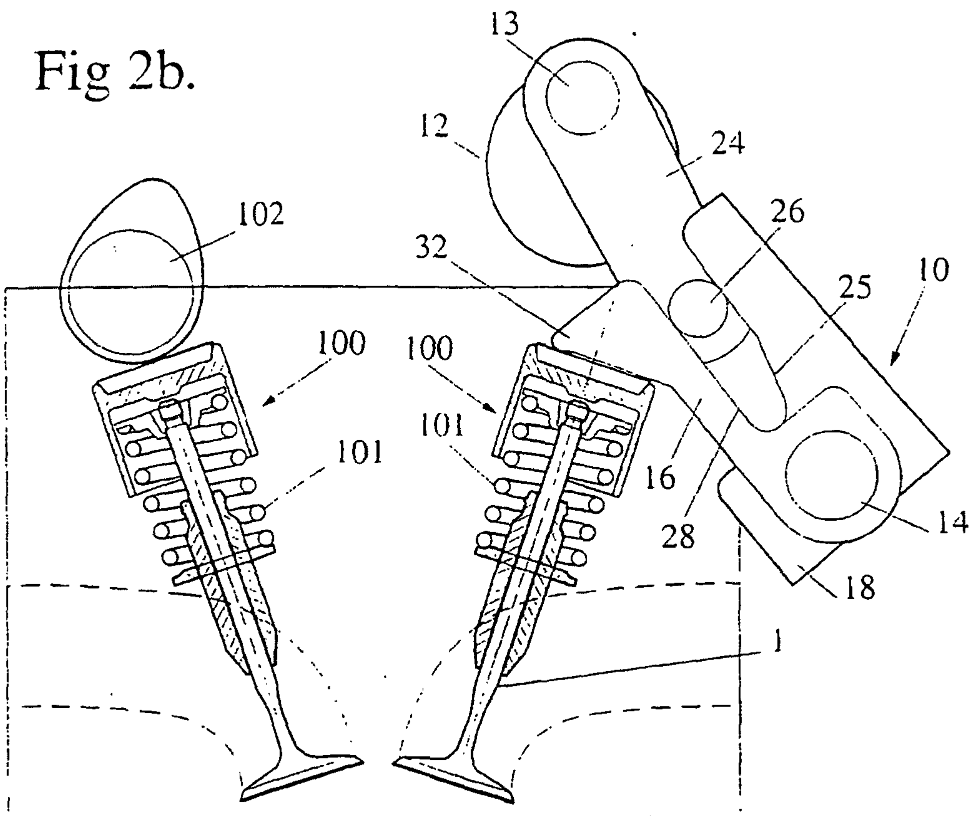

13
12
24
102
26
32
10
25
100
100
101
101
16
28
14
18
1
Fig 2b.

Fig 3.

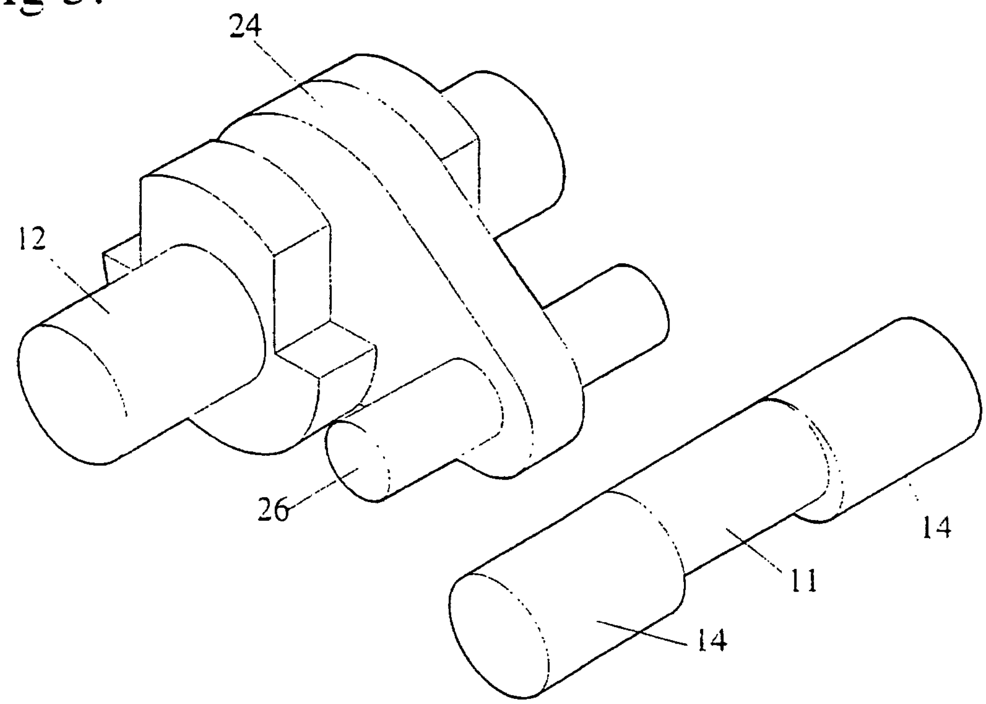



Fig 4.

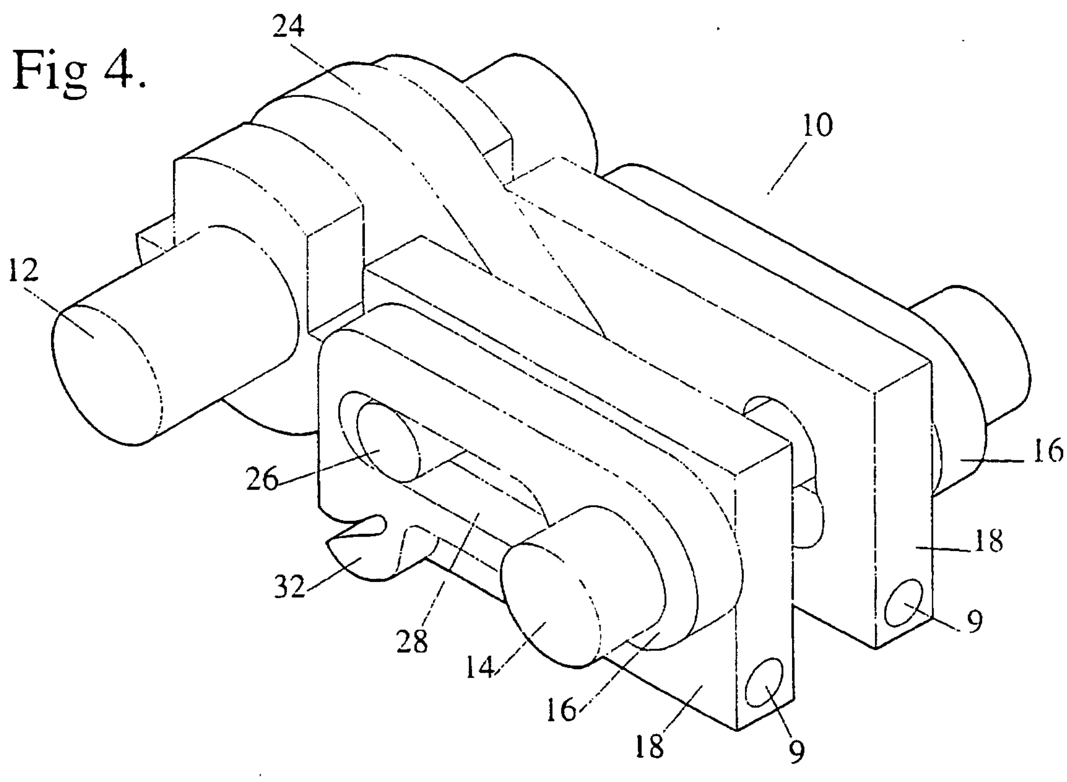
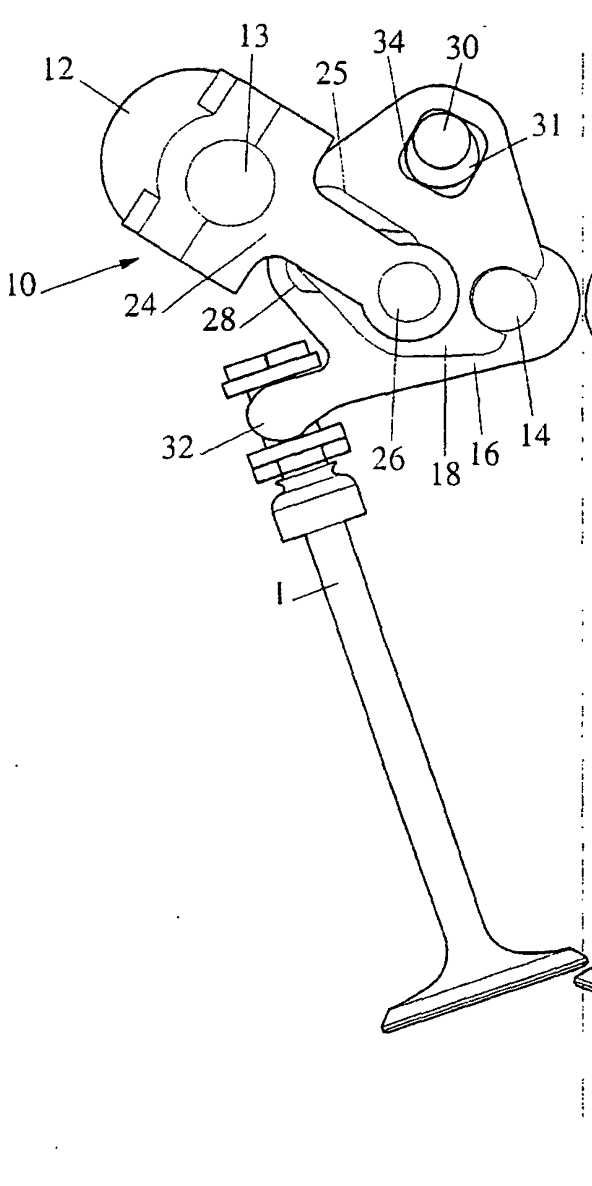
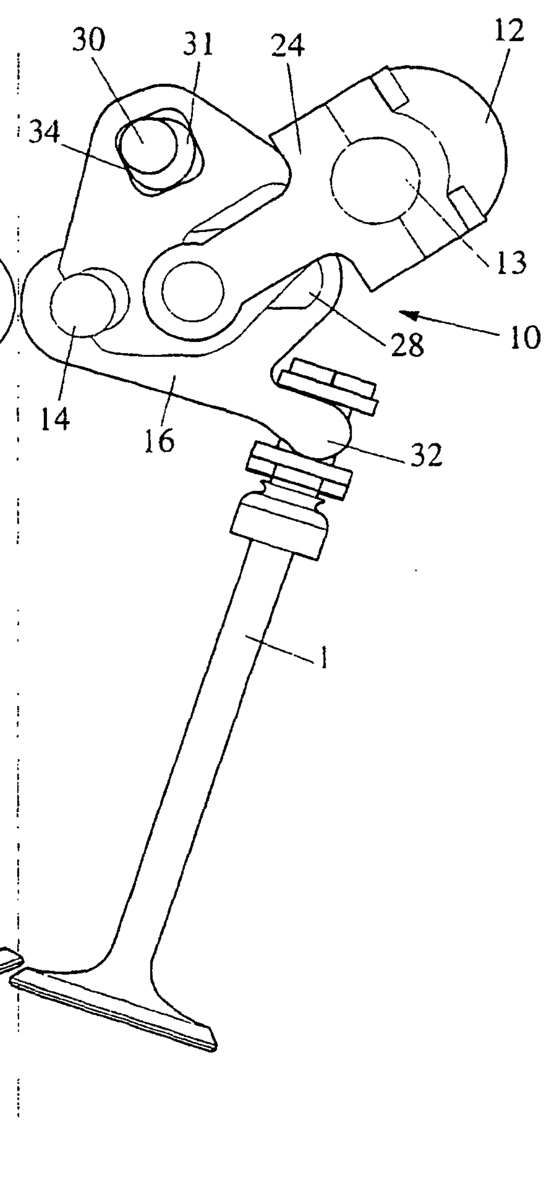


Fig5a.



12
13
34
30
25
31
10
24
28
32
26
18
16
14
1


Fig5b.



30
31
24
12
34
13
10
28
14
16
32
1

Fig6a.

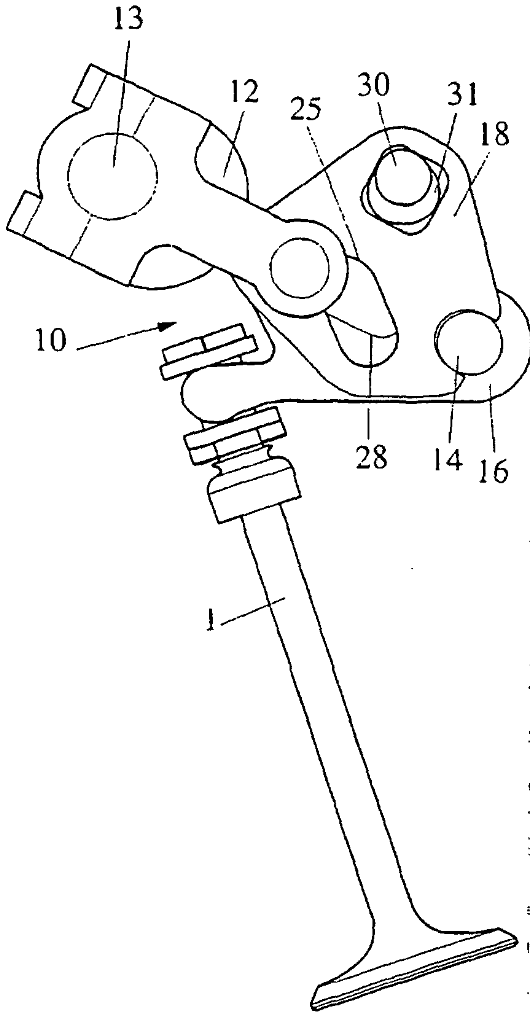
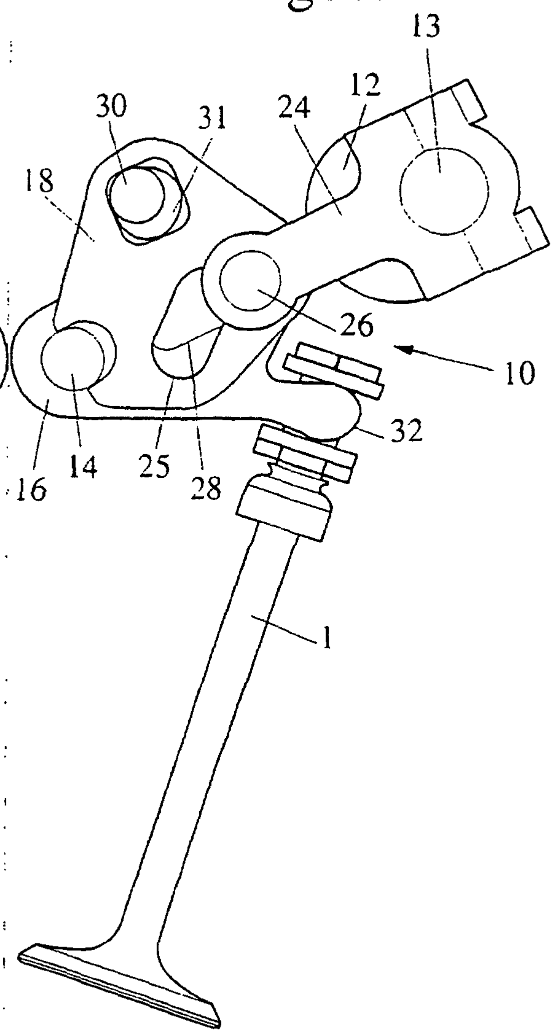



Fig6b.



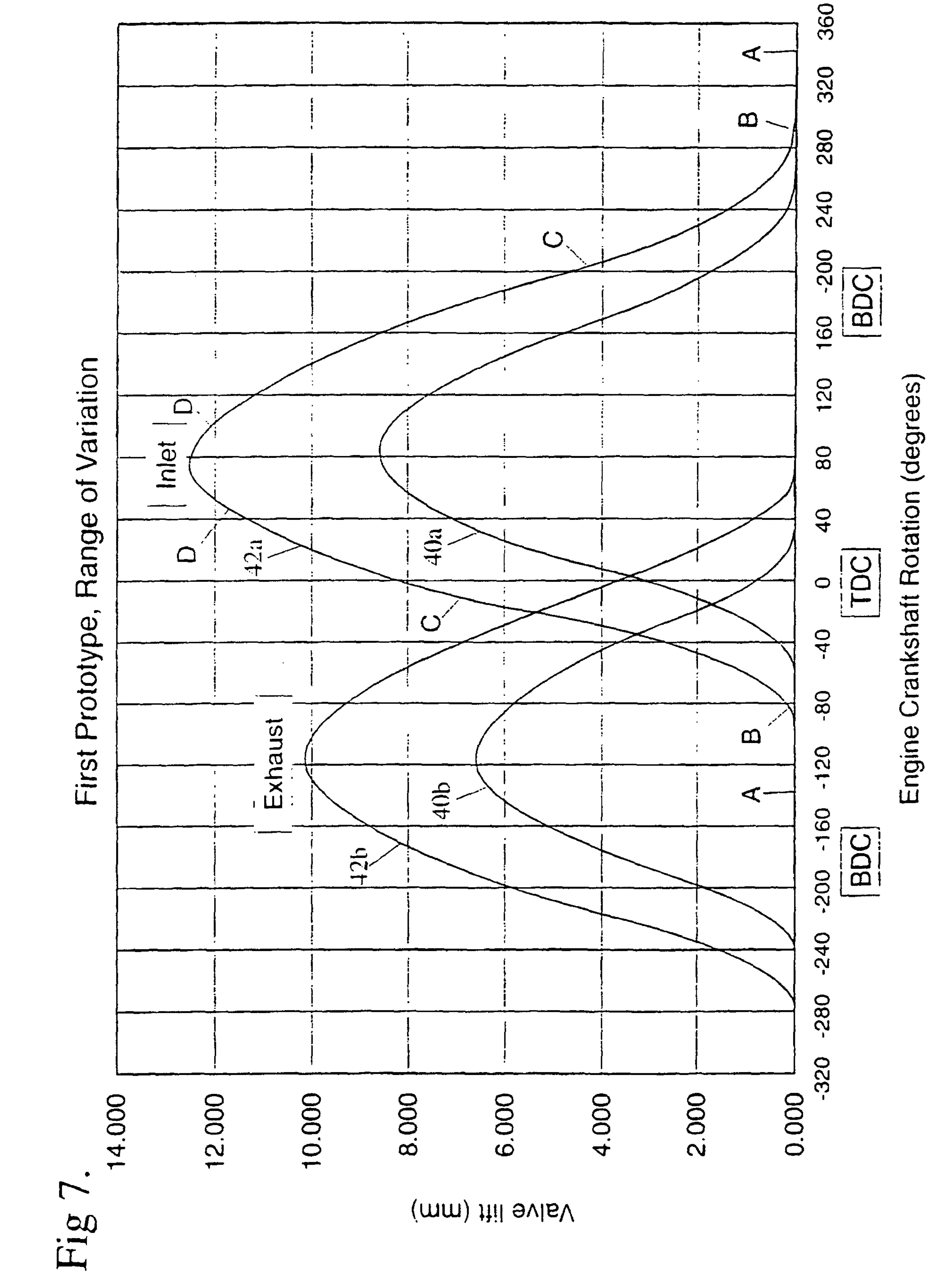

First Prototype, Range of Variation
Valve lift (mm)
14.000
12.000
10.000
8.000
6.000
4.000
2.000
0.000
-320
-280
-240
-200
-160
-120
-80
-40
0
40
80
120
160
200
240
280
320
360
BDC
TDC
BDC
Engine Crankshaft Rotation (degrees)
Exhaust
Inlet
42b
40b
42a
40a
A
B
C
D


Fig 7.

Fig 8.

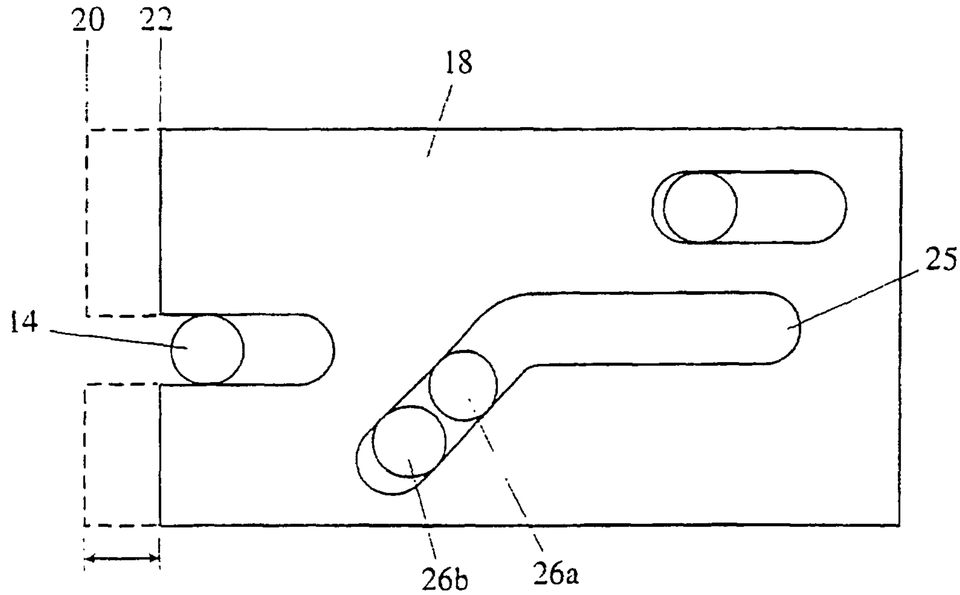



Fig 9.

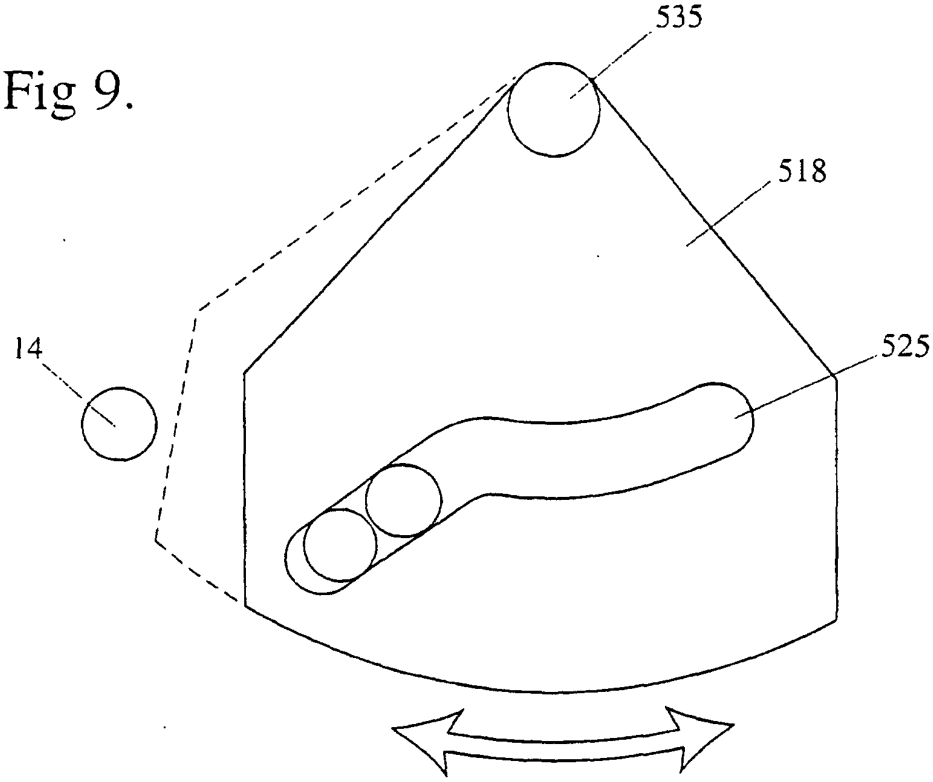


Fig 10a.

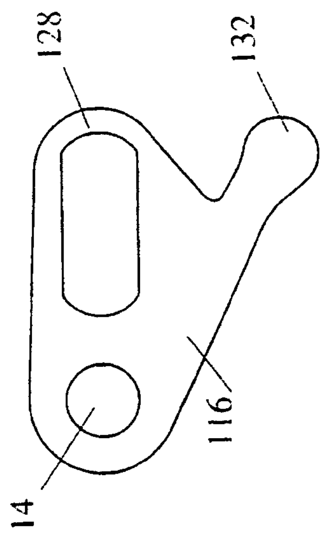



Fig 10b.

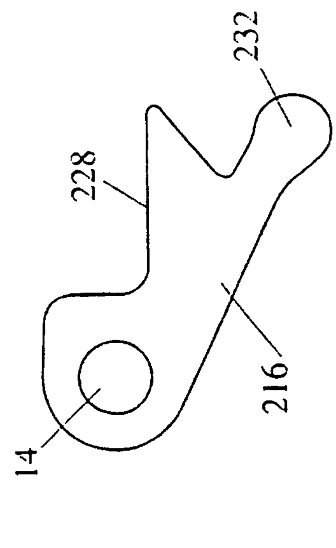



Fig 10c.

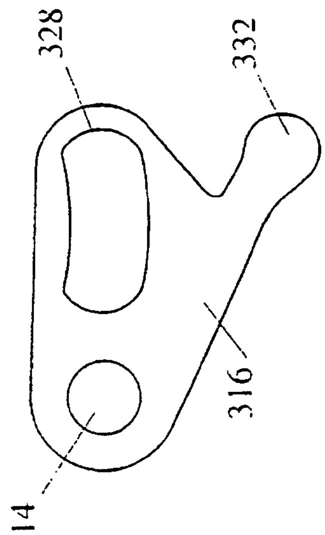



Fig 10d.

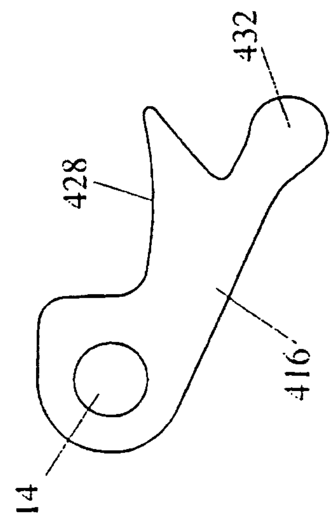


Fig 11.

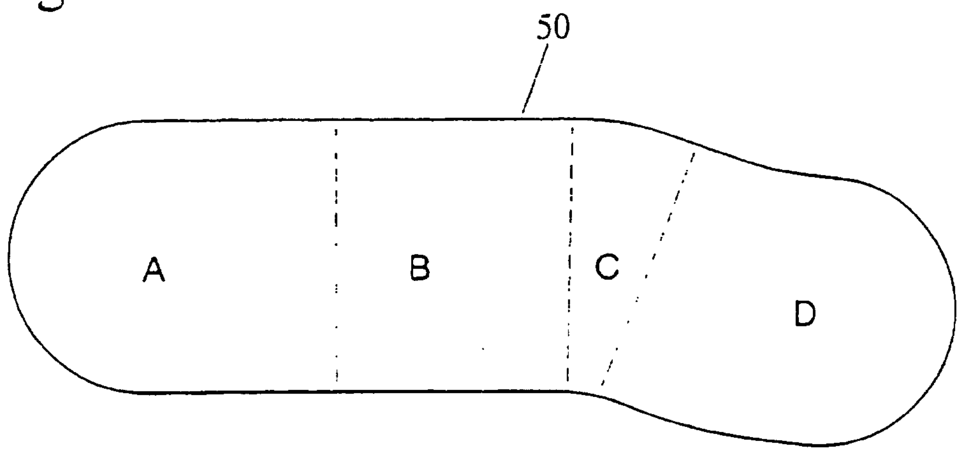



Fig 12.

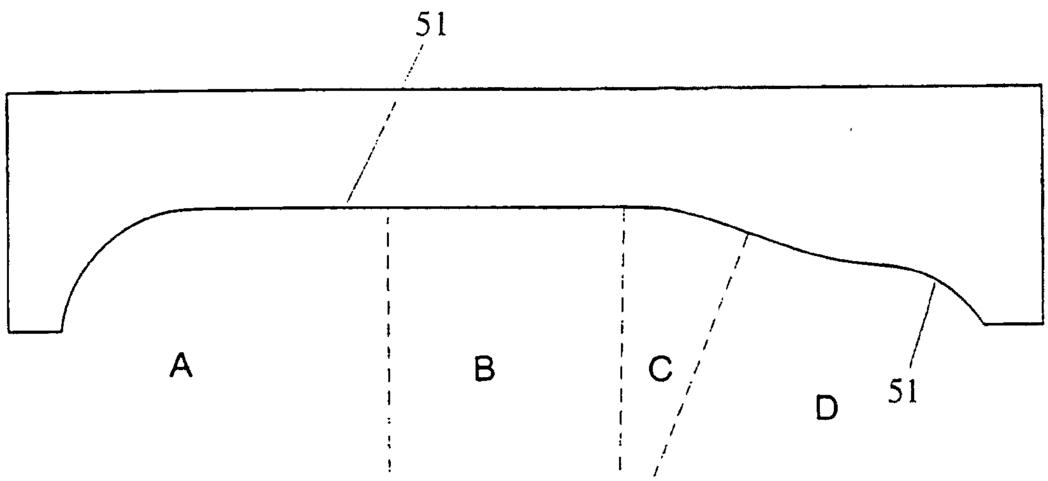


Fig 13.

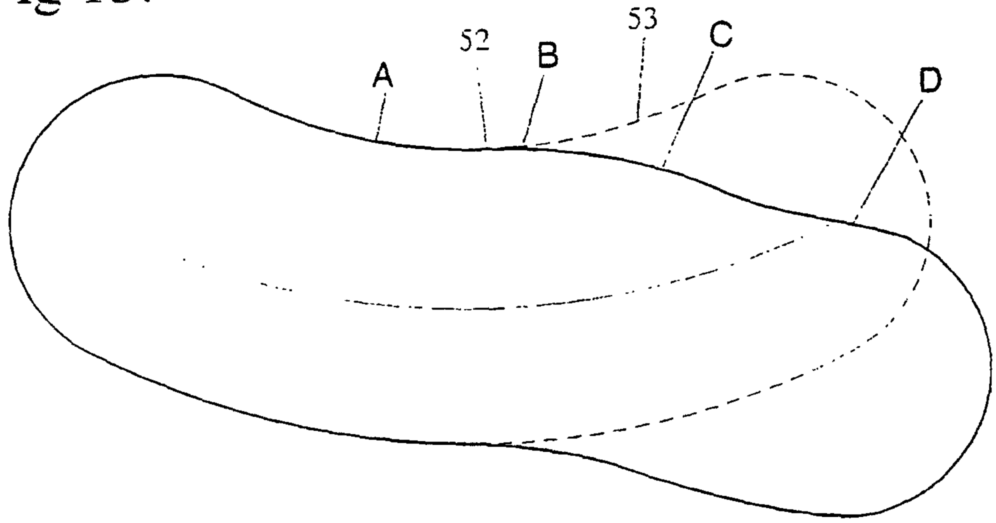
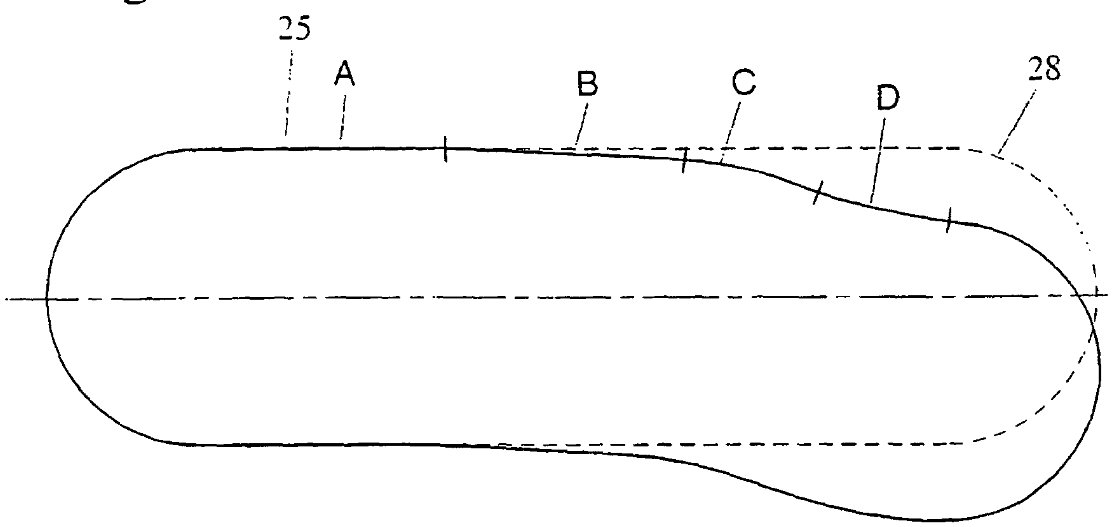



Fig 14.



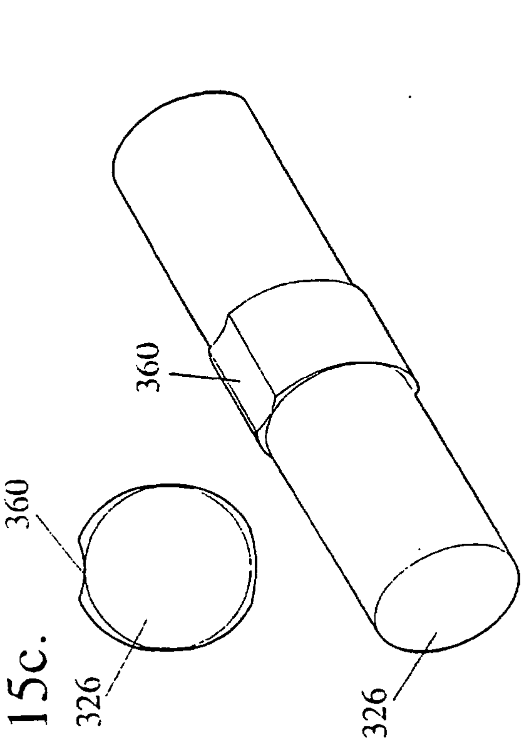


Fig 15c.

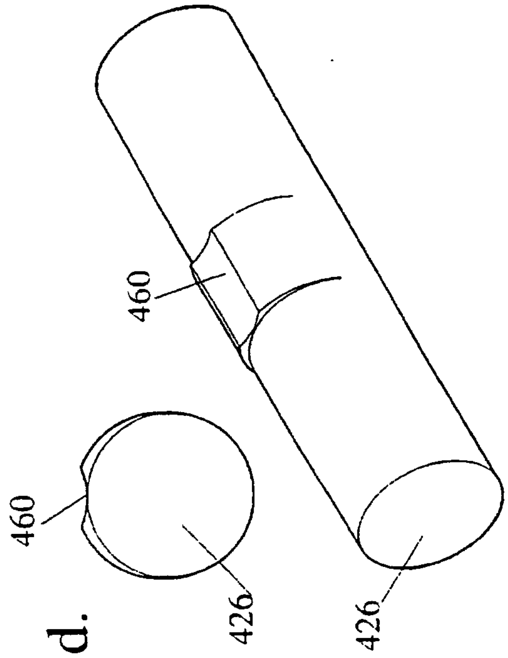


Fig 15d.

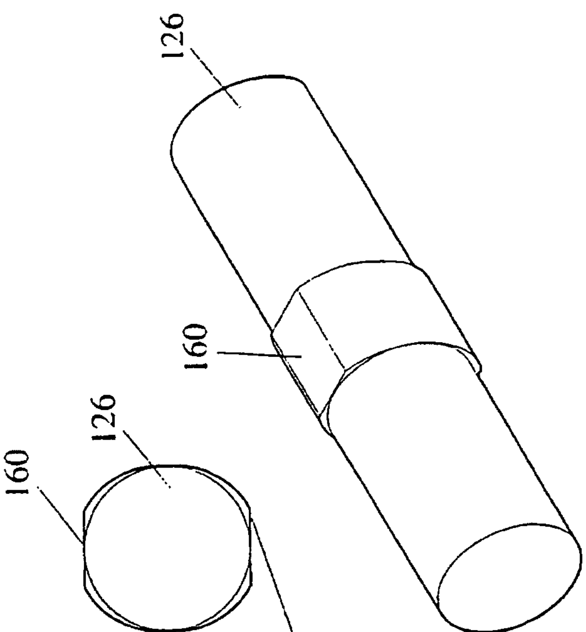


Fig 15a.

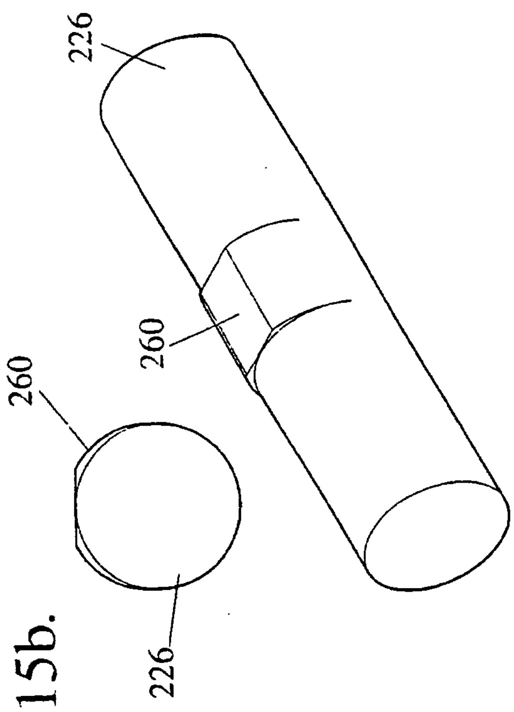


Fig 15b.

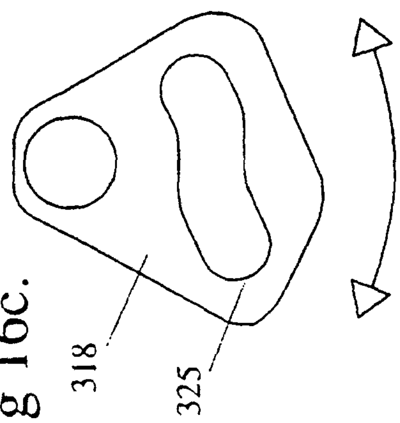



Fig 16c.

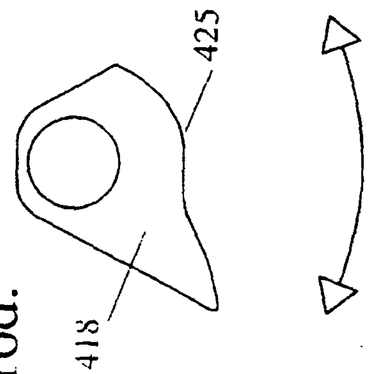



Fig 16d.

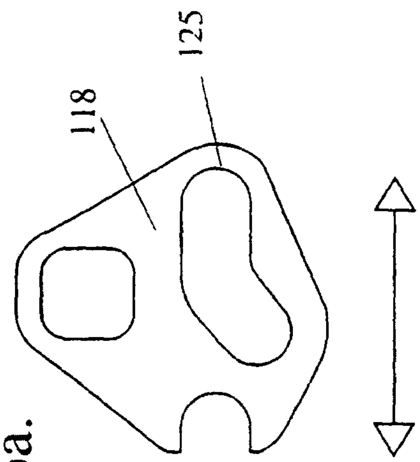



Fig 16a.

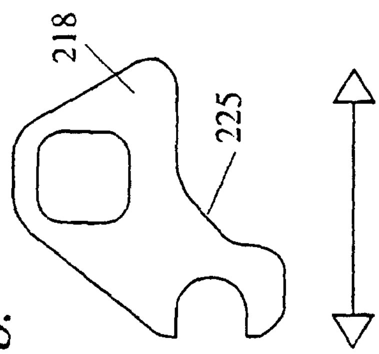



Fig 16b.

Fig 17a.

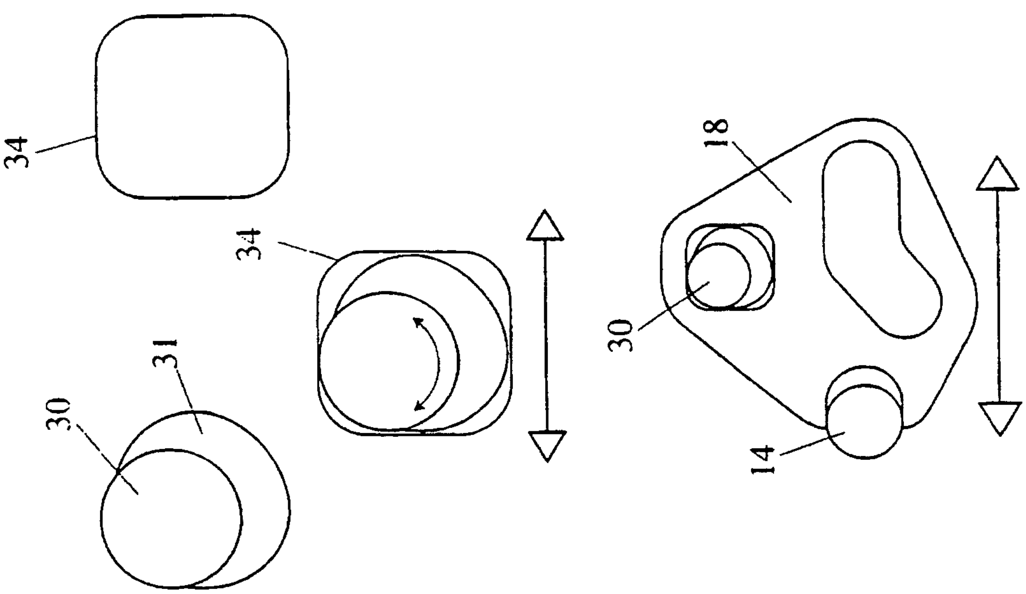



Fig 17b.

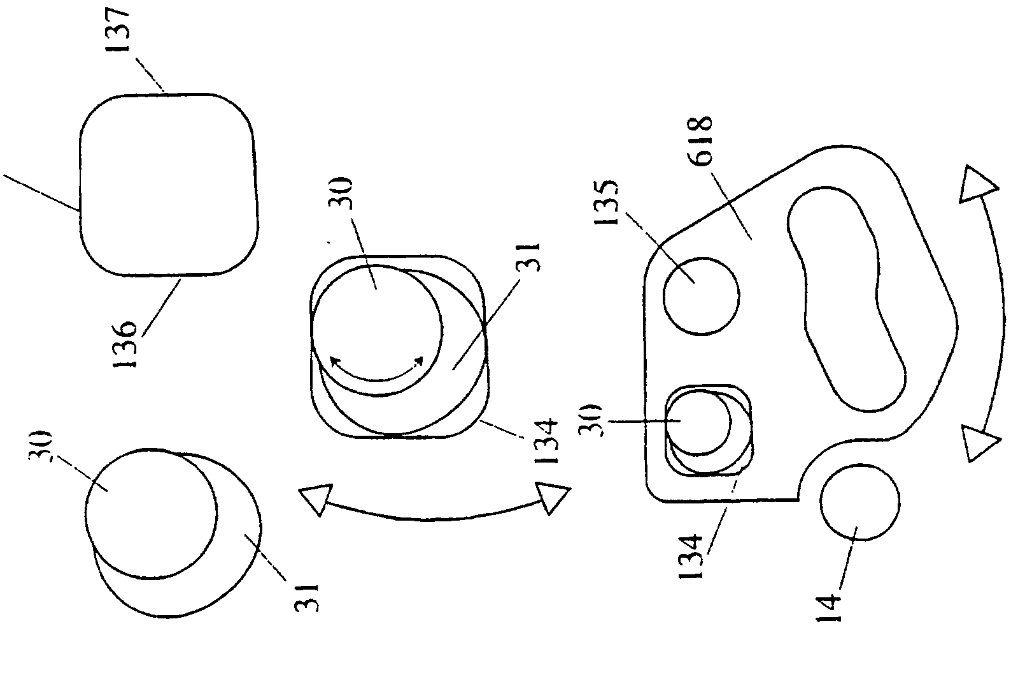


Fig 18a.

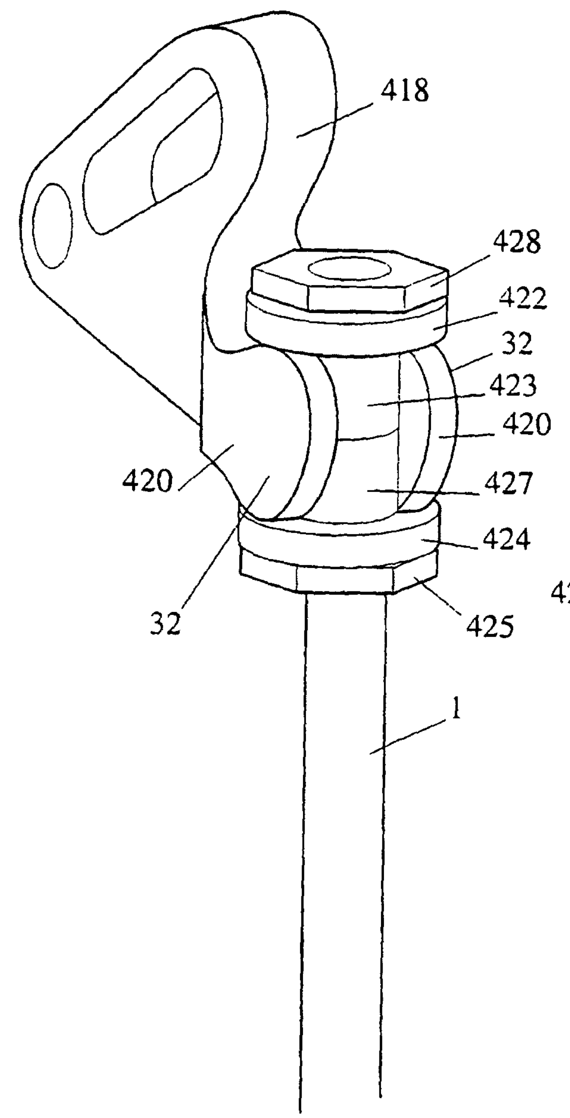



Fig 18b.