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(54) **Engine braking devices and methods**

(57) Apparatus and method for converting an internal combustion engine from a normal engine operation (20) to an engine braking operation (10). The engine includes exhaust valve train components comprising at least one exhaust valve (300) and at least one cam (230). The apparatus comprises actuation means (100) having at least one component integrated into at least one of the exhaust valve train components, such as a rocker arm (210) or a valve bridge (400). The actuation means (100) has an inoperative position and an operative position. In the inoperative position, the actuation means (100) is retracted and the small braking cam lobes (232 & 233) are skipped to generate a main valve lift profile (220m) for the normal engine operation (20). In the operative position, the actuation means (100) is extended so that the motion from all the cam lobes (220, 232 & 233) is transmitted to the at least one exhaust valve (300) for the engine braking operation (10).

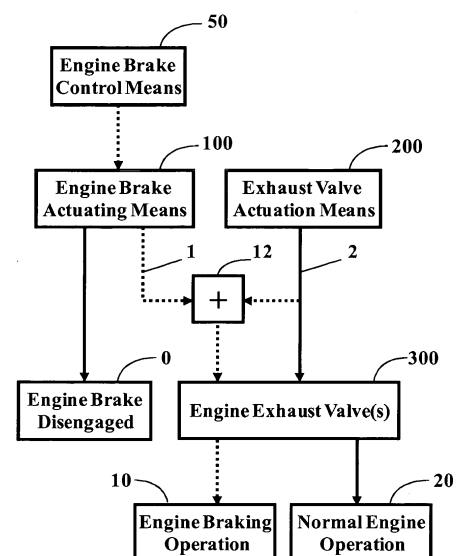


FIG. 1

Description

BACKGROUND OF THE INVENTION - FIELD OF INVENTION

[0001] The present invention relates generally to the braking of an internal combustion engine, specifically to engine braking devices and methods.

BACKGROUND OF THE INVENTION - PRIOR ART

[0002] It is well known in the art to employ an internal combustion engine as brake means by converting the engine temporarily into a compressor. It is also well known that such conversion may be carried out by cutting off the fuel and opening the exhaust valve(s) at or near the end of the compression stroke of the engine piston. By allowing compressed gas (typically, air) to be released, energy absorbed by the engine to compress the gas during the compression stroke is not returned to the engine piston during the subsequent expansion or "power" stroke, but dissipated through the exhaust and radiator systems of the engine. The net result is an effective braking of the engine.

[0003] An engine brake is desirable for an internal combustion engine, particularly for a compression ignition type engine, also known as a diesel engine. Such engine offers substantially no braking when it is rotated through the drive shaft by the inertia and mass of a forward moving vehicle. As vehicle technology has advanced, its hauling capacity has increased, while at the same time rolling and wind resistances have decreased. Accordingly, there is a heightened braking need for a diesel-powered vehicle. While the normal drum or disc type wheel brakes of the vehicle are capable of absorbing a large amount of energy over a short period of time, their repeated use, for example, when operating in hilly terrain, could cause brake overheating and failure. The use of an engine brake will substantially reduce the use of the wheel brakes, minimize their wear, and obviate the danger of accidents resulting from brake failure.

[0004] There is also a desire to use an engine brake when shifting gears in the gearbox of the vehicle. This is apt to be an even more important aspect in commercial vehicles such as trucks and buses that are ever more frequently equipped with automatic or semi-automatic gearboxes. Such gearboxes can be likened to conventional manual gearboxes, with the difference being that the shifting of gears is carried out by means of a control device, instead of manually by the driver. In order to reduce loss of driving power of the engine during up-shift, it is an advantage if the engine speed can be matched to the new gear ratio as soon as possible. It is known to selectively introduce an engine brake during an up-shift when certain operating parameters are obtained, in order to achieve a rapid decrease of engine speed during the gear shifting process.

[0005] There are different types of engine brakes. Typ-

ically, an engine braking operation is achieved by adding an auxiliary engine valve event called an engine braking event to the normal engine valve event. Depending on how the engine valve event is produced, an engine brake can be defined as:

- (a) Type I engine brake - the engine braking event is produced by importing motions from a neighboring cam, which generates the so called Jake brake;
- (b) Type II engine brake - the engine braking event is produced by altering existing cam profile, which generates a lost motion type engine brake;
- (c) Type III engine brake - the engine braking event is produced by using a dedicated cam for engine braking, which generates a dedicated cam (rocker) brake;
- (d) Type IV engine brake - the engine braking event is produced by modifying the existing engine valve lift, which normally generates a bleeder type engine brake;
- (e) Type V engine brake - the engine braking event is produced by using a dedicated valve train for engine braking, which generates a dedicated valve (the fifth valve) engine brake. The engine brake can also be divided into two big categories, i.e., the compression release engine brake (CREB) and the bleeder type engine brake (BTEB).

[0006] Compression release engine brakes open the exhaust valve(s) at or near the end of the compression stroke of the engine piston (also known as top dead center or TDC). They typically include hydraulic circuits for transmitting a mechanical input to the exhaust valve(s) to be opened. Such hydraulic circuits typically include a master piston that is reciprocated in a master piston bore by a mechanical input from the engine. Hydraulic fluid in the circuit transmits the master piston motion to a slave piston in the circuit, which in turn, reciprocates in a slave piston bore in response to the flow of hydraulic fluid in the circuit. The slave piston acts either directly or indirectly on the exhaust valve(s) to be opened during the engine braking.

[0007] An example of a prior art CREB is provided by the disclosure of Cummins, U.S. Patent No. 3,220,392, which is hereby incorporated by reference. Engine braking systems based on the patent have enjoyed great commercial success. However, the prior art engine braking system is a bolt-on accessory that fits above the overhead. In order to provide space for mounting the braking system, a spacer may be positioned between the cylinder head and the valve cover that is bolted to the spacer. This arrangement may add unnecessary height, weight, and costs to the engine. Many of the above-noted problems result from viewing the braking system as an accessory to the engine rather than as part of the engine itself.

[0008] There is a need for design systems that reduce the weight, size and cost of engine braking systems, and

improve the inter-relation of various ancillary equipments, such as the turbocharger and the exhaust brake with the retarding system. In addition, the market for compression release engine brakes has moved from the after-market, to original equipment manufacturers. Engine manufacturers have shown an increased willingness to make design modifications to their engines that would increase the performance and reliability and broaden the operating parameters of the compression release-type engine brake.

(a) Earlier Integrated Rocker Brake

[0009] One possible solution is to integrate components of the braking system with the rest of the engine components. One attempt at integrating parts of the compression braking system is found in U.S. Patent No. 3,367,312 to Jonson, which discloses an engine braking system including a rocker arm having a plunger, or piston, positioned in a cylinder integrally formed in one end of the rocker arm wherein the plunger can be locked in an outer position by hydraulic pressure to permit braking system operation. Jonson also discloses a spring for biasing the plunger outward from the cylinder into continuous contact with the exhaust valve to permit the cam-actuated rocker lever to operate the exhaust valve in both the power and braking modes. A control valve is used to control the flow of pressurized fluid to the rocker arm cylinder so as to permit selective switching between braking operation and normal power operation.

[0010] However, the control valve unit of Jonson's compression braking system is positioned separately from the rocker arm assembly, resulting in unnecessarily long fluid delivery passages and a longer response time. This also leads to an unnecessarily large amount of oil that must be compressed before activation of the braking system can occur, resulting in large compliance and less control over the timing of the compression braking. Moreover, the control valve is a manually operated rotary type valve requiring actuation by the driver often resulting in unreliable and inefficient braking operation. Also, rotary valves are subject to undesirable fluid leakage between the rotary valve member and its associated cylindrical bore.

(b) Integrated Rocker Brake with Two-Valve Opening for Engine Braking

[0011] Another integrated engine braking system for commercial vehicles is known from U.S. Patent No. 5,564,385 ("the '385 patent") in which a stroke-limited hydraulic piston is arranged at the operating end of a rocker arm for taking up valve play in the valve mechanism of the engine. A pressure regulating valve is utilized for supplying pressurized oil to the hydraulic piston for taking up valve play in the rocker arm. The oil is supplied to the rocker arm by means of a canal, which is provided with an exhaust in the shape of a very narrow hole through

which oil can flow, and in this way be made to affect the valve body to, depending on operation, be positioned in any of the predetermined positions. For this purpose, the control valve is also provided with an adjustable magnet valve arranged for drainage of oil that has been fed through the narrow hole.

[0012] Although the engine brake system disclosed in the '385 patent has enjoyed considerable commercial success, it has some drawbacks. One of the drawbacks is that it includes a small and carefully defined hole for the transport of oil, which causes a high sensitivity to clogging and tolerances. In addition, this previously known valve causes a relatively slow coupling and decoupling, which is particularly noticeable in connection with gear shifting. Also, the design is sensitive to external disturbances, for example in the form of temperature changes and pollution such as, for example, dirt particles or coatings.

[0013] Another drawback is related to the hydraulic actuation of the engine brake system, which inherits with high compliance. High compliance leads to large valve lift deflection, which leads to increased valve load. And increased valve load leads back to higher compliance. In order to reduce hydraulic compliance, the hydraulic piston must be designed with a large diameter. The large diameter hydraulic piston takes a long time to attain its extended position. Therefore the system taught by the '385 patent is not suitable for use in reducing engine speed at an up-shift.

[0014] Another problem with such prior art engine brakes is that the normal operation of the exhaust valve is affected during brake operation. Clearance between the cam follower and camshaft is effectively reduced during brake operation. This means that the first lobe on the camshaft opens the exhaust valve further than normal for the exhaust stroke during engine brake operation. In some cases it is necessary to provide recesses in the pistons so that the exhaust valves do not strike the pistons when the brake is operational. These recesses, and the abnormally extended exhaust valves, interfere with optimal engine design from the point of view of other considerations such as emission controls.

[0015] An additional disadvantage of the known arrangement is that it does not have an easy way or a proper lash adjusting means to set the valve lash.

(c) Integrated Rocker Brake with One-Valve Opening for Engine Braking

[0016] Instead of opening two exhaust valves during engine braking, U.S. Patent No. 6,234,143 ("the '143 patent") discloses an integrated rocker brake with one-valve opening for engine braking. An engine brake actuator is disposed in the rocker arm between the pivot point and the distal end. The rocker arm and the valve bridge of the engine are so arranged that the hydraulic or braking piston of the brake actuator is able to actuate on the inner valve near the pivot point of the rocker arm. By actuating

only one exhaust valve, the engine braking load is greatly reduced.

[0017] The integrated engine brake system, however, has the following drawbacks. First, after the braking valve is lifted by the brake piston, the valve bridge is tilted and the followed normal valve actuation on both the braking valve and non-braking valve by the rocker arm is asymmetric or unbalanced. Large side load could be experienced on both valve stems or on the valve bridge guide if the bridge is guided. Second, the brake system can only fit on a particular type of engines that have the "parallel" arrangement of the rocker arm and the valve bridge.

(d) Integrated Rocker Brake with Reset Valve

[0018] U.S. Patent No. 6,253,730 ("the '730 patent") discloses an integrated rocker brake with a reset valve trying to avoid the asymmetric loading on the valves or the valve bridge caused by the engine braking operation as disclosed by the '143 patent. The reset valve will reset or retract the hydraulic piston in the rocker arm before the braking valve reaches its peak braking lift so that the braking valve will return back to its seat before the main valve lift event starts, and the rocker arm can act on the leveled valve bridge and open both the braking valve and the non-braking valve without any asymmetric loading.

[0019] However, resetting the braking valve lift around the compression TDC is very problematic. First, the duration and magnitude of the valve lift for engine braking is very small and even smaller for resetting. Second, the resetting happens at around the peak engine braking load and causes high pressure or large load on the reset valve. The timing for the resetting is critical. If the resetting happens too soon, there will be too much braking valve lift loss (lower lift and earlier closing) and lower braking performance. If the resetting happens too late, the braking valve will not be able to close before the main valve event starts and cause asymmetric loading. Therefore, the integrated rocker engine brake according to the '730 patent may not work well at high engine speeds when the reset duration and height is extremely small and the braking load or pressure on the reset valve is very high.

[0020] It is clear from the above description that the prior-art engine brake systems have one or more of the following drawbacks:

- (a) The system can only be installed on a particular type of engines.
- (b) The system has slow response (on & off) time.
- (c) The system is hydraulically driven and has large compliance resulting in high braking load.
- (d) The system causes asymmetric loading on valves or valve bridge guide.
- (e) The system has too many parts, high complexity, and not work well at high engine speeds.
- (f) The system has no easy way to set lash for engine braking valves.
- (g) The system is not reliable and sensitive to exter-

nal disturbances.

(h) The system affects normal engine performance (efficiency and emission).

5 SUMMARY OF THE INVENTION

[0021] The engine braking apparatus of the present invention addresses and overcomes the foregoing drawbacks of prior art engine braking systems.

10 **[0022]** One object of the present invention is to provide an engine braking apparatus that has fast response (on and off) time.

[0023] Another object of the present invention is to provide an engine braking apparatus with fewer components, reduced complexity, lower cost, and increased system reliability.

15 **[0024]** A further object of the present invention is to provide such an engine braking apparatus that contains a braking valve lash adjusting mechanism so that it does not increase the manufacturing tolerance requirements of many of the components.

20 **[0025]** Still a further object of the present invention is to provide an engine braking apparatus that is effective at all engine speeds and not sensitive to external disturbances.

25 **[0026]** Yet a further object of the present invention is to provide engine brake actuation means that transmit force, or the engine braking load, through mechanical linkage means that does not have high compliance and overloading problems associated with traditional hydraulic means used by prior art engine braking systems.

30 **[0027]** Still another object of the present invention is to provide an engine braking apparatus that will not affect the normal engine operation.

35 **[0028]** The engine braking apparatus of the present invention converts an internal combustion engine from a normal engine operation to an engine braking operation. The engine includes exhaust valve train components containing at least one exhaust valve and at least one cam for cyclically opening and closing the at least one exhaust valve.

40 **[0029]** The apparatus includes an engine brake actuation means having at least one component integrated into at least one of the exhaust valve train components, such as the rocker arm or the valve bridge. The actuation means has an inoperative position and an operative position. In the inoperative position, the actuation means is retracted and disengaged from the normal engine operation. In the operative position the actuation means is extended to open the at least one exhaust valve for the engine braking operation. The apparatus also has an engine brake control means for moving the engine brake actuation means between the inoperative position and the operative position to achieve the conversion between the normal engine operation and the engine braking operation.

45 **[0030]** The actuation means further includes mechanical linkage means for transmitting load generated by en-

engine braking operation. The mechanical linkage means includes at least one system selected from the group consisting of: a piston-sliding device, a ball-locking device, a wedge device, a toggle or linkage device, and a piston-coupling device.

[0031] The apparatus also includes a reset means for moving the actuation means from the operative position to the inoperative position during the engine braking operation so that the valve lift profile is reset to a smaller profile.

[0032] The engine braking apparatus according to the embodiments of the present invention have many advantages over the prior art engine braking systems, such as faster response; better performance, fewer components, reduced complexity, and lower cost.

BRIEF DESCRIPTION OF THE DRAWINGS

[0033] These and other advantages of the present invention will become more apparent from the following description of the preferred embodiments in connection with the following figures.

FIG. 1 is a flow chart illustrating the general relationship between a normal engine operation and an added engine braking operation according to one version of the present invention.

FIG. 2 is a function chart showing relationship between a normal engine operation and an added engine braking operation according to one version of the present invention.

FIG. 3 is a flow chart illustrating the engine braking operation control according to one version of the present invention.

FIGS. 4A and 4B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a first embodiment of the present invention.

FIGS. 5A and 5B are schematic diagrams of an engine brake control mean at its "On" position and its "Off" or draining position according to one version of the present invention.

FIGS. 6A and 6B are schematic diagrams of a solenoid valve at the "Off" and "On" positions according to one version of the present invention.

FIGS. 7A and 7B are schematic diagrams of a solenoid valve at the "Off" and "On" positions according to another version of the present invention.

FIGS. 8A and 8B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a second embodiment of the present invention.

FIG. 9 has exhaust valve lift profiles according to one version of the present invention.

FIG. 10 is a schematic diagram of an engine braking apparatus with a reset means.

FIG. 10A-A shows a cross section of the reset means in FIG. 7.

FIGS. 11A and 11B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a third embodiment of the present invention.

FIGS. 12A and 12B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a fourth embodiment of the present invention.

FIGS. 13A and 13B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a fifth embodiment of the present invention.

FIGS. 14A and 14B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a sixth embodiment of the present invention.

FIGS. 14C and 14D show details of the piston coupling device used in the embodiment shown in FIGS. 14A and 14B at the "Off" and "On" positions.

FIGS. 15A and 15B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a seventh embodiment of the present invention.

FIGS. 15C and 15D show details of the piston coupling device used in the embodiment shown in FIGS. 15A and 15B at the "Off" and "On" positions.

FIGS. 16A and 16B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to an eighth embodiment of the present invention.

FIGS. 17A and 17B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a ninth embodiment of the present invention.

FIGS. 18A and 18B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a tenth embodiment of the present invention.

FIG. 19 is a schematic diagram of an engine braking apparatus at the "On" position according to an eleventh embodiment of the present invention.

FIGS. 20A and 20B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a twelfth embodiment of the present invention.

FIGS. 20C and 20D are schematic diagrams of braking pistons used in the embodiment of FIGS. 20A and 20B.

FIGS. 21 A and 21B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a thirteenth embodiment of the present invention.

FIGS. 22A and 22B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a fourteenth embodiment of the present invention.

FIG. 23 is a schematic diagram of an engine braking apparatus at the "Off" position according to a fif-

teenth embodiment of the present invention.

FIGS. 24A and 24B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a sixteenth embodiment of the present invention.

FIGS. 24C and 24D are schematic diagrams of a braking actuation means used in the embodiment of FIGS. 24A and 24B.

FIGS. 25A and 25B are schematic diagrams of a braking actuation means at the "Off" and "On" positions according to another version of the present invention.

FIG. 26 is a schematic diagram of an engine braking apparatus at the "Off" position according to a seventeenth embodiment of the present invention.

FIG. 27 is a schematic diagram of a braking actuation means at the "Off" position according to yet another version of the present invention.

FIG. 28 is a schematic diagram of a braking actuation means at the "Off" position according to yet another version of the present invention.

FIG. 29 is a schematic diagram of an engine braking apparatus at the "Off" position according to an eighteenth embodiment of the present invention.

FIG. 30 is a schematic diagram of an engine braking apparatus at the "On" position according to a nineteenth embodiment of the present invention.

FIG. 31 is a schematic diagram of an engine braking apparatus at the "On" position according to a twentieth embodiment of the present invention.

FIG. 32 is a schematic diagram of an engine braking apparatus at the "Off" position according to a twenty first embodiment of the present invention.

FIG. 33 is a schematic diagram of an engine braking apparatus at the "Off" position according to a twenty second embodiment of the present invention.

FIG. 34 is a schematic diagram of an engine braking device at the "Off" position according to a twenty third embodiment of the present invention.

FIG. 35 is a schematic diagram of an engine braking device at the "Off" position according to a twenty fourth embodiment of the present invention.

FIG. 36 is a schematic diagram of an engine braking device at the "Off" position according to a twenty fifth embodiment of the present invention.

FIGS. 37A and 37B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a twenty sixth embodiment of the present invention.

FIG. 38 is a schematic diagram of an engine braking apparatus at the "On" position according to a twenty seventh embodiment of the present invention.

FIG. 39 is a schematic diagram of an engine braking apparatus at the "On" position according to a twenty eighth embodiment of the present invention.

FIGS. 40A and 40B are schematic diagrams of an engine braking apparatus at the "On" position according to a twenty ninth embodiment of the present

invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0034] Reference will now be made in detail to presently preferred embodiments of the invention, examples of which are illustrated in the accompanying drawings. Each example is provided by way of explanation, not limitation, of the invention. In fact, it will be apparent to those skilled in the art that modifications and variations can be made in the present invention without departing from the scope and spirit thereof. For instance, features illustrated or described as part of one embodiment may be used on another embodiment to yield a still further embodiment. Thus, it is intended that the present invention covers such modifications and variations as come within the scope of the appended claims and their equivalents.

[0035] FIG. 1 is a flow chart illustrating the general relationship between a normal engine operation 20 and an added engine braking operation 10. An engine usually contains two exhaust valves 300 and an exhaust valve actuation means (or lifter) 200 for cyclically opening and closing the exhaust valves 300 during the normal engine operation 20. The engine braking operation 10 is achieved by turning on an engine brake control means 50 to move the engine brake actuation means 100 from an inoperative position 0 to an operative position 1. By default, the engine brake control means 50 is off, the engine brake actuation means 100 stays at the inoperative position 0, and the engine is at the firing position 2.

[0036] FIG. 2 is a function chart illustrating the general relationship between the normal engine operation 20 and the added engine braking operation 10 according to one version of the present invention. For the normal engine operation 20, the small cam lobe(s) on the exhaust cam 230 are skipped, as shown in block 240, due to a gap 234 among the valve train components, to produce the main exhaust valve lift profile 220m for the normal engine valve event 20N. For the engine braking operation 10, the engine brake control means 50 controls the motion of the engine brake actuation means 100 between an inoperative position 0 and an operative position 1. At the inoperative position 0, the actuation means 100 retracts to form the gap 234, while at the operative position 1 (the control means 50 is turned on), the actuation means 100 extends to take up the gap 234 as shown in block 120. Without the gap 234, motion from all the cam lobes, small and large, is picked up by the rocker arm as shown in block 125. The braking valve lift profile, however, depends on whether there is an engine brake reset means 150.

[0037] If there is no engine brake reset means, motion from all the cam lobes will be transmitted to the engine valve(s) to generate the engine valve lift profile 220v for the engine braking valve event 10B. But with the engine brake reset means 150, the engine brake actuation means 100 will be temporarily switched from the extend-

ed position to the retracted position during each cycle of the engine braking operation 10, which will truncate the valve lift profile from the large cam lobe to generate the engine valve lift profile 220h for the engine braking valve event 10R. Note that the reset means 150 starts when the cam lift gets into the higher portion of the large cam lobe, which is higher than the small cam lobes. Therefore, only the higher portion of the large valve lift profile is truncated. Once the cam lift is back into the lower portion of the large cam lobe, which is below the height of the small cam lobes, the reset means 150 is disengaged and the engine brake actuation means 100 is extended to the operative position again to take up the gap 234 before the small cam lobes start so that the secondary valve lift profile is retained.

[0038] FIG. 3 is a flow chart illustrating the engine braking operation control according to one version of the present invention. It is assumed that the control starts with the normal engine operation block 710. The next control block 720 determines whether engine braking is desired. If it is not, the engine brake control means 50 is turned off, as shown in control block 722, and the engine brake actuation means 100 retracts to the inoperative position 0 (control block 724) to skip all the small cam lobes (control block 726) to produce only the main valve lift profile in control block 728 for the normal engine operation 20.

[0039] If engine braking is needed, the engine brake control means 50 will be turned on, as shown in control block 730, and the engine brake actuation means 100 will be extended to form a mechanical linkage, as shown in control block 740, so that all cam motion is picked up by the rocker arm and the integrated engine brake actuation means. The next control block 750 determines if there is an engine brake reset means. If there is no reset means, a full valve lift profile is generated from both the large and small cam lobes, as shown in control block 760. Now the control goes back to the block 720 to start a new cycle of engine braking control.

[0040] If the control block 750 shows that there is an engine brake reset means, then the next control block will be 770 in which the reset means 150 retracts the engine brake actuation means 100 so that the valve lift profile from the large cam lobe is truncated. The resetting happens during the higher portion of the large valve lift profile. Once the valve lift gets back to the lower portion of the large valve lift profile, the reset means 150 is disengaged and the actuation means 100 is extended again to form the mechanical linkage, which happens before the small cam lobe starts, as shown in control block 780. Therefore, the reset means 150 works with the engine brake actuation means 100 to produce a truncated large valve lift profile and the full secondary valve lift profile from the small cam lobes, as shown in control block 790. The engine braking control now goes back to block 720 and the control cycle repeats.

[0041] FIGS. 4A and 4B are schematic diagrams of an engine braking apparatus at the "Off" and "On" position

according to one embodiment of the present invention. The engine brake actuation means 100 is integrated into a rocker arm 210 of the engine exhaust valve train or the valve lifter 200. The valve train has components that include a cam 230, a cam follower 235, the rocker arm 210, a valve bridge 400, and the exhaust valves 300a and 300b (or simply 300). The exhaust valves 300 are biased upwards against their seats 320 on the engine cylinder head 500 by engine valve springs 310a and 310b (or simply 310) to seal gas from flowing between the engine cylinder (not shown) and the exhaust manifolds 600. The rocker arm 210 is pivotally mounted on a rocker shaft 205 for transmitting mechanical input or motion from the cam 230 to the exhaust valves 300 for their cyclical opening and closing.

[0042] There may be other valve train components that are not shown here for simplicity, such as an elephant foot that may be attached to the lower portion 162 of the braking piston 160 (FIG. 4B). The cam 230 contains a large lobe 220 above the inner base circle (IBC) 225 mainly for the normal engine operation and two small lobes 232 and 233 for the engine braking operation. The rocker arm 210 is biased against the valve bridge 400 by a spring 198, and a gap 234 is formed between the cam 230 and the cam follower 235 when the engine brake is not turned on (FIG. 4A). The gap 234 is set by a lash adjusting mechanism to such a height that the small cam lobes will be skipped when the engine brake is not needed. The lash adjusting screw 110 is secured on the rocker arm 210 by a lock nut 105 and is also part of the engine brake actuation means 100. Due to the gap 234 among the valve train components, a spring means that may include the spring 198 and its assembly or mounting. The spring 198 is so designed that its preload will be high enough to prevent any of the valve train components from no-following even at the highest engine speed, but at the same time, be low enough to allow the engine brake actuation means 100 to be turned on when needed. One end of the spring 198 is mounted on the engine or a fixed component of the engine, and the other end of the spring 198 is mounted on one of the valve train components, such as the top 215 of the rocker arm 210.

[0043] The engine brake actuation means 100 is a ball-locking device with a plurality of balls 175 restrained by three surfaces on three elements, as shown in FIG. 4B. The first surface is the tapered surface 192 on the bottom of the lash adjusting screw 110. The second surface is the flat surface on the top of a braking piston 160 that is slidably disposed in a bore 190 of a ball-locking piston 165. The stroke of the braking piston 160 is 195, which takes up the gap 234 (FIG. 4B). The third surface is either on the annular groove 170 when the ball-locking device is at the retracted or "Off" position as shown in FIG. 4A or on the bore 190 when the ball-locking device is at the extended or "On" position as shown in FIG. 4B.

[0044] The movement of the engine brake actuation means 100 is controlled by the engine brake control means 50 as shown in FIGS. 5A and 5B, which is shown

as an electro-hydro-mechanical system (or a three-way flow control valve) containing a coil structure 51. The flow control valve 50 has a spool 58 and is turned on and off by an electric current through the positive and negative terminals 55 and 57. As the spool 58 slides, it opens or closes a port (an orifice or a drill) 111 or 222 to allow hydraulic fluid, for example, engine lube oil, into or out of an engine braking fluid circuit containing a flow passage 211 and a radial orifice 212 in the rocker shaft 205, an undercut 213 and a flow passage 214 in the rocker arm 210, and a slot or undercut 180 on the ball-locking piston 165 (FIG. 4B). Note that the engine brake control means 50 could be remotely located and used for controlling engine brakes over multiple engine cylinders and the braking fluid circuit may reach other components of the engine and of the actuation means 100.

[0045] When engine brake is needed, the engine brake control means 50 is turned on (FIG. 5A) and the engine oil is transmitted to the engine brake actuation means 100 through the braking fluid circuit. FIG. 4B shows that the engine oil from flow passage 214 can get to the bottom of the lash adjusting screw 110 because its stem 191 is smaller than the bore 190 of the ball-locking piston 165 in which the braking piston 160 slides. Oil pressure overcomes the force of spring 198 and pushes up the rocker arm 210 for a clockwise rotation to take up the gap 234 between the cam 230 and the cam follower 235 (FIG. 4B). As the lash adjusting screw 110 moves up along with the rocker arm 210, the balls 175 move inwards along the tapered surface 192 and out of the annular groove 170 in the ball-locking piston 165. Now the ball-locking piston 165 can move down in a bore 260 in the rocker arm 210, since the oil pressure overcomes the force of spring 177 on spring seat 176. Once the ball-locking piston 165 is stopped on the shoulder of the brake piston 160, the ball-locking device is locked at its extended position or the operative position as shown in FIG. 4B, which takes up the gap 234 and forms a mechanical linkage. Without the gap 234, all the motion from the cam 230 is transmitted to the exhaust valves 300 to produce an enlarged main valve lift profile and a secondary lift profile for the engine braking operation.

[0046] When engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure acting on the ball-locking piston 165, which will be pushed upwards by the spring 177 towards the top of the bore 260. Once the annular groove 170 in the ball-locking piston 165 is aligned with the balls 175, they will be pushed outwards and into the annular groove 170 by the downward motion of the tapered surface 192 on the lash adjusting screw 110 under the force of spring 198. Now the ball-locking device is at the retracted position or the inoperative position and the gap 234 between the cam 230 and the cam follower 235 is formed to skip part of the cam motion, i.e., the lower portion of cam 230 shown in FIG. 4A to produce the main valve lift profile for the normal engine operation.

[0047] It can be seen that the present invention pro-

vides engine brake actuation means that transmits force, or the engine braking load, through mechanical linkage means that does not have high compliance and overloading problems associated with traditional hydraulic means used by the prior art engine braking systems. Therefore, there will be much less valve lift loss due to lower compliance. Both the stroke and the diameter of the braking piston 160 can be designed much smaller than the prior art with hydraulic means, which will greatly reduce the engine braking response time, the moment of inertia and the effect of excessive high valve lift on engine operation. Also, the gap 234 among the valve train components will be smaller, which leads to less potential of no-follow of the valve train components.

[0048] FIGS. 6A and 6B are schematic diagrams of the flow control valve 50 at the "Off" and "On" positions according to one version of the present invention. It has a coil structure 51 on top of the valve and a valve body 60 attached to the coil structure. The coil structure is well known and its details are not shown here for simplicity. There is a groove 62 on the valve body 60, which is used for mounting of the valve so that its orientation is not limited.

[0049] The valve body 60 has the inlet port 111, the outlet port 211 and the discharge port 222. A ball 67 is mounted in a bore 75b and separates the inlet port 111 and the outlet port 211. A movable valve member, for example, a cylindrical body 64 is disposed in a bore 75a to separate the outlet port 211 and the discharge port 222. A plunger 66 is disposed between the cylindrical body 64 and the ball 67. The three ports are separated or sealed from each other by O-rings 65 and 68. A screen 69 can be installed before the inlet port 111 to prevent any contaminants from getting into the valve.

[0050] Oil pressure from the inlet port 111 keeps the ball 67 to its seat 73b and pushes the cylindrical body 64 away from its seat 73a. The flow control valve 50 is now at the "Off" position as shown in FIG. 6A. Oil from the inlet port 111 can't get into the outlet port 211, while oil from the outlet port 211 can flow to the discharge port 222.

[0051] When engine braking is needed, electric current is supplied to the coil through terminal 55 and 58. Electromagnetic force pushes an actuation pin 63 downward and the cylindrical body 64 to its seat 73a. At the same time, the ball 67 is pushed away from its seat 73b and opening against the inlet oil pressure. The flow control valve 50 is now at the "On" position as shown in FIG. 6B. Oil from the inlet port 111 can flow into the outlet port 211, while oil from the outlet port 211 is blocked from flowing to the discharge port 222.

[0052] FIGS. 7A and 7B are schematic diagrams of the flow control valve 50 at the "Off" and "On" positions according to another version of the present invention. The only difference between this control valve and the previous one is that the cylindrical body 64 is replaced with a ball and the plunger 66 is changed to a guided pin.

[0053] FIGS. 8A and 8B show a different version of the embodiment in FIGS. 4A and 4B with an added engine

brake reset means 150 to interact with the engine brake actuation means 100. The reset means 150 comprises a reset piston 166 that is slidably disposed in a reset bore 169 in the rocker arm 210. During the normal engine operation, the reset piston 166 is biased up to the top of the reset bore 169 (FIG. 8A) by a spring 199 that is secured to the rocker arm 210 by a screw 179 (FIG. 8B). The gap 185 between the reset piston and the engine block is so designed that the reset piston 166 will not touch the engine block during the whole cam rotation when engine brake is not actuated (FIG. 8A).

[0054] With the reset means 150, the electro-hydro-mechanical system of the engine brake control means 50, as shown in FIGS. 5A and 5B, does not need to have a three-way solenoid valve 51 because the reset means 150 is also a flow draining means and will drain the engine oil in the engine brake actuation means 100 to turn off the engine brake when needed. Therefore there is no need for the drain port 222, and the three-way solenoid valve 51 can be replaced by a two-way solenoid valve to open and close the oil supply port 111.

[0055] During the engine braking operation, oil is transmitted to the higher chamber over the top of the reset piston 166 through a flow path 214a as shown in FIG. 8B. Oil pressure overcomes the force of spring 199 and pushes the reset piston 166 down to a stop 178, which allows oil flow to the ball-locking device through the flow path 214 but blocks the drain passage 167. The gap 185 between the reset piston 166 and the engine block is reduced but still large enough that the rocker rotation by the small cam lobes 232 and 233 will not reset the engine brake actuation means 100. Only during the anticlockwise rocker arm rotation by the higher portion of the large cam lobe 220, the reset piston 166 will touch the engine block and stop moving down while the reset bore 169 continues the downward motion with the rocker arm 210. The reset piston 166 will block the flow passage 214a and connect the flow passage 214 to the drain passage 167 to release oil pressure from the engine brake actuation means 100. Without oil pressure, the ball-locking piston 165 will be pushed upwards by the spring 177 towards the top of the bore 260 in the rocker arm 210 and unlock the ball-locking device to the retracted position as shown in FIG. 8A. A portion of the cam lift equal to the gap 234 will be skipped or lost due to the resetting, and the valve train will get shorter so that the enlarged main valve lift profile is truncated back to the main valve lift profile. When the cam rotation passes the peak of the large cam lobe 220, the rocker arm 210 will rotate clockwise and move away from the engine block so that the reset piston 166 will slide down in the reset bore 169 under the oil pressure. When the cam lift gets into the bottom part of the enlarged cam lobe 220 or below the peak lift of the small lobes 232 and 233, the drain passage 167 is blocked and the reset mean 150 is disengaged. The oil supply to the ball-locking device is resumed from the passage 214a to the passage 214. Under oil pressure, the ball-locking device is extended and locked up

again to the operative position, and the gap 234 between the cam 230 and cam follower 235 is taken up, which happens on IBC 225 and before the small cam lobe 232. Therefore, with the reset means 150, the engine valve lift for the engine braking operation will have all the valve lifts from the small cam lobes 232 and 233 but a truncated valve lift from the large cam lobe 220.

[0056] FIG. 9 illustrates the engine exhaust valve lift profiles according to one version of the present invention.

The main valve lift profile 220m is for the normal engine operation and the enlarged main valve lift profile 220v plus the secondary valve lift profile with valve lifts 232v and 233v is for the engine braking operation when there is no engine brake resetting. There is also a hybrid valve lift profile for the engine braking operation, which is obtained with the engine brake reset means 150.

[0057] During the normal engine operation, the valve lift 220a from the lower portion of cam 230, plus 232v and 233v from the small cam lobes 232 and 233, is skipped due to the gap 234 among the valve train components. Only the higher portion 220b is transmitted to the engine valves 300 to generate the main valve lift profile 220m which starts at point 225a and ends at point 225b with a peak lift of 220b. The lower portion 220a and the higher portion 220b are divided by the transition line passing through the transition point 220t. The height 232p of the lower portion 220a is close to that of the valve lifts 232v and 233v, while the higher portion 220b is about the same as the main valve lift profile 220m.

[0058] During the engine braking operation, the engine brake actuation means 100 is extended and the gap 234 among the valve train components is taken up. All the motion from the cam 230 can be transmitted to the exhaust valves 300. However, the valve lift profile depends on the existence of the reset means 150. If there is no reset means as shown in FIGS. 4A and 4B, then the valve lift profile will start at point 225d as shown in FIG. 9, go over the braking gas recirculation (BGR) bump 232v, be followed by the compression release braking (CRB) bump 233v, then pass the transition point 220t between the lower portion 220a and the higher portion 220b, move up to the reset point 220r (but no resetting) and over the peak 220e of the enlarged main valve lift profile 220v, finally close at point 225c with zero valve lift.

[0059] If there is an engine brake reset means 150 as shown in FIGS. 8A and 8B, then the valve lift profile during the engine braking operation will be the same as the no-reset braking valve lift profile until it hits the reset point 220r (FIG. 9). Then the valve lift will drop back from the reset point 220r on the enlarged main valve lift profile 220v to the point 220s on the main valve lift profile 220m, and finally close at point 225b, much earlier than the point 225c. Theoretically, the reset point 220r can be anywhere between the transition point 220t and the peak enlarged valve lift 220e. But making the reset point 220r closer to the peak enlarged valve lift 220e reduces the oil consumption and the reset piston travel.

[0060] The engine brake reset means 150 according

to the present invention eliminates the drawbacks of those disclosed by the prior art, for example, the '730 patent. First, the timing and magnitude (or height) of the resetting is not critical. The resetting does not happen during the engine braking lift profile 233v, but during the higher portion 220b of the enlarged main valve lift profile 220v. Second, there is no high oil pressure or large load acting on the reset valve or piston because the engine braking load from the current engine brake system is not supported by a hydraulic means but a mechanical linkage means. Resetting is basically decoupling or disengaging the mechanical linkage. Therefore, the reset means disclosed here is more reliable, more tolerant to variation and easier to design and manufacture.

[0061] FIG. 10 and its cross-section drawing FIG. 10A-A show a different version of the embodiment in FIGS. 8A and 8B with an added oil retaining means 350 to the reset means 150. The oil retaining means 350 comprises an oil retaining piston 155 that is biased downwards by a spring 156 to seal a drain orifice 167a. The spring 156 is seated on a spring seat 158 and the piston 155 is slidably disposed in a bore 154 in the rocker arm 210. The oil retaining means 350 is designed to keep engine oil in the engine brake fluid circuit mainly for lubrication purpose.

[0062] Two levels of oil supply pressure could be provided to the engine braking fluid circuit. During the engine braking operation, the engine lube oil with full supply pressure (for example, 30 psi gage) flows into the braking circuit to actuate the engine braking means 100, while during the normal engine operation, oil with a lower level pressure (for example, 5 psi gage) is not able to actuate the engine brake actuation means 100, the reset piston 166, nor the oil retaining piston 155. However, the oil can still flow through the orifice 152 in the reset piston 166 (FIG. 10) and into the engine brake actuation means 100 for system lubrication. Keeping the engine oil in the engine brake fluid circuit also makes the engine braking operation turn on faster. In another word, it reduces engine braking control response time.

[0063] During the engine braking operation, oil released from the actuation means 100 by the reset means 150 has enough pressure to push the oil retaining piston 155 upwards against the spring 156 and open the drain hole 167a so that oil can flow from the actuation means 100 to the ambient through the flow passages 214, 167 and 167a to complete the engine brake resetting process.

[0064] FIGS. 11A and 11B show another embodiment of the present invention with a different ball-locking device. Again, the balls 175 are restrained by three surfaces on three different elements of the engine brake actuation means 100. The first surface is a tapered surface on the braking piston 160. The second is the bottom flat surface on the adjusting screw 110, and the third is either the small diameter surface of the ball-locking piston 165 when the ball-locking device is at the retracted position (FIG. 11A) or the larger diameter surface when the ball-locking device is at the extended position (FIG. 11B). As

with the previous embodiments, the lash adjusting mechanism is incorporated into the engine brake actuation means 100. A washer can be added between the screw 110 and the balls 175 to reduce the size of the screw 110.

[0065] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the engine brake actuation means 100 through the engine brake fluid circuit. Oil pressure overcomes the force of spring 198 and pushes up the rocker arm 210 for a clockwise rotation to take up the gap 234 between the cam 230 and the cam follower 235 as shown in FIG. 11A. As the lash adjusting screw 110 moves up along with the rocker arm 210, the balls 175 move up and outwards along the tapered surface on the braking piston 160. The ball-locking piston 165 also moves up with the lash adjusting screw 110. When the balls 175 are out of the way, the ball-locking piston 165 moves up further into the bore in the lash adjusting screw 110 with the oil pressure overcoming the force of spring 177. Once the ball-locking piston 165 is stopped on the lash adjusting screw 110, the ball-locking device is locked to the extended or operative position to form a mechanical linkage, as shown in FIG. 11B. The motion from the whole cam 230 picked up by the rocker arm 210. But due to the engine brake reset means 150, a portion of the cam lift equal to the gap 234 will be truncated from the higher portion of the enlarged cam lobe 220 so that the engine valve lift for the engine braking operation will have all the valve lifts from the small cam lobes 232 and 233 but a truncated valve lift from the enlarged cam lobe 220. If there is no engine brake reset means, then the full cam motion from all the cam lobes, large and small, is transmitted to the exhaust valves 300 to produce an enlarged main valve lift profile and a secondary lift profile for the engine braking operation.

[0066] When engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure acting on the ball-locking piston 165, which will be pushed down towards the braking piston 160 by the spring 177. Note that there is an orifice at the top of the lash adjusting screw 110 to eliminate hydraulic lock. Once the ball-locking piston 165 is down against the braking piston 160, the balls 175 will move down and inwards along the tapered surface on the braking piston 160, and the lash adjusting screw 110 can move down with the rocker arm 210 under the force of spring 198. Now the ball-locking device is at the retracted or inoperative position and the gap 234 between the cam 230 and the cam follower 235 is formed to skip the lower portion of the cam 230 including the small cam lobes 232 and 233 to produce the main valve lift profile for the normal engine operation.

[0067] FIGS. 12A and 12B show an embodiment of the engine brake actuation means 100 with another ball-locking device in the rocker arm 210 and over the valve bridge 400. The balls 175 are always restrained by holes in the braking piston 160 that is normally retracted in the bore 190 under the load of spring 198. The ball-locking

piston 165 is biased to the bottom of 260 in the braking piston 160 by the spring 177 that has a seat 176 mounted on the rocker arm 210 with a screw 179.

[0068] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the engine brake actuation means 100 through the engine brake fluid circuit. Oil pressure overcomes the force of spring 198 and pushes up the rocker arm 210 for a clockwise rotation to take up the gap 234 between the cam 230 and the cam follower 235, as shown in FIG. 12A. The annular groove 170 in the rocker arm 210 will align with the balls 175 that will move outwards and into the groove 170 under the urge of the upward motion of the ball-locking piston 165. Note that the braking piston 160 is pushed against the valve bridge 400 and does not move when the cam 230 is at the IBC 225. Once the balls 175 are in the groove 170, the ball-locking piston 165 will slide up in the bore 260 in the braking piston 160 because oil gets to the bottom from the flow passage 196 and the oil pressure overcomes the force by spring 177. Once the ball-locking piston 165 is at the top of the bore 190 in the rocker arm 210, the balls 175 are locked into the groove 170 by the larger outer diameter of the ball-locking piston as shown in FIG. 12B. Now the ball-locking device is at the extended position with a stroke or travel 195 that will take up the gap 234 and form a mechanical linkage. The motion from the whole cam 230 is transmitted to the exhaust valves 300 to produce an enlarged main valve lift profile and a secondary lift profile for the engine braking operation. A reset means can be easily added to modify the enlarged main valve lift.

[0069] When engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure acting on the ball-locking piston 165, which will be pushed down to the bottom of the bore 260 in the braking piston 160 by the spring 177. Once the ball-locking piston 165 is down against the braking piston 160, the balls 175 can move inwards and out of the annular groove 170, and the rocker arm 210 will move down under the force of spring 198. Now the ball-locking device is at the retracted position and the gap 234 between the cam 230 and the cam follower 235 is formed to skip part of the cam motion, i.e., from the lower portion of the cam 230 including the small cam lobes 232 and 233 shown in FIG. 12A to produce the main valve lift profile for the normal engine operation.

[0070] FIGS. 13A and 13B show a similar embodiment to that shown in FIGS. 12A and 12B except that the ball-locking piston 165 and spring 177 are fully contained in the bore 190 in the rocker arm 210. The flow orifice 168 is added to eliminate the hydraulic lock, which enables the motion of the ball-locking piston 165 in the bore 260. The flow passage or orifice 196 is optional and can be eliminated. However, without the orifice 196, a three-way solenoid valve is needed to turn off the engine brake.

[0071] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the engine brake actuation means 100

through the engine brake fluid circuit. Oil pressure overcomes the force of spring 198 and pushes up the rocker arm 210 for a clockwise rotation to take up the gap 234 between the cam 230 and the cam follower 235, as shown in FIG. 13A. As the rocker arm 210 moves up, the flow orifice 168 will be uncovered, and the annular groove 170 aligned with the balls 175 that will move outwards and into the groove 170 under the urge of the downward motion of the ball-locking piston 165. Once the balls 175 are in the groove 170, the ball-locking piston 165 will move down because the oil pressure overcomes the force of spring 177. The balls 175 are locked into the groove 170 by the larger outer diameter surface of the ball-locking piston 165. The oil flow through the orifice 168 is blocked when the ball-locking piston 165 sits on the braking piston 160 to reduce oil consumption. As shown in FIG. 13B, the ball-locking device is now at the extended position with a stroke or travel 195 that will take up the gap 234 to form a mechanical linkage. Without the gap 234, all cam motion is transmitted to the exhaust valves 300 to produce an enlarged main valve lift profile and a secondary lift profile for the engine braking operation.

[0072] When engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure acting on the ball-locking piston 165, which will slide up in the braking piston 160 under the force of spring 177. The balls 175 will move inwards and out of the annular groove 170, and the rocker arm 210 will move down under the force of spring 198. Now the ball-locking device is at the retracted position and the gap 234 between the cam 230 and the cam follower 235 is formed to skip the lower portion of the cam 230 including the small cam lobes 232 and 233, as shown in FIG. 13A.

[0073] FIGS. 14A and 14B show an embodiment of the engine brake actuation means 100 with a piston-coupling device 123 in the rocker arm 210 whose details are shown in FIGS. 14C and 14D. There are three pistons 164a, 164b and 164c slidably disposed in the bores 183a, 183b and 183c of three sleeves 163a, 163b and 163c. Sleeve 163b is fixed in the braking piston 160 while sleeves 163a and 163c are fixed in the rocker arm 210. Sleeves 163a and 163b have a step or a half-cut 138a and 138b (FIG. 14C) so that they can be easily aligned (FIGS. 14B and 14D). Also, the step 138a on sleeve 163a protrudes out of the bore 190 and fits into an axial groove or cut 138 on the braking piston 160 as a guide.

[0074] During the normal engine operation, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure to actuate the actuation means 100. The three pistons 164a, 164b and 164c are biased to the right against the sleeve 163c by the spring seat 178b that is slidably disposed in the sleeve 163a and loaded by the spring 177. The pistons 164a and 164b are now contained in the sleeve 163b and can slide upward in the bore 190 with the braking piston 160 to the inoperative position. The stroke of the braking piston is 195, which is equal to the valve lift by the braking cam

lobes 232 and 233. Part of the motion, i.e., from the lower portion of the cam 230 will not be transmitted to the valves 300 but absorbed by the relative motion of the braking piston 160 in the bore 190 in the rocker arm 210 (FIG. 14A). Only the remaining part of the motion, i.e., from the higher portion of the enlarged cam lobe 220 is transmitted to the exhaust valves 300 for the normal engine operation.

[0075] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the engine brake actuation means 100. The spring 177a biases the braking piston 160 down toward the valve bridge 400, which is stopped when the step 138a of sleeve 163a contacts the step 138b of the sleeve 163b. Now the sleeves are aligned to each other, as shown in FIGS. 14B and 14D. Oil pressure overcomes the force of spring 177 and pushes the pistons 164a, 164b and 164c to the left and stopped by the spring seat 178b on the sleeve 163a. Now the braking piston 160 cannot move up in the bore 190 in the rocker arm 210 but locked to the operative position. A mechanical linkage is formed by the coupled pistons and sleeves as shown in FIG. 14D. All the cam motion from the small and large cam lobes is transmitted to the exhaust valves 300 for the engine braking operation.

[0076] FIGS. 15A and 15B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to a variation from the embodiment shown in FIGS. 14A and 14B. The rocker arm 210 is biased down against the braking piston 160 to the valve bridge 400 by a spring 198 mounted on the rocker arm top 215 so that a gap 234 is formed between the cam 230 and the cam follower 235 when the engine brake is at the "Off" or inoperative position as shown in FIG. 15A. The motion of the lower portion of the cam 230 including the small braking cam lobes 232 and 233 will be skipped. Only the higher portion of the enlarged cam lobe 220 is transmitted to the exhaust valves 300 for the normal engine operation.

[0077] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the top of the braking piston 160 through the braking fluid circuit that further includes the flow passage 217 around the sleeve 163c, the flow passage 113 in the braking piston 160, the orifices 1970 in the sleeve 163b (FIG. 15C), the annular groove 197g on the piston 164b, and the orifice 197 in the braking piston 160. Oil pressure overcomes the force of spring 198 and pushes the rocker arm 210 up to rotate clockwise. The rocker arm 210 will stop the upward motion when the step 138a on the sleeve 163a contacts the step 138b on the sleeve 163b. The total travel or stroke of the braking piston 160 in the rocker arm 210 is 195, which will take up the gap 234 between the cam 230 and the cam follower 235. Now all the sleeves as well as the pistons are aligned, as shown in FIGS. 15B and 15D. Oil pressure overcomes the force of spring 177 and pushes the pistons 164a, 164b and 164c to the left and stopped by the spring seat

178b on the sleeve 163a. The braking piston 160 cannot move up in the rocker arm 210 but locked to the operative position. A mechanical linkage is formed by the coupled pistons and sleeves as shown in FIG. 15D. All the cam motion is transmitted to the exhaust valves 300 for the engine braking operation.

[0078] If the engine brake actuation means 100 is reset or turned off, the oil pressure on the piston 164c will drop faster than that on the braking piston 160 because orifices 1970 in the sleeve 163b are blocked by the piston 164b. Higher oil pressure above the braking piston 160 pushes the steps 138a and 138b on the sleeves 163a and 163b against each other and helps reducing the friction force on the sliding pistons 164a and 164c so that the force of the spring 177 is high enough to push the pistons right to the decoupled or inoperative position. Then the groove 197g in the piston 164b will align with the orifices 1970 in the sleeve 163b and the oil above the braking piston 160 can flow out so that the braking piston 160 will return to the inoperative position as shown in FIGS. 15A.

[0079] FIGS. 16A and 16B show a similar embodiment to that shown in FIGS. 12A and 12B except that the engine brake actuation means 100 is integrated into the valve bridge 400, not in the rocker arm 210. The engine brake reset means 150 is now a part of the actuation means 100, which includes a ball-locking piston 165 and a reset stop 182. The ball-locking piston 165 can slide in the bore 260 in the braking piston 160. The reset stop 182 is below the ball-locking piston 165 and fixed on the engine. The lash adjusting mechanism includes a lash adjusting screw 110 secured on the rocker arm 210 by a lock nut 105.

[0080] During the normal engine operation or when engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there is little or no oil pressure acting on the engine brake actuation means 100. The rocker arm 210 is biased against the braking piston 160 towards the valve bridge 400 by the spring 198. The engine brake actuation means 100 is at the inoperative position. A gap 234 is formed between the cam 230 and the cam follower 235 as shown in FIG. 16A, and part of the cam motion, i.e., from the small cam lobes 232 and 233 is skipped. Only the remaining part of the motion, i.e., from the higher portion of the enlarged cam lobe 220 is transmitted to the exhaust valves 300 to produce the main valve lift profile. At the same time, the ball-locking piston 165 is biased up by a spring 177r and a gap 185 is formed between the ball-locking piston 165 and the reset stop 182. The gap 185 is so designed that the ball-locking piston 165 will not touch the reset stop 182 during the normal engine operation.

[0081] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to supply engine oil to the underneath of the braking piston 160 through the engine braking fluid circuit including the flow passage 115 in the lash adjusting screw 110, an orifice 197 on top of the engine braking piston 160, and a flow passage 196 in the ball-locking piston 165 (FIG. 16A).

Oil pressure overcomes the force of spring 198 and pushes up the braking piston 160 with the rocker arm 210 pivoting clockwise on the rocker shaft 205 to take up the gap 234. As the braking piston 160 slides up in the bore 190 in the valve bridge 400, the balls 175 will align with and move into the annular groove 170 in the valve bridge 400 under the urge of the ball-locking piston 165 that is forced down by the oil pressure overcoming the force of spring 177r mounted on the valve bridge 400 by a screw 179. The ball-locking piston 165 is now seating on the bottom of the bore 190 in the valve bridge 400, and the balls 175 are locked into the groove 170 by the larger outer diameter surface of the ball-locking piston 165 (FIG. 16B). Now the ball-locking device is locked to the extended position or operative position with a lift 195 that is designed to take up the gap 234 to form a mechanical linkage. The motion from the whole cam 230 is picked up by the rocker arm 210, but not necessarily transmitted to the exhaust valves 300 due to the engine brake reset means 150.

[0082] The maximum downward motion of the valve bridge 400 and the braking piston 160 by the enlarged cam lobe 220 is larger than the gap 185. The ball-locking piston 165 in the braking piston 160 will touch the reset stop 182 and stop moving downward before the valve bridge 400 reaches its maximum lift. Therefore, the ball-locking piston 165 is also the resetting piston. A relative motion is created between the ball-locking piston 165 and the braking piston 160 and the ball-locking device is unlocked from the extended (operative) position back to the retracted (inoperative) position. The braking piston 160 drops to the bottom of the bore 190 in the valve bridge 400 and a portion of the valve lift equal to the gap height 195 (FIG. 16B) will be truncated or lost to switch the enlarged main valve lift profile to the main valve lift profile. Once the cam rotation passes the large cam lobe 220, the rocker arm 210 will pivot clockwise, the valve bridge 400 and the braking piston 160 will move up. The ball-locking piston 165 will separate from the reset stop 182. When the cam lift gets into the bottom part of the enlarged cam lobe 220 or below the peak lift of the small lobes 232 and 233, the ball-locking device will be extended and locked to the operative position again when the cam 230 rotates on the IBC 225 in front of the small cam lobe 232. Therefore, with the reset means 150 the engine valve lift profile for the engine braking operation will have all the valve lifts from the small cam lobes 232 and 233 but a truncated valve lift from the enlarged cam lobe 220.

[0083] The engine brake reset means 150 may work without the reset spring 177r because the ball-locking piston 165 can be unseated by the reset stop 182 to reset and turn off the engine brake actuation means 100. When the ball-locking piston 165 is unseated, there may be oil leakage through the annular gap between the small piston or stem of the ball-locking piston 165 and the bore 450 in the valve bridge 400. The engine brake reset means 150 can also be disabled by removing the reset stop 182, then the motion of the whole cam is transmitted

to the exhaust valves 300 to produce an enlarged main valve lift profile and a secondary valve lift profile for the engine braking operation. Without the reset stop 182, the reset spring 177r is needed to unlock the ball-locking device and turn off the engine brake. Also, the reset stop 182 could be a variable. It can be actuated to vary the gap 185 to get different reset valve lift profiles. It can also sit on a spring. The spring force is large enough to reset the ball-locking device, but small enough to avoid hard clash to cause any engine damage due to improper design.

[0084] FIGS. 17A and 17B are schematic diagrams of another embodiment of the present invention with the engine brake actuation means 100 integrated into the valve bridge 400. The engine brake actuation means 100 is a ball-locking device similar to that shown in FIGS. 11A and 11B. A plurality of balls 175 are restrained by three surfaces on three different elements of the engine brake actuation means 100. The first surface is the tapered surface 192 on the braking piston 160 that is slidably disposed in a large bore 190 in the valve bridge 400. The second surface is the bottom flat surface of the bore 190, and the third surface is on the ball-locking piston or plunger 165 that is slidably disposed in a small bore 450 in the valve bridge. The engine brake reset means 150 includes the ball-locking piston 165 and a reset stop 182 on the engine cylinder head 500.

[0085] The engine braking operation including the resetting mechanism of this embodiment is similar to the embodiment shown in FIGS. 16A and 16B and not described here for simplicity.

[0086] FIGS. 18A and 18B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to another embodiment of the present invention. The exhaust valve lifter 200 includes a cam 230, a cam follower 235, a push rod or tube 201, and the rocker arm 210. Usually, there is a valve lash adjusting means either on the push rod side or on the valve bridge side. Here, a lash adjusting screw 110 is in contact with the push rod 201 and secured on the rocker arm 210 by a lock nut 105. The exhaust cam 230 contains an enlarged cam lobe 220 above the inner base circle 225 mainly for the normal engine operation. The enlarged cam lobe 220 is larger than a regular or normal exhaust cam lobe because a small cam lobe 233 is added for the engine braking operation. Another small cam lobe 232 could be added for braking gas recirculation to enhance engine braking performance. The rocker arm 210 can pivot on the rocker shaft 205. The other end of the rocker arm 210 is connected to an elephant foot 114 through a connector 113.

[0087] The engine brake actuation means 100 contains two braking pistons (also known as actuation pistons or hydraulic pistons) 160a and 160b (or simply 160) slidably disposed in bores 190a and 190b (or simply 190) in the valve bridge 400 between the inoperative position and the operative position. A supporting means 250 is designed to prevent the exhaust valve train from having

no-follow. The supporting means 250 contains an engine valve spring, for example, the outer spring 312a or 312b (or simply 312), and a spring seat 122a or 122b (or simply 122) that is biased against the valve spring retainer 302a or 302b (or simply 302) by the outer spring 312. The valve bridge 400 is supported by the two spring seats 122, not by the two exhaust valves 300 as usual (or prior art). Therefore, a gap 234 can be formed between the valve bridge 400 and the exhaust valves 300 to skip the motion from the lower portion of the cam 230, including the small cam lobes 232 and 233, during the normal engine operation.

[0088] Instead of being heavily loaded by a special "external" spring designed for preventing no-follow as disclosed by U.S. Patent Pub. No. 2005/0211206, the two braking pistons 160 shown in FIGS. 18A and 18B are not subjected to any load from the springs 312. Instead, springs 177a and 177b (or simply 177) are dedicated to the engine braking operation, not affected by the moment of inertia or no-follow of the valve train. The braking spring 177a or 177b is so designed that when the control fluid from the flow control valve 50 is at or below a first level pressure (as low as the ambient pressure), the braking piston 160 will not move from the inoperative position to the operative position; but when the control fluid is at or above a second level pressure (the engine lube oil pressure), the braking piston 160 will move from the inoperative position to the operative position. At the normal engine operation, the braking pistons 160 are biased to the inoperative position as shown in FIG. 18A.

[0089] With the supporting means 250, the braking springs 177 are less critical. Actually, by controlling the opening pressure of the check valve 172b and the first level pressure of the control fluid, the braking springs 177 may not be needed at all. To the other extreme, the outer engine valve springs 312 may not be needed for supporting the valve bridge 400, then the braking springs 177 may be used for controlling both the no-follow and the engine braking operation. In such a case, the spring seats 122 won't be necessary, but the braking springs 177 need to be stronger and the second level pressure of the control fluid to be higher to actuate the engine brake.

[0090] When engine braking is needed, the flow control valve 50 is turned on (FIG. 5B) to allow the control fluid with the second level or higher pressure to flow to the braking pistons 160 as shown in FIGS. 18A and 18B through the braking fluid circuit. The braking fluid circuit includes a flow passage 211 and a radial orifice 212 in the rocker shaft 205, a groove or cut 213 and a flow passage 214 in the rocker arm 210, a flow passage 115 in the connector 113 to the elephant foot 114, and flow passages 410 and 412 in the valve bridge 400. A check valve means 172b is disposed before the braking pistons 160 in the flow passage 410. Oil pressure at the second level or higher overcomes the load of the braking springs 177 and pushes the braking pistons 160 down and out of the bores 190 to the exhaust valves 300. The braking pistons 160 are at the operative position now and their motion or

stroke is limited by the exhaust valves 300 and equal to the gap 234 as shown in FIG. 18B. The control fluid displaced the braking pistons 160 takes up the gap 234 and forms a hydraulic linkage between the valve bridge 400 and the exhaust valves 300. As the cam 230 rotates, the motion from the whole cam including the small braking cam lobes 232 and 233 is transmitted to the exhaust valves 300 through the hydraulic linkage, since the braking pistons 160 are hydraulically locked to the operative position by the check valve means 172b and the resetting piston 165 whose function will be explained later.

[0091] The engine brake actuation means 100 also includes a safety valve 172s installed in the valve bridge 400 and hydraulically connected to the bore 412. It is a pressure relief type check valve and designed to be open only when the fluid pressure over the braking pistons 160 is above a predetermined value so that the related system components will not be overloaded. The predetermined value mainly depends on the load limit of the exhaust valve train and the engine brake actuation means 100.

[0092] The engine brake resetting means 150 is designed to modify the valve lift profile produced by the enlarged exhaust cam lobe 220. It includes a drain orifice 450 in the valve bridge 400 and the resetting piston 165 slidably disposed in the valve bridge 400 between a draining position and a feeding position. In the draining position (FIG. 18A), the resetting piston 165 opens the drain orifice 450, blocks the flow passage 410 and drain out the control fluid to let the two braking pistons 160a and 160b move from the operative position to the inoperative position. In the feeding position (FIG. 18B), the resetting piston 165 closes the drain orifice 450, opens the flow passage 410 and allows the control fluid with the second level or higher pressure to flow to the two braking pistons 160 and move them from the inoperative position to the operative position.

[0093] The resetting means 150 also includes a resetting spring 177r and a resetting piston stop 182. The resetting spring 177r is mounted on the valve bridge 400 by a crew 179 and biases the resetting piston 165 up to the draining position during the normal engine operation as shown in FIG. 18A. The resetting piston stop 182 is on the engine and below the resetting piston 165 with a resetting gap 185. As the cam 230 rotates and the resetting piston 165 moves down with the valve bridge 400 toward the resetting piston stop 182, the resetting gap 185 gets smaller. The resetting gap 185 is so designed that when the resetting piston 165 is at the draining position (FIG. 18A), it will not contact the resetting piston stop 182 during the whole cam rotation or cycle. During the engine braking operation, the control fluid overcomes the preload of the resetting spring 177r and pushes the resetting piston 165 from the draining position downward to the feeding position (FIG. 18B). As the enlarged exhaust cam lobe 220 moves the valve bridge 400 and the two exhaust valves 300 down to approach their maximum or peak lift, the resetting piston stop 182 will stop the resetting piston 165 from going downward with the valve

bridge 400, which changes the resetting piston 165 from the feeding position to the draining position. The control fluid drains out of the opened drain orifice 450 and the hydraulic linkage between the valve bridge 400 and the two exhaust valves 300 is temporarily lost. The two braking pistons 160 move upward from the operative position to the inoperative position. The lift of the two exhaust valves 300 is reset from the lift profile generated by the enlarged exhaust cam lobe 220 to a predetermined smaller lift profile, for example, the valve lift profile that would be generated by a regular exhaust cam for engines without an engine brake system. The resetting gap 185 can be varied by using an adjustable resetting piston stop 182 to meet the predetermined smaller valve lift profile.

[0094] Once the cam rotation passes the peak lift of the enlarged exhaust cam lobe 220, the valve bridge 400 will move upward and the resetting piston 165 in the valve bridge 400 will change from the draining position back to the feeding position. Control fluid with the second level pressure can flow to the braking piston 160a and 160b again and move them from the inoperative position back to the operative position to form the hydraulic linkage between the valve bridge 400 and the two exhaust valves 300. Therefore, the motion from the lower portion of the cam 230 including the small cam lobes 232 and 233 will be always transmitted to the exhaust valves. Only the motion from the higher portion of the cam 230 will be truncated by the resetting means 150.

[0095] FIG. 19 is a schematic diagram of an engine braking apparatus at the "On" position according to a second embodiment of the present invention. The only difference between this embodiment and the first one is the supporting means 250. In this embodiment, the supporting means 250 is located below the valve bridge 400 and between the two exhaust valves 300. The engine valve springs (the outer valve springs) are not utilized here. Instead, a dedicated supporting spring 312 is used to bias the spring seat 122 to a spring retainer 302r fixed on the resetting piston stop 182 that is also acting as a guide to the sliding of the spring seat 122. The resetting piston stop 182 can be a screw and the spring retainer 302r a lock nut so that the position of the spring seat 122 can be adjusted. There is a hole, or cut, 124 in the spring seat 122 to eliminate any hydraulic lock. The valve bridge 400 is supported by the spring seat 122. The gap 234 between the valve bridge 400 and the two valves 300 is formed, and the two braking pistons 160 are not subjected to any load from the supporting spring 312. The working mechanism and operation of this embodiment are the same as the first embodiment shown in FIGS. 18A and 18B and not explained here for simplicity.

[0096] FIGS. 20A and 20B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to yet another embodiment of the present invention. The engine is an overhead cam engine. Therefore, the valve lash adjusting system is moved to the valve bridge side. There is a flow passage 115 in the valve lash adjusting screw 110. The elephant foot 114 is attached

to the lower part 112 of the screw, while the upper part of the screw 110 is tightened to the rocker arm 210 by the lock nut 105. Note that the embodiment presented here can also be applied to a push rod/tube engine.

[0097] The two braking pistons 160a and 160b (or simply 160) with details in FIG. 20C are slidably disposed in a bore 415 in the valve bridge 400. The braking pistons 160 contain a first surface 140 commensurate with the operative position and a second surface 145 commensurate with the inoperative position. The two surfaces are on two flat cuts on the braking pistons 160 and have a height difference 130 (FIG. 20C). The braking pistons 160 are biased to the inoperative position by the braking springs 177a and 177b (or simply 177) to form a gap 234 between the valve bridge 400 and the exhaust valves 300 as shown in FIG. 20A. The gap 234 is equaled to or slightly larger than the height difference 130. One end of the braking spring 177a sits on a spring seat 176 that is mounted on the braking piston 160a. The other end of the spring 177a sits on another spring seat 178 disposed in a bore 183. The spring seat 178 shown in FIG. 20C is stopped at the end of the bore 183. But when the braking pistons 160 are assembled in the valve bridge 400, the spring seat 178 is normally stopped by pins 142a and 142b (or simply 142) fixed in the valve bridge 400 as shown in FIGS. 20A and 20B. There is a slot 137 or axial cut across the bore 183 (FIG. 20C) in the braking pistons 160, which has a width slightly larger than the pins 142. The pin 142a and the slot 137 with the two end surfaces 137a and 137b form a motion limiting means to control the movement of the braking piston 160a between the inoperative position and the operative position. They also form an anti-rotation means to the braking pistons 160 so that the first and second surface 140 and 145 always face downward.

[0098] When engine braking is needed, the flow control valve 50 is turned on (FIG. 5B) to allow the control fluid with the second level or higher pressure to flow through the engine braking fluid circuit and into a pressure chamber 425 in the valve bridge 400 (FIG. 20A). The fluid pressure overcomes the preload of the braking springs 177 and pushes the braking pistons 160 toward the exhaust valves 300. When the end surface 137b of the slot 137 (FIG. 20C) hits the pin 142a (FIG. 20B), the braking piston 160a is moved from the inoperative position the operative position with the operative surface 140 (FIG. 20C) over the valve 300a (FIG. 20B). The gap 234 between the valve bridge 400 and the two valves 300 (FIG. 20A) is taken up (eliminated or greatly reduced) and a linkage is formed (FIG. 20B). As the cam 230 rotates, the motion from the whole cam 230 is transmitted to the exhaust valves 300 without resetting.

[0099] When engine braking is not needed, the flow control valve 50 is turned off as shown in FIG. 5A and the control fluid will drain out of the braking fluid circuit. The pressure in the chamber 425 in the valve bridge 400 (FIG. 20A) will drop from the second level or higher to the first level or lower. The braking pistons 160 will be

pushed back into the valve bridge 400 by the braking springs 177. Once the inoperative surface 145 (FIG. 20C) is over the exhaust valves 300 as shown in FIG. 20A, the braking pistons 160 are at the inoperative position and the gap 234 is formed to skip the motion from the lower portion of the cam 230, including the small braking cam lobes 232 and 233 for the normal engine operation.

[0100] FIG. 20D shows a different braking piston assembly that can be used for the embodiment shown in FIGS. 20A and 20B. The two braking pistons 160a and 160b are biased toward each other by the braking springs 177a and 177b that are held and guided by a screw 148 and a screw nut 149. The inoperative surface 145 and the operative surface 140 are cylindrical. When assembled in the valve bridge 400, the two braking pistons 160a and 160b are separated by a clip ring 159 (FIG. 20D) that can be mounted in a groove (not shown) at the center of the bore 415 (FIGS. 20A and 20B).

[0101] FIGS. 21A and 21B are schematic diagrams of another embodiment similar to the embodiment shown in FIGS. 20A and 20B. However, there is only one braking piston 160 integrated into the valve bridge 400. Also, there is a dedicated braking valve lifter 200b, which includes a dedicated braking cam 230b and rocker arm 210b, for opening one exhaust valve 300a for engine braking.

[0102] FIGS. 22A and 22B are schematic diagrams of a new embodiment with a hydraulic braking piston 160 integrated into the valve bridge 400. A check ball 170 is used to form a hydraulic lock when engine braking is needed. A bleeding orifice 197 is put into the braking piston 160 for bleeding off the trapped engine oil when the engine braking is turned off.

[0103] FIG. 23 is a schematic diagram of another embodiment of the present invention. The engine brake actuation means 100 includes a dedicated valve lifter 200b and a hydraulic system integrated in the exhaust valve train. The hydraulic system includes a piston-sliding device with a braking piston 160 slidably disposed in the valve bridge 400 between an inoperative position and an operative position. The braking piston 160 contains an operative surface 140 commensurate with the operative position for the engine braking operation. The inoperative surface 145 commensurate with the inoperative position for the normal engine operation is on the valve bridge 400 and separated from the elephant foot 114b by a gap 234. The gap 234 is equal to or slightly larger than the height difference 130 between the two surfaces 140 and 145. The braking piston 160 is biased to the inoperative position by a spring 177a. One end of the spring 177a is on the braking piston 160 and the other end on a spring seat 178b that is secured on the valve bridge 400 by at least one screw 179. Seat 178b is also used as a stop to the braking piston 160, which limits the travel of the braking piston 160.

[0104] The dedicated braking valve lifter 200b includes a dedicated cam 230b, a cam follower 235b, a rocker arm 210b, and a lash adjusting system containing the

adjusting screw 110b, the lock nut 105b, and the elephant foot 114b. The braking cam 230b only has the small cam lobes 232 and 233 above the IBC 225b for the engine braking operation, while the standard exhaust cam 230r has only the regular exhaust lobe 220r above the IBC 225 for the normal engine operation. Only one exhaust valve 300a is used for engine braking. The engine braking valve train is formed by the dedicated braking valve lifter 200b and the exhaust valve 300a.

[0105] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to allow engine oil to flow through the engine braking fluid circuit and into a pressure chamber 425 in the valve bridge 400 as shown in FIG. 23. The engine oil pressure overcomes the preload of the spring 177a, and pushes the braking piston 160 out of the bore 415 in the valve bridge 400 from the retracted position to the extended position. The braking piston 160 is stopped at the spring seat 178b, and the operative surface 140 on the braking piston 160 is under the elephant foot 114b. Now the braking piston 160 is fully extended to the operative position and the gap 234 in the engine braking valve train is taken up to form a mechanical linkage. All the cam motion, from the dedicated braking cam 230b and the standard exhaust cam 230r, is transmitted to the exhaust valves 300a and 300b. There is no hydraulic compliance from hydraulic linkage as used by prior art engine braking systems.

[0106] When engine braking is not needed, the engine brake control means 50 is turned off (FIG. 5B) and there will be little or no oil supplied to the engine braking fluid circuit. The oil pressure in the chamber 425 is not high enough and the braking piston 160 will be pushed back into the valve bridge 400 by the spring 177a. The braking rocker arm 210b is biased against the braking cam 230b and away from the inoperative surface 145 by a spring 198b. The gap 234 in the valve train as shown in FIG. 23 is formed. Now the braking piston is retracted and disengaged from the dedicated braking valve lifter 200b. Part of the cam motion, i.e., from the braking cam lobes 232 and 233 is skipped. Only the motion from the standard exhaust cam 230r is transmitted to the exhaust valves 300 for the normal engine operation.

[0107] Note that the bleeding orifice 418 in the valve bridge 400 is optional and used as a flow draining means for turning off the engine brake faster or eliminating the need of the drain port 222 in FIGS. 5A and 5B so that a two-way solenoid valve may be used to replace the three-way solenoid valve 51. Spring 198 may be desirable, for example, at the top surface 215 of the rocker arm 210, to bias the rocker arm 210 against the valve bridge 400 for a better sealing of the fluid from the passage 214 in the rocker arm to the passage 410 in the valve bridge 400.

[0108] FIGS. 24A and 24B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions. It is a similar embodiment to that shown in FIG. 23 except that the braking actuation means 100 integrated in the valve bridge 400 is a toggle system. In addition to the braking piston 160, there is an actuation piston 164.

A linkage bar or toggle 184 is placed between the two pistons.

[0109] When engine braking is needed, engine oil will flow into a pressure chamber 425 in the valve bridge 400 as shown in FIG. 24A. The engine oil pressure overcomes the preload of the spring 156 (for details of the braking actuation means 100, please refer to FIGS. 24C and 24D) and pushes the actuation piston 164 out of the bore 166 in the valve bridge 400, which makes the toggle 184 from the slant position to the straight position. The braking piston 160 is moved up from the inoperative position to the operative position, as shown in FIG. 24B. The gap 234 between the lash adjusting screw 110b and the braking piston 160 is taken up to form a mechanical linkage. All the braking cam motion from the dedicated braking cam 230b is transmitted to the exhaust valve 300a for the engine braking operation.

[0110] When engine braking is not needed, there will be little or no oil supplied to the engine braking fluid circuit. The oil pressure in the chamber 425 is not high enough and the actuation piston 164 will be pushed back into the valve bridge 400 by the spring 156. The toggle 184 will move from the straight position back to the slant position and the braking piston 160 be pushed lower into the valve bridge 400 by the spring 177. The gap 234 as shown in FIG. 24A is formed and the braking cam motion from the braking cam lobes 232 and 233 is skipped.

[0111] FIGS. 25A and 25B are schematic diagrams of an engine braking actuation means 100 with a different toggle system at the "Off" and "On" positions. It contains two linkage bars or toggles 184 and 186, and a braking piston 160 that slides in a vertical bore 190 in the brake housing 210. The upper pin 184 has two spherical ends: one engaged with a spherical socket 122 in the adjusting screw 110, and the other with another spherical socket 125 in the lower pin 186 whose lower end sits in a third spherical socket 128 in the braking piston 160. FIG. 25A shows the retracted position of the toggle device where the two pins guided in the slot 137 that is cut through a guiding piston 162 are pushed to the left by the spring 156. The guiding piston 162 slides in a horizontal bore 260 in the braking housing 210. There is an actuation piston 164 that slides in the guiding piston 162. The slot 137 in the guiding piston 162 has a width that is about the same as or slightly larger than the diameter of the two pins and a length that is smaller than the diameter of the bore 190. There will be always contact (no separation) among the braking piston, the lower pin, the upper pin and the adjusting screw due to the upward force of the spring 177 that is secured to the brake housing 210 with at least one screw 179.

[0112] When engine brake is needed, the engine braking control means is turned on and oil pressure can push both pistons 162 and 164 to the right against the preloads of the spring 156 and 177. Note that the actuation piston 164 can be moved to the right further to lock the two pins or toggles 184 and 186 in a straight position, aligned with the adjusting screw 110b and the braking piston 160, as

shown in FIG. 25B. Now the toggle device is locked to its extended position on the operative position. The motion limiting means for this toggle device is unique. The angle between the two pins or toggles 184 and 186 decides the height difference 130, while the angle itself is controlled by the two pistons 162 and 164. The bleeding orifice 168 in the guiding piston 162 is designed to eliminate hydraulic lock.

[0113] The lash adjusting system is incorporated into the engine braking actuation means 100. The braking valve lash adjusting screw 110b is fastened to the braking housing 210 by a locking screw nut 105b. The two linkage pins or toggles 184 and 186 and the braking piston 160 can move up and down by the adjusting screw 110b. If needed, chamfers 163 and 243 can be provided to allow the lash adjusting movement of the system.

[0114] Just like the braking actuation means 100 shown in FIGS. 24C and 24D, the device shown in FIGS. 25A and 25B can also be integrated into different valve train components, which means that the braking housing 210 could be a rocker arm or a valve bridge.

[0115] FIG. 26 shows an embodiment of an engine braking apparatus with the braking actuation means 100 shown in FIGS. 25A and 25B integrated into a dedicated braking rocker arm 210b.

[0116] FIG. 27 shows a similar braking actuation means 100 to that shown in FIGS. 25A and 25B except that the actuation piston 164 is not inside the guiding piston 162. The guiding piston 162 has a "U" type cut 137 that guides the two toggles 184 and 186. The guiding piston 162 also has a limited stroke so that a braking valve lash adjustment is possible.

[0117] FIG. 28 shows another modification of the braking actuation means 100 shown in FIGS.

25A and 25B. Here, only one horizontal piston 162 is used, which acts as an actuation piston and a guiding piston. Also, two identical toggles 186 are used and a ball 189 is added between the two toggles. A thin washer plate 188 holds the ball 189 in the washer's middle hole. The washer plate 188 is retained or guided in three slots, two slots (185) in the two toggles 186 and one (187) in the piston 162. Under such arrangement, the washer plate 188 can move up and down for valve lash adjustment, and the two toggles 186 can rotate when the actuation piston 162 is moved horizontally by oil pressure or spring force.

[0118] FIG. 29 shows a similar embodiment to that shown in FIGS. 21A and 21B except that the braking piston 160 is integrated into the rocker arm 210 so that both of the two exhaust valves 300a and 300b will be opened during the engine braking operation.

[0119] The braking piston 160 contains a first surface 140 commensurate with the operative position and a second surface 145 commensurate with the inoperative position. The two surfaces are two flat cuts on the braking piston 160 and have a height difference 130. The braking piston 160 is biased into the bore 216 in the rocker arm 210 to the inoperative position by the braking spring 177a.

One end of the braking spring 177a sits on a spring seat 176 mounted on the braking piston 160. The other end of the spring 177a sits on another spring seat 178b slidable disposed in a bore 183 in the braking piston 160. The spring seat 178b is normally stopped by a pin 142 fixed in the rocker arm 210. There is a slot or axial cut 137 across the bore 183 in the braking piston 160, which has a width slightly larger than the pin 142. The pin 142 and the slot 137 form a motion limiting means to control the movement of the braking piston 160 between the inoperative position and the operative position. They also form an anti-rotation means to guide the braking piston 160 so that the first and second surfaces 140 and 145 always face upward to the elephant foot 114b.

[0120] FIG. 30 is a schematic diagram of an engine braking apparatus at its "On" position according to a variation from the embodiment shown in FIG. 29. There are two major changes. First, the dedicated braking valve lifter 200b in FIG. 29 is replaced by an engine brake housing 125 fixed on the engine. Second, the cam 230 containing the enlarged exhaust cam lobe 220 as well as the small braking cam lobes 232 and 233 is replaced by the regular cam 230r containing only the regular exhaust cam lobe 220r. Therefore, the embodiment shown in FIG. 30 is for BTEB, while the one in FIG. 29 is for CREB.

[0121] When engine braking is needed, the engine brake control means 50 is turned on (FIG. 5A) to allow engine oil to flow through the engine braking fluid circuit and into the bore 216 in the rocker arm 210. As the cam 230r pushes the rocker arm 210 rotating anticlockwise to open the exhaust valves 300, the braking piston 160 will move down with the rocker arm 210 and away from the lash adjusting screw 110b. The engine oil pressure overcomes the preload of the spring 177a and pushes the braking piston 160 out of the bore 216 from the inoperative position to the operative position as shown in FIG. 30. The braking piston 160 is stopped at the pin 142 fixed in the rocker arm 210, and the operative surface 140 on the braking piston 160 is under the adjusting screw 110b. As the cam 230r continues its rotation and passes the peak of the cam lobe 220r, the rocker arm will rotate clockwise and the braking piston 160 will move up towards the adjusting screw 110b. Due to the height difference 130 between the operative surface 140 and the inoperative surface 145, the exhaust valves 300 could not return to their seats 320 but are held open for the BTEB. The braking valve opening is 330 and about 0.4 to 2.0mm, much smaller than the normal exhaust valve opening (>10mm). Corresponding to the braking valve opening 330, there is a gap 234 between the cam 230r and cam follower 235 since the rocker arm 210 is also stopped by the lash adjusting screw 110b through the braking piston 160 and cannot fully return to its regular top position. Therefore, the engine braking load is not passed to the exhaust valve train, e.g., rocker arm 210 and cam 230r, but to the housing 125 mounted on the engine.

[0122] When engine braking is not needed, the engine

brake control means 50 is turned off (FIG. 5B) and there will be little or no oil supplied to the engine braking fluid circuit. The oil pressure in the bore 216 is not high enough to overcome the force by spring 177a and the braking piston 160 will be pushed back into the bore 216 to the inoperative position. The inoperative surface 145 now is under the valve lash adjusting screw 110b with a regular exhaust valve lash between them, and the braking piston 160 will not contact the lash adjusting screw 110b during the whole cam rotation. The exhaust valves 300 will return to their seats 320 and there will be no gap 234 between the cam and cam follower. Now the actuation means 100 is at the inoperative position and disengaged from the normal engine operation.

[0123] FIG. 31 is a schematic diagram of an engine braking apparatus at the "Off" position according to a variation from the embodiment shown in FIG. 29, where the braking actuation means 100 is moved into a dedicated braking rocker arm 200b. Also, only one valve 300a is opened for engine braking.

[0124] FIG. 32 is a schematic diagram of an engine braking apparatus at the "Off" position according to a variation from the embodiment shown in FIG. 23. Instead of using a dedicated braking valve lifter 200b, the braking valve lifter of the engine brake actuation means 100 is integrated into the exhaust valve lifter 200. The braking cam 230b and the regular cam 230r in FIG. 23 are combined into a new cam 230 shown in FIG. 32. The new cam 230 contains the small braking cam lobes 232 and 233 as well as an enlarged exhaust cam lobe 220. The lower portion of the enlarged exhaust cam lobe 220 has about the same height as the small cam lobes 232 and 233, while the higher portion is about the same as the regular exhaust cam lobe 220r. A spring 198e is put between the lash adjusting screw 110 and the lash adjusting piston 112 to prevent no-follow of the exhaust valve train components. A different type of spring, for example, a flat spring or a torsion spring, can be used and be put at different location as long as the same purposes can be achieved. A gap 234 is designed between the lash adjusting screw 110 and the lash adjusting piston 112 so that part of the motion from the cam 230 including the small braking cam lobes 232 and 233 is skipped during the normal engine operation.

[0125] The engine braking operation of this embodiment is similar to the embodiment shown in FIG. 23 and only the difference is described here. The exhaust valve (the braking valve) 300a is opened earlier by the lower portion of the enlarged cam lobe 220 through the braking elephant foot 114b, while the other (the non-braking valve) 300b opened later by the higher portion of the enlarged cam lobe 220 through the regular elephant foot 114 due to the gap 234. By the same token, the braking valve 300a will be closed later than the non-braking valve 300b. Therefore, there will be a small tilt of the valve bride 400, which will create an unbalanced loading condition when the regular elephant foot 114 acts on the valve bridge 400 opening both exhaust valves 300. The uni-

versal pad 430 is provided between the valve bridge 400 and the valves 300 to better handle the unbalanced load on the exhaust valves 300. Also, the braking load is passed to the exhaust valve lifter 200.

[0126] The engine braking apparatus shown in FIG. 32 can be easily converted from the compression release type engine braking to the bleeder type engine braking. First, replace the cam 230 with the regular cam 230r shown in FIG. 30. Second, eliminate the gap 234 between the lash adjusting screw 110 and the lash adjusting piston 112.

[0127] FIG. 33 is a schematic diagram of an engine braking apparatus at the "Off" position using the braking actuation means 100 of a toggle system shown in FIGS. 24C and 24D.

[0128] FIG. 34 is a schematic diagram of an engine braking apparatus at the "Off" position according to yet another embodiment of the present invention. The braking piston 160 slides in a bore 166 in the braking housing 125 (could be a rocker arm or a valve bridge). There is a wedge or a slant surface 140 on the braking piston 160. The gap 130 between the slant surface 140 and the brake actuation linkage 210 is controlled through the adjusting screw 175 that is secured on the braking housing 125 by a lock nut 173. When engine braking is needed, oil is supplied to the port 214 and the oil pressure overcomes the preload of spring 177. The braking piston 160 is pushed to the left and its end surface 163 is stopped against the adjusting screw 175. The stroke 136 and the slope of the slant surface 140 of the braking piston 160 decide the final gap 130. The braking piston 160 is guided through a pin 142 fixed in the braking housing 125 and a slot 137 cut axially in the braking piston 160.

[0129] FIG. 35 is a schematic diagram of another engine braking apparatus with a wedge system at the "Off" position. The only difference from FIG. 34 is that the slant surface 140 on the braking piston 160 is perpendicular to the brake actuation linkage 210.

[0130] FIG. 36 is a schematic diagram of an engine braking apparatus at the "Off" position according to yet another embodiment of the present invention. The braking piston 160 slides in a bore 166 in a braking housing, here the valve bridge 400. There is a wedge or a slant surface 140 on the braking piston 160. The gap 130 between the slant surface 140 and the brake actuation linkage 210 is controlled through the adjusting screw 175 that is secured on the valve bridge 400 by a lock nut 173. When engine braking is needed, oil is supplied to the port 214 and the oil pressure overcomes the preload of spring 177. The braking piston 160 is pushed to the left with a stroke 136. The stroke 136 and the slope of the slant surface 140 of the braking piston 160 decide the final gap 130. The braking piston 160 is guided through a pin 142 fixed in valve bridge 400 and a slot 137 cut axially in the braking piston 160.

[0131] FIGS. 37A and 37B are schematic diagrams of an engine braking apparatus at the "Off" and "On" positions according to another embodiment of the present

invention. The regular exhaust cam 230r is used. Therefore, this is a bleeder type engine brake opening one exhaust valve for engine braking. A ball-locking device similar to that shown in FIGS. 13A and 13B is disposed slidably in the valve bridge 400 and below the braking elephant foot 114b.

[0132] When engine braking is needed, the control means 50 is turned on (FIG. 5A) to supply engine oil to the engine brake actuation means 100 through the engine brake fluid circuit. Oil pressure overcomes the force of spring 177a and pushes upwards the braking piston 160 as well as the ball-locking piston 165. As the cam 230 rotates, the braking piston 160 will move down with the valve bridge 400 and further away from the braking elephant foot 114b. Before the cam rotation reaches the peak lift of the cam lobe 220r, the braking piston 160 will be fully extended out of the bore 190 and to the clip ring 176. During the upward motion of the braking piston 160, the balls 175 contained in the braking piston 160 will align with and move into the annular groove 170 in the valve bridge 400. Once the balls 175 are in the groove 170, the ball-locking piston 165 will move up because the oil pressure overcomes the force of spring 177. The balls 175 are locked into the groove 170 by the larger outer diameter surface of the ball-locking piston 165 to form a mechanical linkage between the braking piston 160 and the valve bridge 400 (FIG. 37B). The braking piston 160 is now at the extended or operative position with a stroke 195 that is larger than the initial valve lash 132 (FIG. 37A). After the cam rotation passes the peak lift of the cam lobe 220r, the braking piston 160 will move up with the valve bridge 400 as well as the exhaust valves 300. However, the braking exhaust valve 300a cannot return to its seat 320 but is held open due to the mechanical linkage (FIG. 37B). The braking valve opening 330 is equal to the difference between the braking piston stroke 195 and the initial valve lash 132 (FIG. 37A).

[0133] When engine braking is not needed, the control means 50 is turned off (FIG. 5B) and there will be little or no oil pressure acting on the ball-locking piston 165, which will slide down in the braking piston 160 under the force of spring 177. The balls 175 will move inwards and out of the annular groove 170, and the braking piston 160 will move down under the force of spring 177a. Now the ball-locking device is at the retracted or inoperative position as shown in FIG. 37A and the engine braking actuation means is disengaged from the normal engine operation. The orifice or flow passage 196 in the ball-locking piston 165 is optional, and could be used to turn off the engine brake.

[0134] FIG. 38 is a schematic diagram of an engine braking apparatus at the "On" position according to an embodiment that combines some of the features shown in FIG. 32 and FIGS. 37A and 37B. The same braking cam 230 as shown in FIG. 32 is used, which contains the small braking cam lobes 232 and 233 as well as the enlarged exhaust cam lobe 220. The same ball-locking device as shown in FIGS. 37A and 37B is used. The new

feature of this embodiment is from the reset means 150 that is incorporated into the actuation means 100. The lash adjusting piston 112 also acts as a reset piston to block the oil flow to the braking piston 160, and the orifices 196 and 197 in the ball-locking device serve as draining passage for the resetting.

[0135] During the engine braking operation, oil pressure overcomes the force of spring 177a and pushes the ball-locking device to the operative position to form a mechanical linkage (FIG. 38). The braking valve lash 132 between the braking piston 160 and the elephant foot 114b is slightly larger than the regular exhaust valve lash. As the cam 230 rotates, the small braking cam lobes 232 and 233 push the braking valve 300a open due to the mechanical linkage. The non-braking valve 300b is still closed due to the gap 234 between the lash adjusting screw 110 and the lash adjusting piston 112. The lower portion of the enlarged cam lobe 220 will also open the braking valve 300a but not the non-braking valve 300b. But the higher portion of the enlarged cam 220 will act on the valve bridge 400 to open both of the two exhaust valves 300 because the gap 234 is taken up by the lower portion of the enlarged cam lobe 220. Therefore, the braking valve 300a opens earlier and closes later than the non-braking valve 300b. There will be a small tilt of the valve bridge 400, which will create an unbalanced loading on the two exhaust valves 300.

[0136] The reset means 150 is designed here to address the unbalanced loading issue. When the lash adjusting screw 110 touches the shoulder of the lash adjusting piston 112, the gap 234 is eliminated and the flow passage 113 in the lash adjusting screw 110 is blocked. Oil under the braking piston 160 will bleed out of the orifices 196 and 197 under the load of spring 177a. The braking piston 160 will retract into the bore 190 and separate from the elephant foot 114b. The braking valve 300a will return to its seat 320 with the same closing timing as the non-braking valve 300b. If the braking piston 160 were still extended without the resetting, the braking elephant foot 114b would act on it and the braking valve 300a would close much later than the non-braking valve 300b. When the rocker arm 210 continues its anti-clockwise rotation after the valves 300 are seated, the gap 234 is re-formed and the flow passage 113 is unblocked so that oil can refill the ball-locking device. The braking piston 160 will be fully extended during the cam IBC 225 in front of the small braking cam lobes 232 and 233 so that their motion can be transmitted to the braking valve 300a, and the engine braking cycle repeats. Therefore, the reset means 150 will modify the valve lift profile produced by the enlarged cam lobe 220, not that by the small braking cam lobes 232 and 233.

[0137] FIG. 39 shows a different version of the embodiment in FIG. 38 with a different reset means 150. A reset piston 166 is slidably disposed in the valve bridge 400 below the elephant foot 114. The reset piston 166 as well as the rocker arm 210 is biased to the valve bridge 400 by a spring 198 to prevent no-follow of any exhaust valve

train components. A reset flow passage 167 is also added in the valve bridge 400, and there is no more need for a bleeding orifice in the ball-locking piston 165.

[0138] When engine braking is needed, the control means 50 is turned on (FIG. 5A) to allow engine oil to flow to the reset piston 166 and the ball-locking device through the brake fluid circuit that further includes the flow passage 197r in the reset piston 166. Oil pressure overcomes the loads of spring 198 and spring 177a and pushes the reset piston 166 and the braking piston 160 upwards to rotate the rocker arm 210 anti-clockwise towards the cam 230. The braking system is now at the "On" or operative position as shown in FIG. 39. The braking piston 160 is stopped at the clip ring 176 with a stroke of 195 that takes up the lash or gap between the elephant foot 114b and the braking piston 160. The reset piston has a stroke of 234r corresponding to the gap 234 that would show up between the cam follower 235 and the cam 230 if the braking system were at the "Off" position. As the cam 230 rotates, the motion from the small braking cam lobes 232 and 233 is transmitted to the exhaust valve 300a through the braking piston 160, the valve bridge 400, and the universal pad 430 for the engine braking operation, since the braking piston 160 is extended and mechanically locked to the operative position by the ball-locking piston 165. The motion from the small braking cam lobes 232 and 233 is not transmitted to the other exhaust valve 300b because of the gap 234r between the reset piston 166 and the valve bridge 400. The oil under the reset piston 166 is pushed back through the flow passage 197r. An accumulator may be needed in the braking fluid circuit to absorb the flow pumped back by the reset piston 166.

[0139] Once the cam rotation gets into the higher portion of the enlarged cam lobe 220, the reset piston 166 will touch the valve bridge 400 and act on both exhaust valves 300a and 300b. But before the reset piston 166 touches the valve bridge 400, it will open the reset flow passage 167 since the reset height 131 is smaller than the gap 234r. The oil under the braking piston 160 will drain out of the passage 167 and the braking piston 160 will retract into the bore 190 under the load of spring 177a. The opened braking exhaust valve 300a will return to its seat 320 and the tilted valve bridge 400 will be leveled. There will be no unbalanced load when the reset piston 166 acts on the valve bridge 400 and open both exhaust valves 300a and 300b by the higher portion of the enlarged cam lobe 220. Once the valves 300 are seated, the rocker arm 210 will continue to rotate anti-clockwise and the reset piston 166 will move up in the valve bridge 400 under oil pressure to block the reset flow passage 167 so that oil can refill and push out the ball-locking device. The ball-locking device will be fully extended to the operative position during the cam IBC 225 in front of the small braking cam lobes 232 and 233 so that their motion can be transmitted to the braking valve 300a, and the engine braking cycle repeats.

[0140] When engine braking is not needed, the control

means 50 is turned off (FIG. 5B) and there will be little or no oil supplied to the ball-locking device. When the reset piston 166 moves down and opens the reset flow passage 167, the oil under the ball-locking device will drain out and the braking piston 160 will retract into the bore 190 under the load of spring 177a. The reset piston 166 is biased to the valve bridge 400 by the spring 198 to form a gap 234 between the cam follower 235 and the cam 230 to skip part of the cam motion, i.e., from the lower portion of the cam 230 including the braking cam lobes 232 and 233. The two exhaust valves 300 will be opened by the higher portion of the enlarged cam lobe 220 through the rocker arm 210, the reset piston 166 and the valve bridge 400. The retracted braking piston 160 will not touch the elephant foot 114b of the braking valve lash adjusting means during the whole cycle of cam rotation. The engine brake actuation means 100 is now at the inoperative position and disengaged from the normal engine operation.

[0141] FIGS. 40A and 40B are schematic diagrams of an embodiment of an engine braking apparatus similar to that shown in FIG. 39 except that a hydraulic piston 160 is used. A check ball 170 is used for the hydraulic lock, while the reset mechanism is similar to that in FIG. 39.

CONCLUSION, RAMIFICATIONS, AND SCOPE

[0142] It is clear from the above description that the engine braking apparatus according to the embodiments of the present invention have one or more of the following advantages over the prior art engine braking systems:

- (a) The apparatus can be installed on all types of engines;
- (b) The apparatus has much faster response (on & off) time;
- (c) The apparatus transmits force, or the engine braking load, through mechanical linkage means that does not have high compliance and overloading problems associated with hydraulic means used by the prior art engine brakes;
- (d) The apparatus has no asymmetric loading on valves or valve bridge associated with some of the prior art engine brakes;
- (e) The apparatus has fewer components, reduced complexity, and lower cost;
- (f) The apparatus has a braking valve lash setting mechanism and thus reduced manufacturing tolerance requirements for the engine brake components;
- (g) The apparatus is simple in construction, more reliable in operation, and effective at all engine speeds; and
- (h) The apparatus does not affect normal engine performance.

[0143] While my above description contains many spe-

cificities, these should not be construed as limitations on the scope of the invention, but rather as an exemplification of the preferred embodiments thereof. Many other variations are possible. For example, the engine braking apparatus disclosed here can be applied to a push tube type engine instead of the overhead cam type engine. It can use one valve for engine braking instead of two valves.

[0144] Also, the spring 198 shown in FIG. 4A and other figures can sit at other locations or even between two engine valve train components, such as between the rocker arm 210 and the valve bridge 400, as far as the spring force is large enough to prevent valve train components from no-following during the normal engine operation and is small enough to allow the engine brake actuation means 100 to be actuated during the engine braking operation. The spring 198 can also take a different type than the coil spring, for example, a flat or leaf spring, a wavy spring, or a torsion spring.

[0145] Also, the engine brake actuation means 100 can be controlled (turned on and off) by other types of control means 50, such as a dedicated hydraulic system, a common rail system, and a pneumatic system. And a poppet type solenoid valve could be used to replace the spool type valve 51 of the control means 50 as shown in FIGS. 5A and 5B.

[0146] Also, the valve lift profile illustrated in FIG. 9 could be different. The BGR lift 232v, the CRB lift 233v, and the enlarged main valve lift 220v could be separated individual bumps or connected to each other. The braking valve event could be a compression release type engine brake with a CRB bump 233v around compression TDC plus a BGR bump 232v around intake BDC, or other types of engine braking, such as a partial cycle bleeder brake with a substantially constant valve lift throughout the compression stroke. There should be no valve lift during most of the intake stroke so that the engine brake actuation means 100 could be changed from the retracted position to the extended position. Accordingly, the small cam lobes 232 and 233 shown in FIG. 4A and other figures could be combined to form a single cam lobe with a substantially constant lift during the engine compression stroke for a partial cycle bleeder brake. The single cam lobe can even be extended to be connected to the enlarged cam lobe 220. Now the "single" cam lobe is in fact just a transition "step" to the enlarged cam lobe 220. In summary, the cam contains at least one small lobe and the at least one small lobe includes the constant lift type for a partial cycle bleeder brake.

[0147] Accordingly, the scope of the invention should be determined not by the embodiments illustrated, but by the appended claims and their legal equivalents.

Claims

1. Apparatus for converting an internal combustion engine from a normal engine operation to an engine

braking operation, the engine including an exhaust valve train comprising two exhaust valves, a valve bridge and an exhaust valve lifter having an exhaust cam with an exhaust cam lobe for cyclically opening and closing the two exhaust valves, said apparatus comprising:

- (a) an actuator comprising a hydraulic system integrated into said exhaust valve train, said hydraulic system comprising a braking piston slidably disposed in said valve bridge between an inoperative position and an operative position; in said inoperative position, said braking piston being retracted and disengaged from said normal engine operation, and in said operative position, said braking piston being extended and opening one of the two exhaust valves for said engine braking operation; and
 - (b) a controller for moving said braking piston between said inoperative position and said operative position.
2. The apparatus of claim 1, wherein said hydraulic system further comprises a flow control valve and a brake fluid circuit in said exhaust valve train.
 3. The apparatus of claim 1 or 2, wherein said actuator further comprises a dedicated valve lifter, said dedicated valve lifter comprising a braking cam, said braking cam having a braking cam lobe, and said braking cam lobe actuating said braking piston for opening the one of the two exhaust valves when said braking piston is in said operative position.
 4. The apparatus of claim 3, wherein said actuator further comprises a spring for biasing said dedicated valve lifter away from said braking piston.
 5. The apparatus of any one of claims 1 to 4, wherein said actuator further comprises a braking valve lifter integrated into said exhaust valve lifter, said braking valve lifter having a braking cam lobe integrated into said exhaust cam, said exhaust cam lobe of the exhaust cam being enlarged for accommodating the integration of said braking cam lobe, and said braking cam lobe actuating said braking piston for opening the braking exhaust valve when said braking piston is in said operative position.
 6. The apparatus of claim 5, further comprising an engine brake reset device for modifying the valve lift profile produced by said enlarged exhaust cam lobe during the engine braking operation, and said reset device comprising a reset piston and at least one reset flow passage.
 7. The apparatus of any one of claims 1 to 6, further comprising a valve lash adjusting device integrated

into said actuator for setting a lash or gap between said actuator and the one of the exhaust valves.

8. The apparatus of any one of claims 1 to 7, wherein said controller comprises an electro-hydro-mechanical system, said electro-hydro-mechanical system comprising a solenoid valve for supplying and cutting off a fluid flow to said braking piston; and said fluid flow controlling the motion of said braking piston between the inoperative position and the operative position.
9. The apparatus of any one of claims 1 to 8, wherein said controller further comprises a flow drain for draining the fluid flow from said braking piston, and said flow drain comprising a bleeding orifice or a flow draining passage.
10. A method of modifying engine valve lift in an internal combustion engine to produce an auxiliary engine valve event that is different from a normal engine operation, the engine having an engine valve train comprising two engine valves, a valve bridge and an engine valve lifter for cyclically opening and closing the two engine valves, said method comprising the steps of:
 - (a) providing an actuator comprising a hydraulic system integrated into said exhaust valve train, said hydraulic system comprising a hydraulic piston slidably disposed in said valve bridge between an inoperative position and an operative position, in said inoperative position, said hydraulic piston being retracted and disengaged from said normal engine operation, and in said operative position, said hydraulic piston being extended and opening one of the two engine valves for said auxiliary engine valve event;
 - (b) providing a controller for moving said hydraulic piston with a fluid flow;
 - (c) turning on said controller and supplying the fluid flow to said hydraulic piston;
 - (d) moving said hydraulic piston from said inoperative position to said operative position;
 - (e) acting on said hydraulic piston by said actuator; and
 - (f) opening one of the two engine valves for said auxiliary engine valve event.
11. The method of claim 10, wherein the actuator includes a dedicated valve lifter, said dedicated valve lifter comprising a dedicated cam, and said dedicated cam having a dedicated cam lobe; and wherein the step of acting on said hydraulic piston by said actuator includes actuating said hydraulic piston by the dedicated cam lobe of said dedicated valve lifter.
12. The method of claim 10 and 11, further comprising

the steps of:

- (a) providing a controller with a flow drain having a bleeding orifice or a flow draining passage;
- (b) turning off said controller and cutting off the fluid flow to said hydraulic piston;
- (c) opening said bleeding orifice or flow draining passage;
- (d) draining the fluid from said hydraulic piston through said flow drain;
- (e) moving said hydraulic piston to the inoperative position; and
- (f) disengaging said actuator from said normal engine operation and turning off said auxiliary engine valve event.

- 13.** A method of modifying engine valve lift in an internal combustion engine to produce an auxiliary engine valve event that is different from a normal engine operation, the engine including engine valve train including two engine valves, a valve bridge and an engine valve lifter for cyclically opening and closing the two engine valves, said method comprising the steps of:

- (a) providing an actuator integrated into said exhaust valve train, said actuator comprising a hydraulic system and a valve lash adjusting device, said hydraulic system comprising a hydraulic piston slidably disposed in said valve bridge, and said valve lash adjusting device being integrated into said engine valve lifter;
- (b) providing a controller for moving said hydraulic piston in said valve bridge between an inoperative position and an operative position with a fluid flow;
- (c) turning on said controller and supplying the fluid flow to said hydraulic piston;
- (d) moving said hydraulic piston from said inoperative position to said operative position;
- (e) acting on said hydraulic piston by said actuator through said valve lash adjusting device; and
- (f) opening one of the two engine valves for said auxiliary engine valve event.

- 14.** The method of claim 13, further comprising the steps of:

- (a) providing an actuator with an auxiliary cam lobe integrated into a cam of said engine valve lifter, a cam lobe of the cam for the normal engine operation being enlarged to accommodate the integration of said auxiliary cam lobe;
- (b) providing a reset device for modifying the valve lift profile produced by the enlarged cam lobe, said reset device comprising a reset piston and a reset flow passage;

- (c) moving said reset piston and uncovering said reset flow passage;
- (d) draining the fluid from said hydraulic piston and moving said hydraulic piston from said operative position to said inoperative position;
- (e) disengaging said hydraulic piston from the engine valves;
- (f) opening and then closing both of the two engine valves by the enlarged cam lobe through said engine valve lifter;
- (g) moving said reset piston and blocking said reset flow passage;
- (h) refilling said hydraulic piston and moving said hydraulic piston from said inoperative position back to said operative position;
- (i) actuating said hydraulic piston with the auxiliary cam lobe through said valve lash adjusting device; and
- (j) opening one of the two engine valves for said auxiliary engine valve event.

- 15.** The method of claim 13, further comprising the steps of:

- (a) moving said engine valve lifter forward by the enlarged cam lobe;
- (b) blocking the fluid flow from said controller to said hydraulic piston;
- (c) moving said valve bridge by said engine valve lifter;
- (d) opening said reset flow passage and draining out the fluid from said hydraulic piston;
- (e) moving said hydraulic piston from said operative position to said inoperative position;
- (f) disengaging said actuator from the engine valves;
- (g) moving both of the two engine valves by the enlarged cam lobe through said engine valve lifter;
- (h) moving said engine valve lifter back and directing the fluid flow from said controller to said hydraulic piston;
- (i) refilling said hydraulic piston and moving said hydraulic piston from said inoperative position back to said operative position;
- (j) blocking the reset flow passage and actuating said hydraulic piston by the auxiliary cam lobe through said valve lash adjusting device; and
- (k) opening the engine valve for said auxiliary engine valve event.

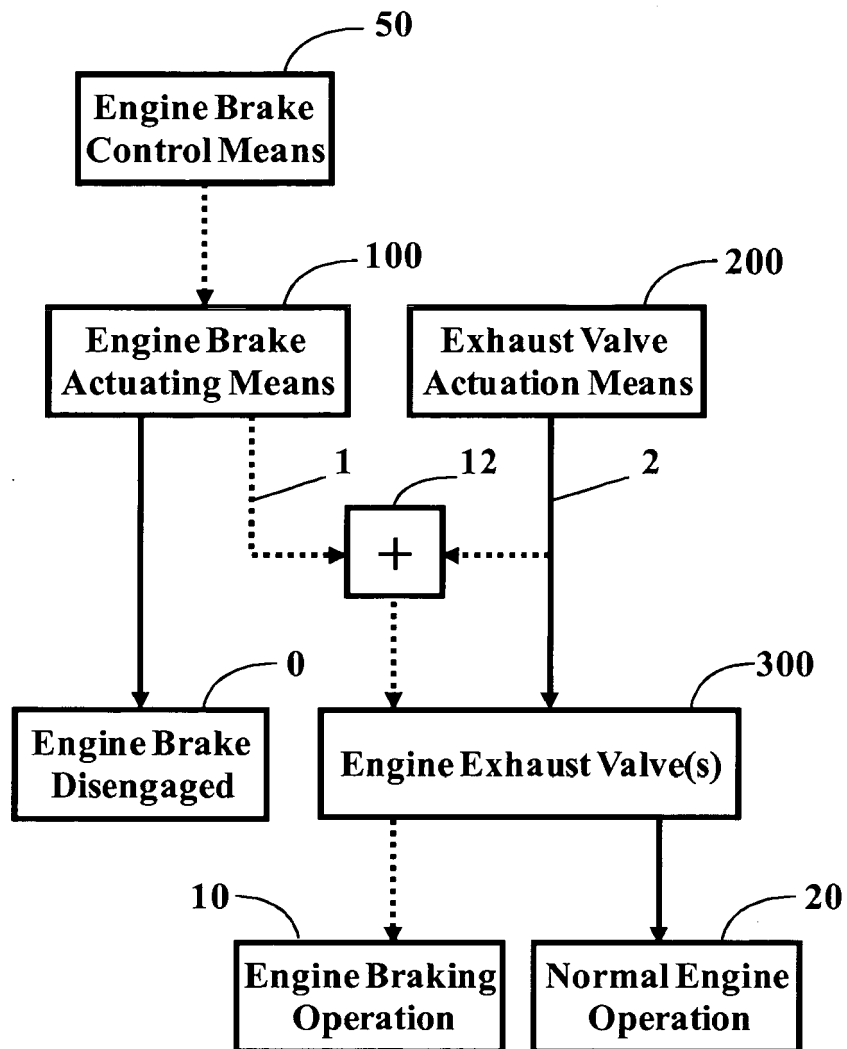


FIG. 1

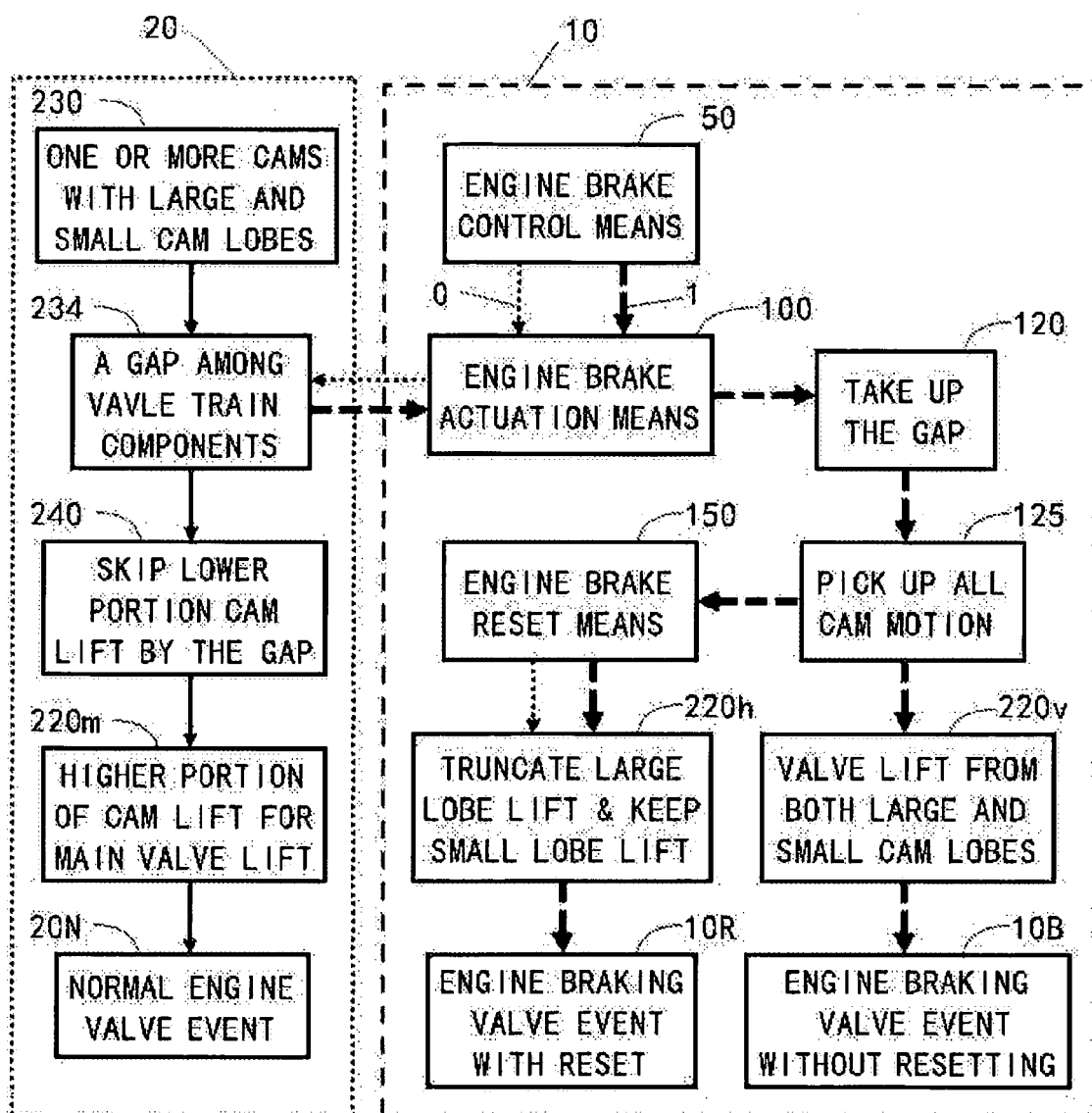


FIG. 2

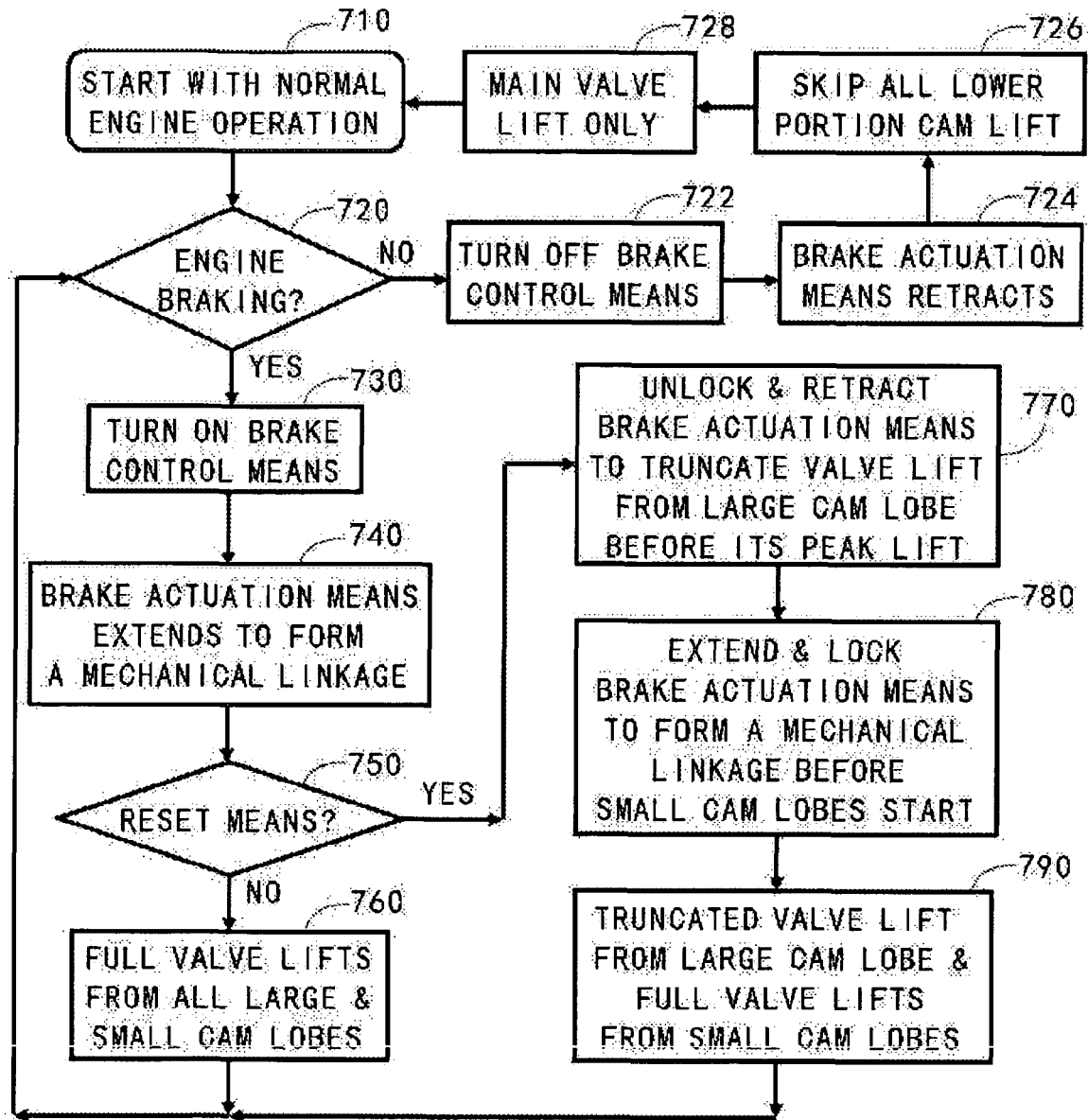
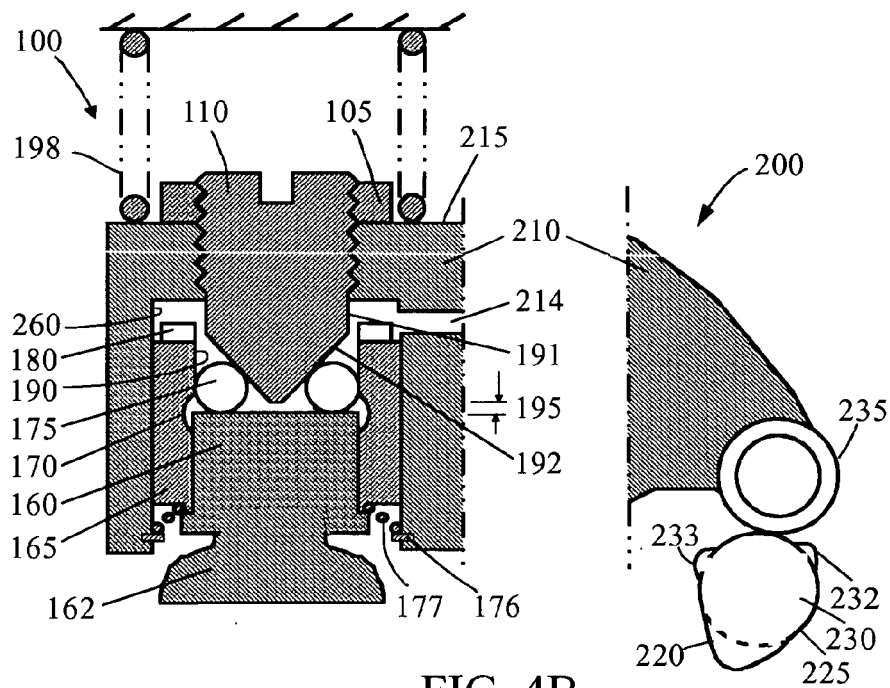
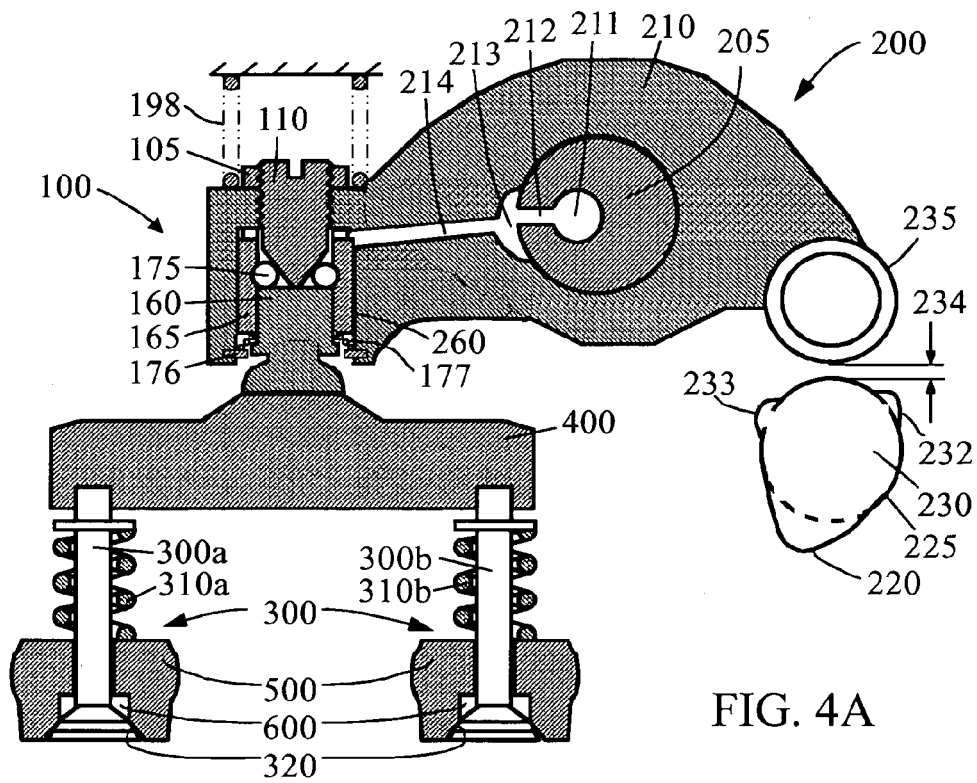
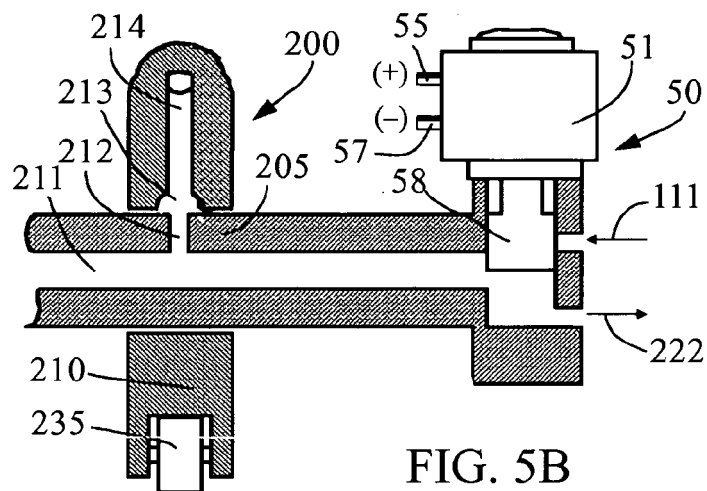
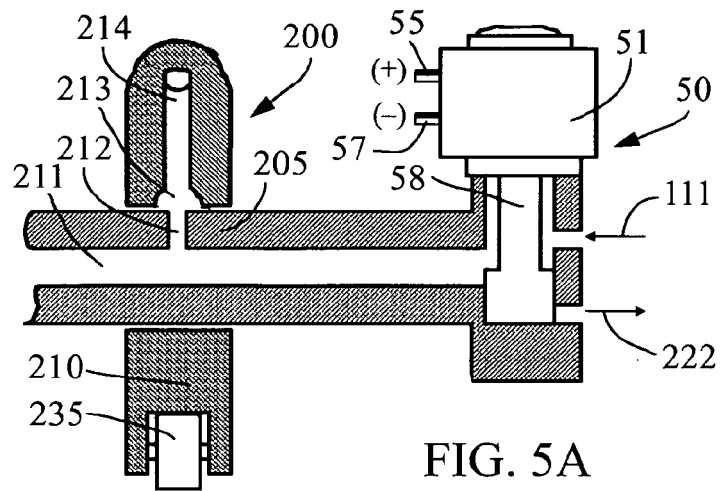


FIG. 3





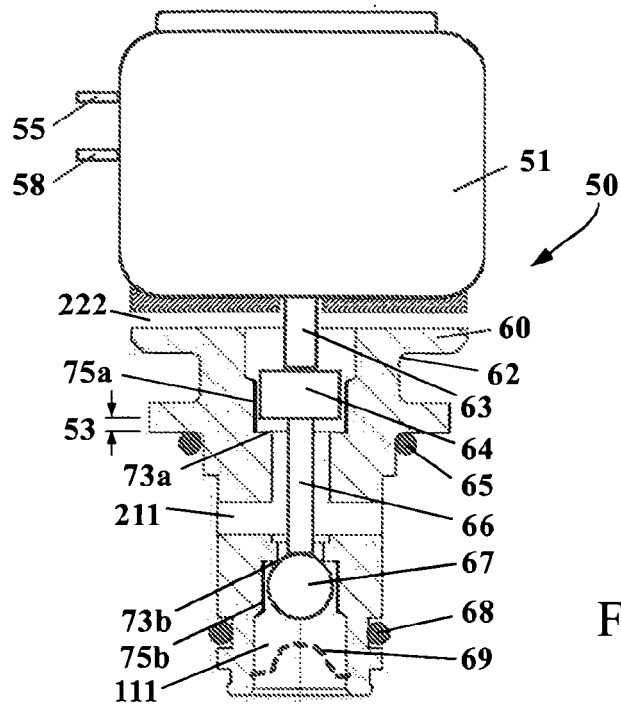


FIG. 6A

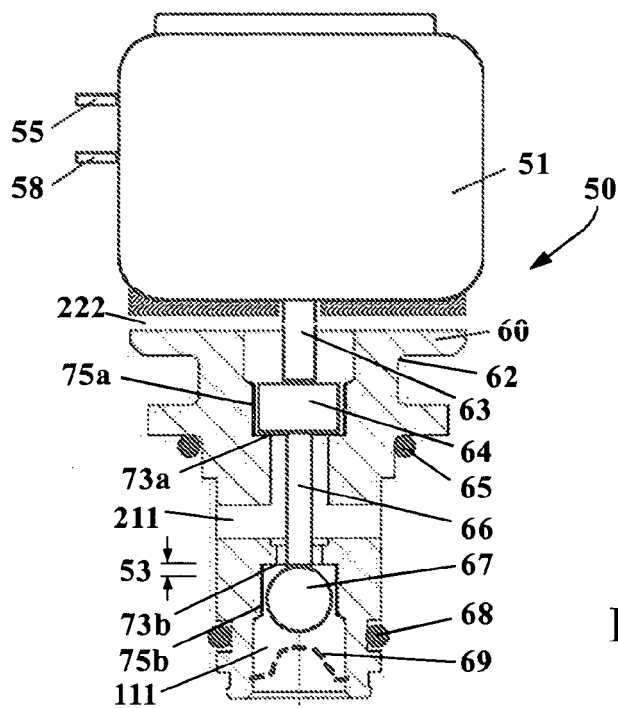


FIG. 6B

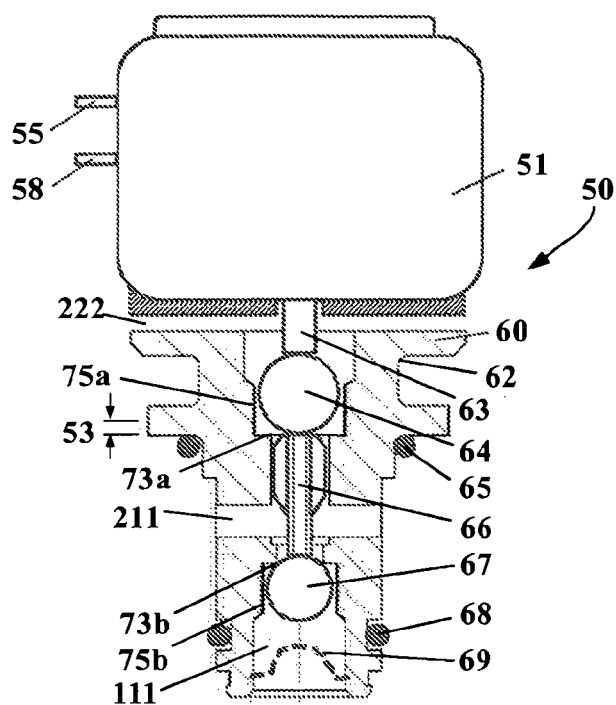


FIG. 7A

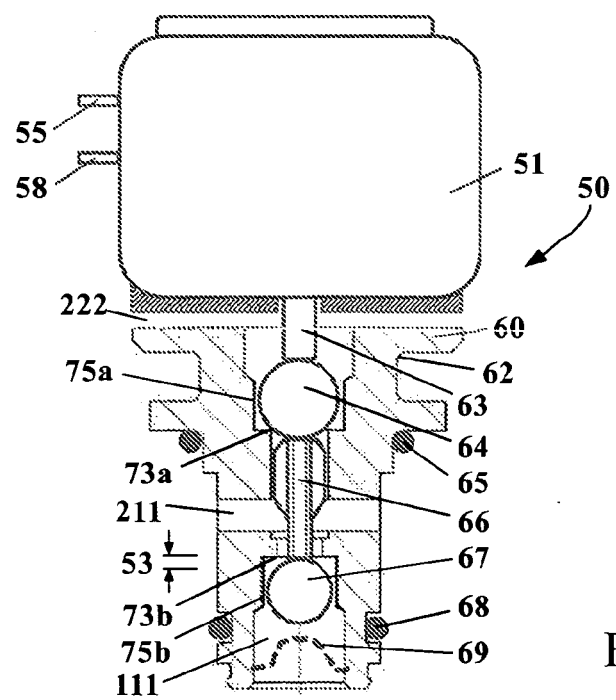
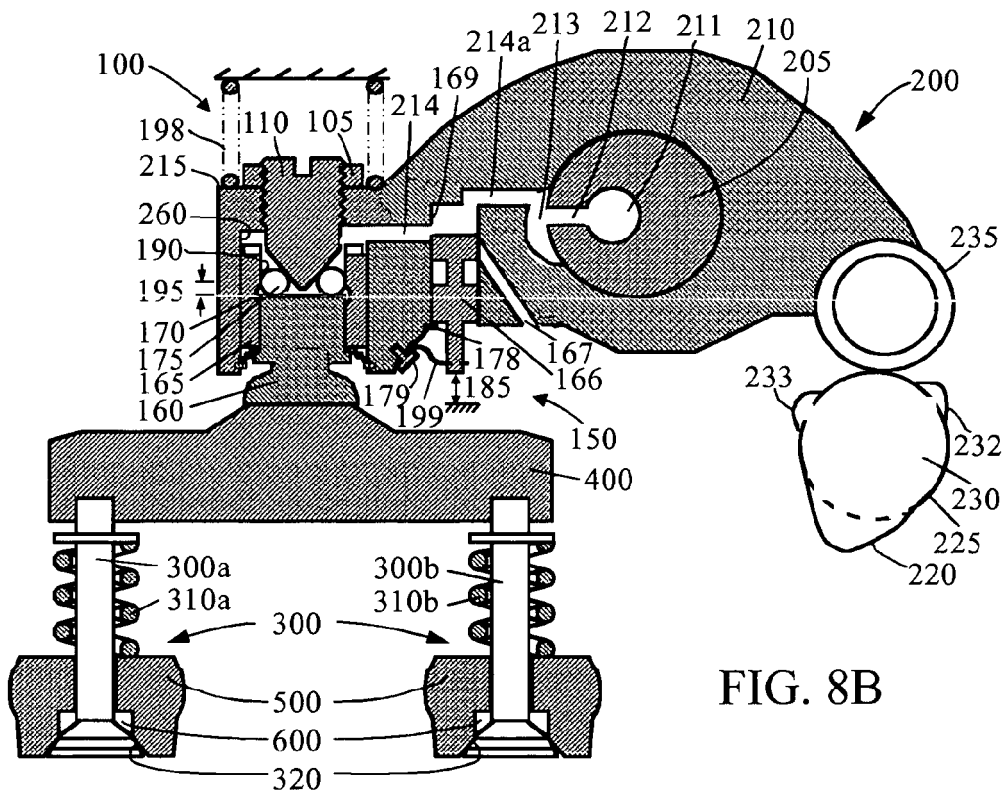
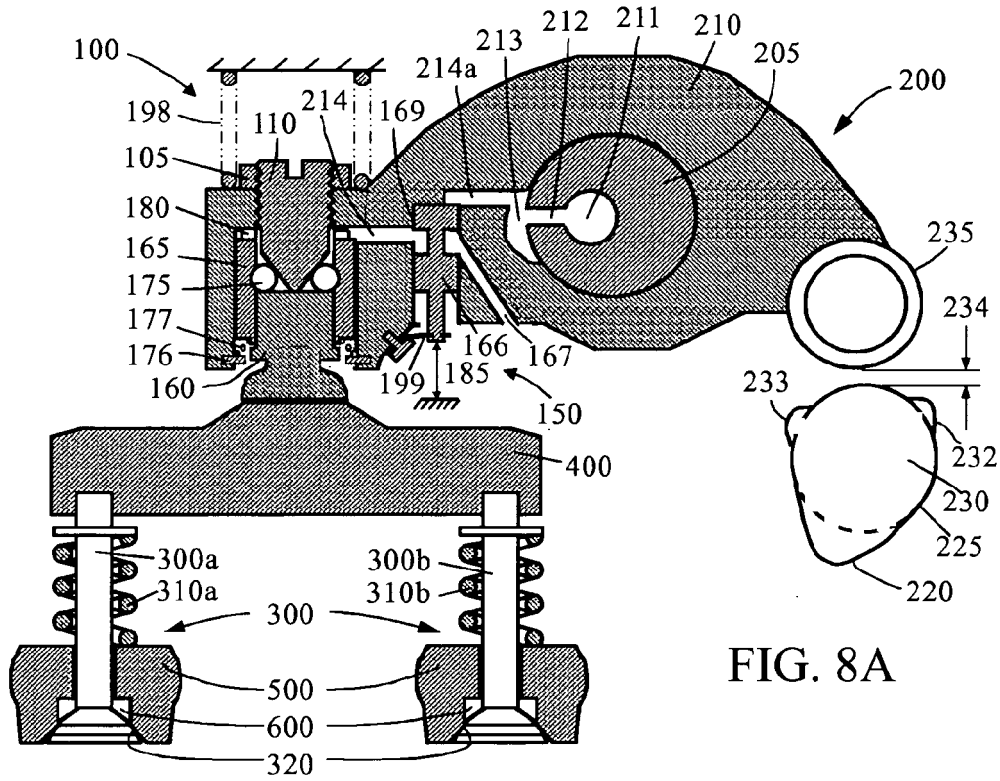


FIG. 7B



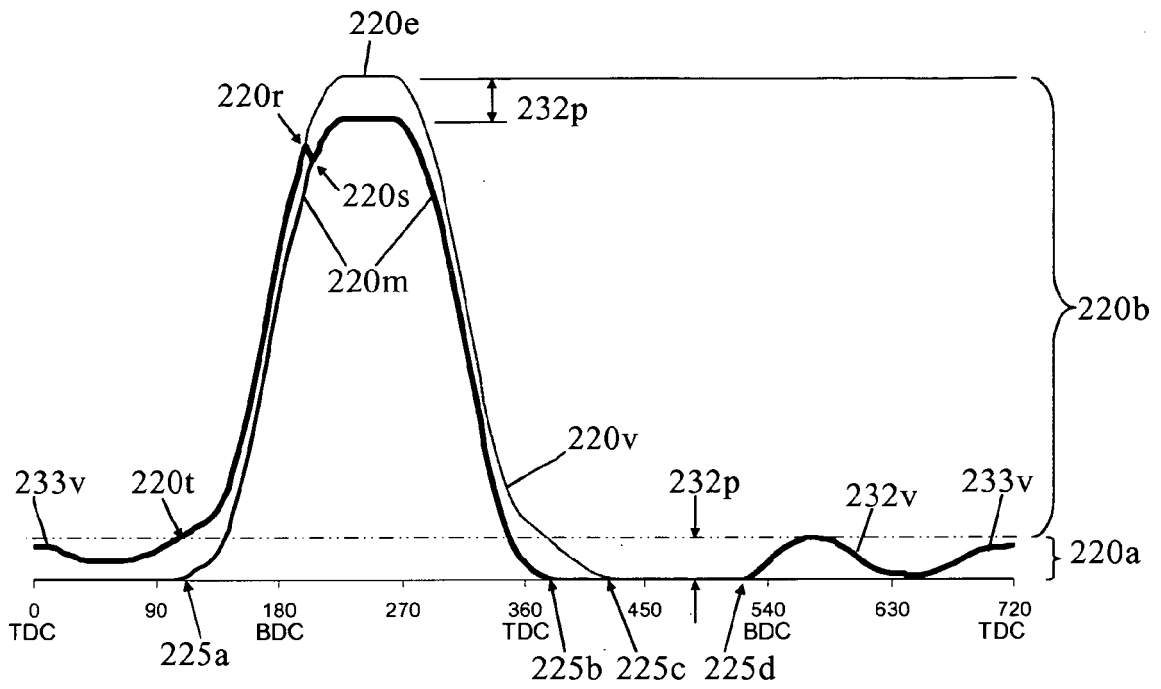


FIG. 9

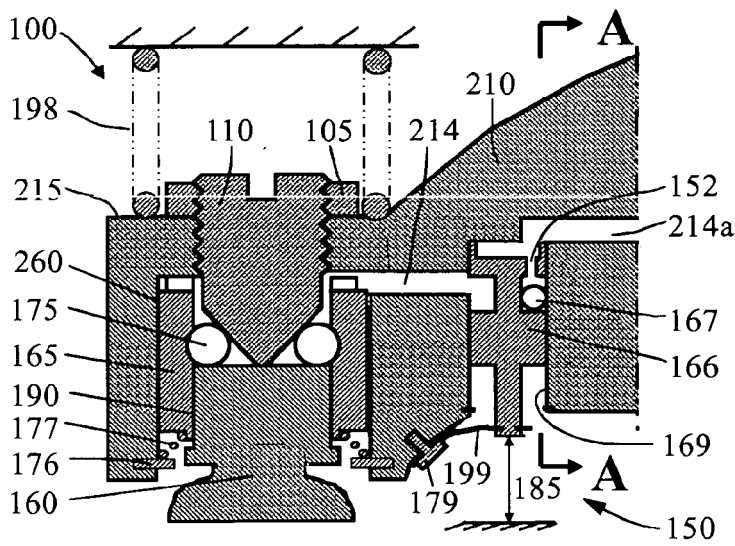


FIG. 10

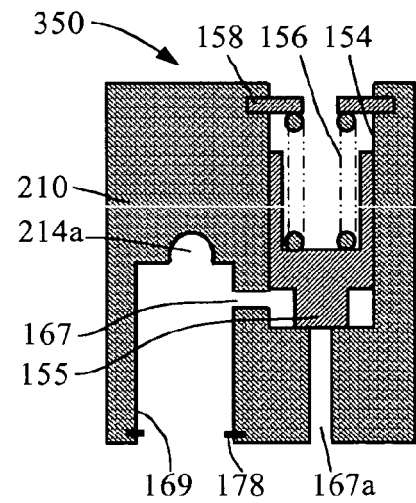


FIG. 10A-A

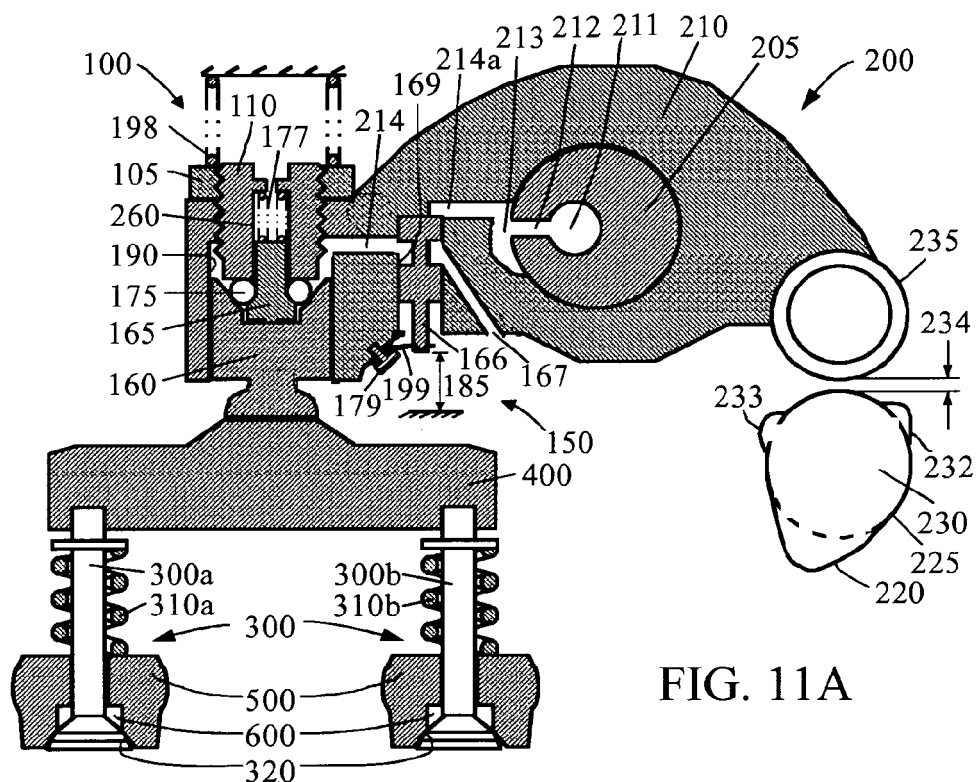


FIG. 11A

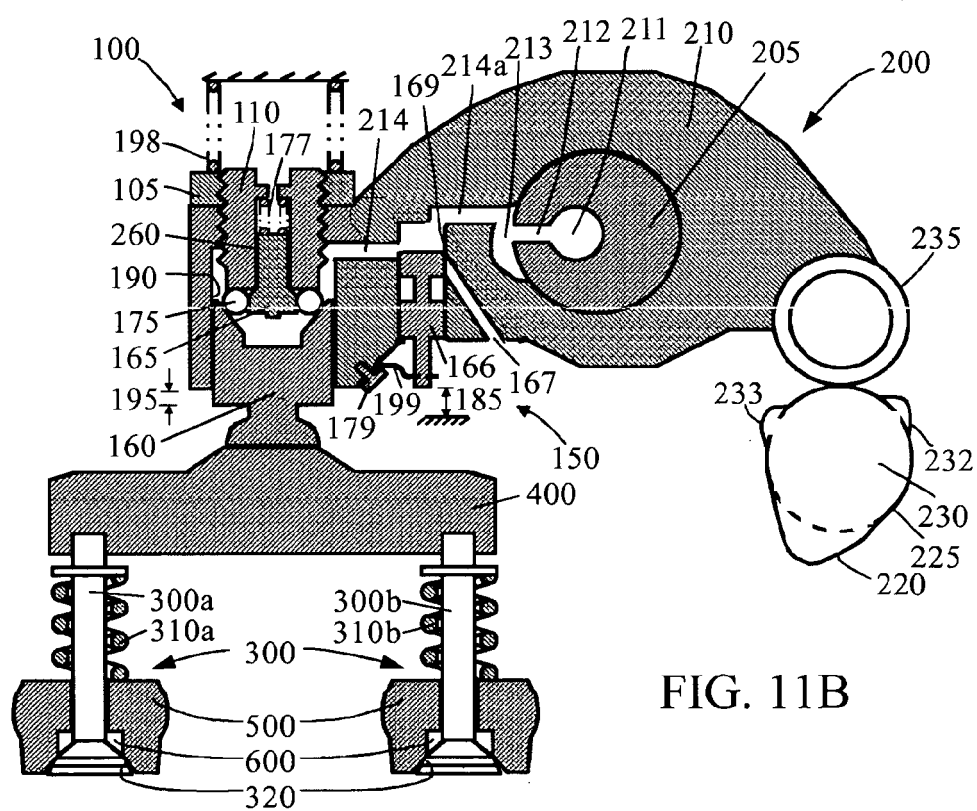
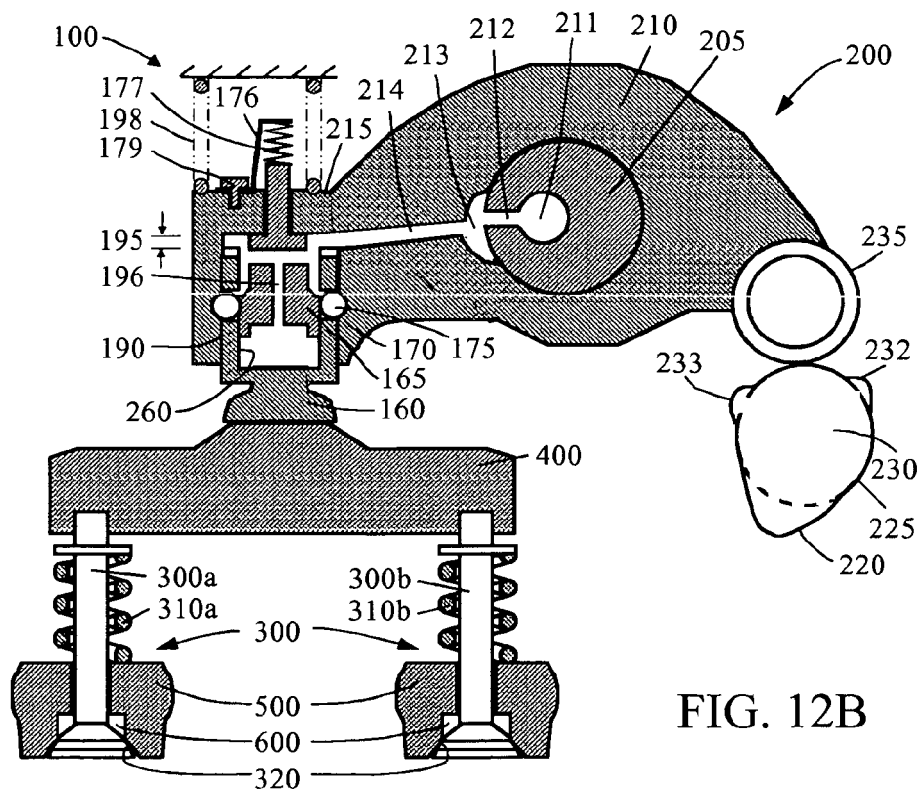
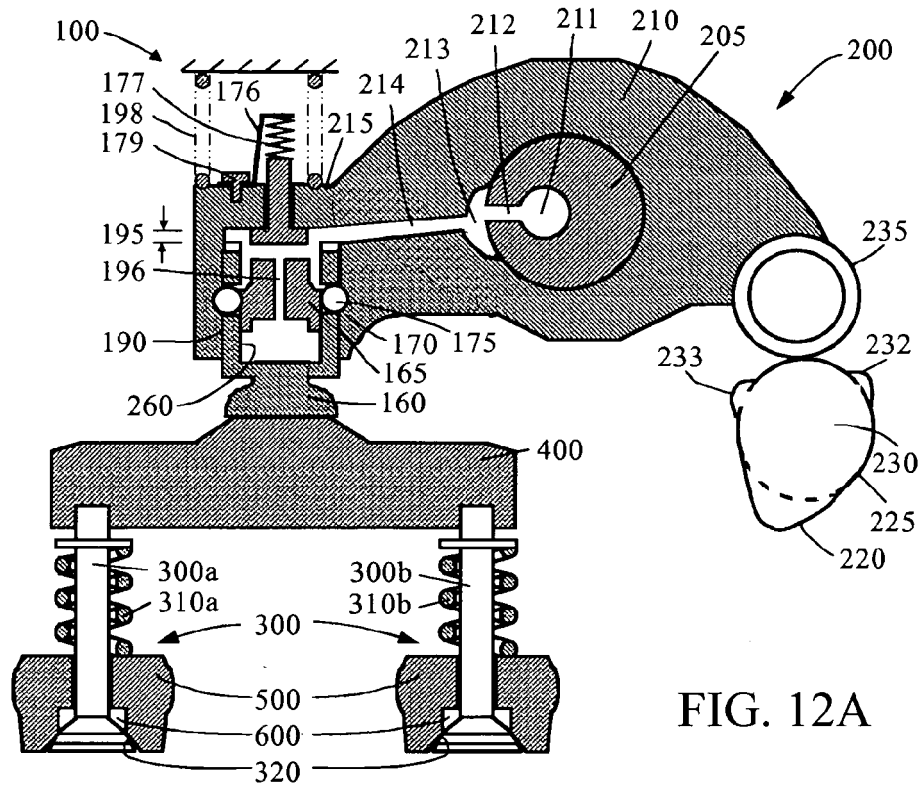
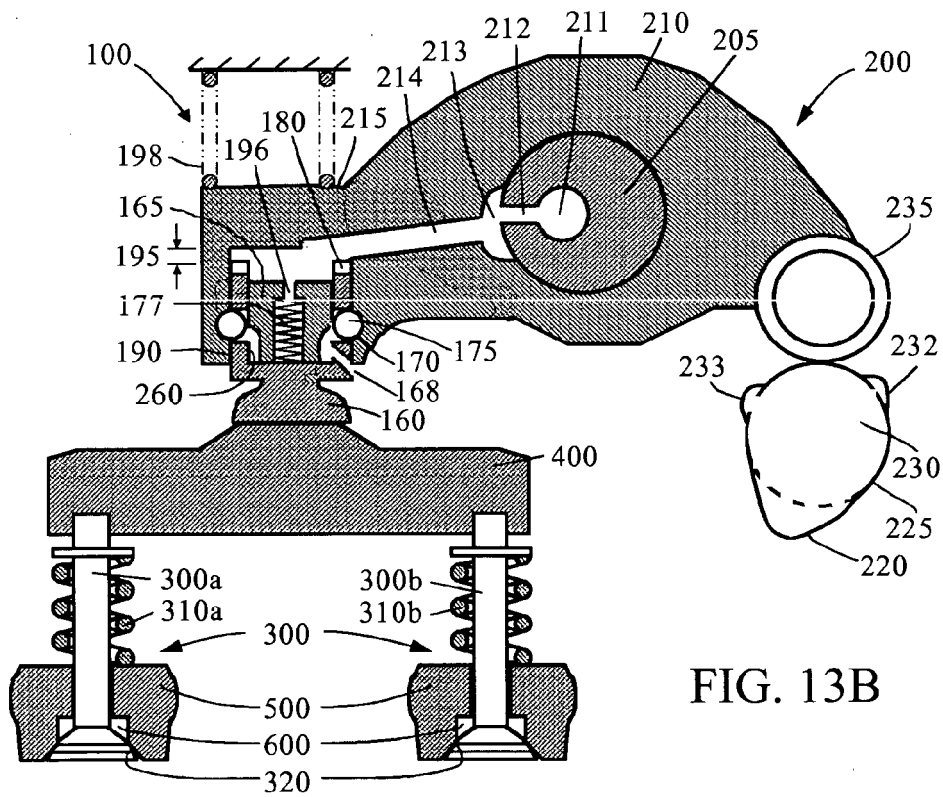
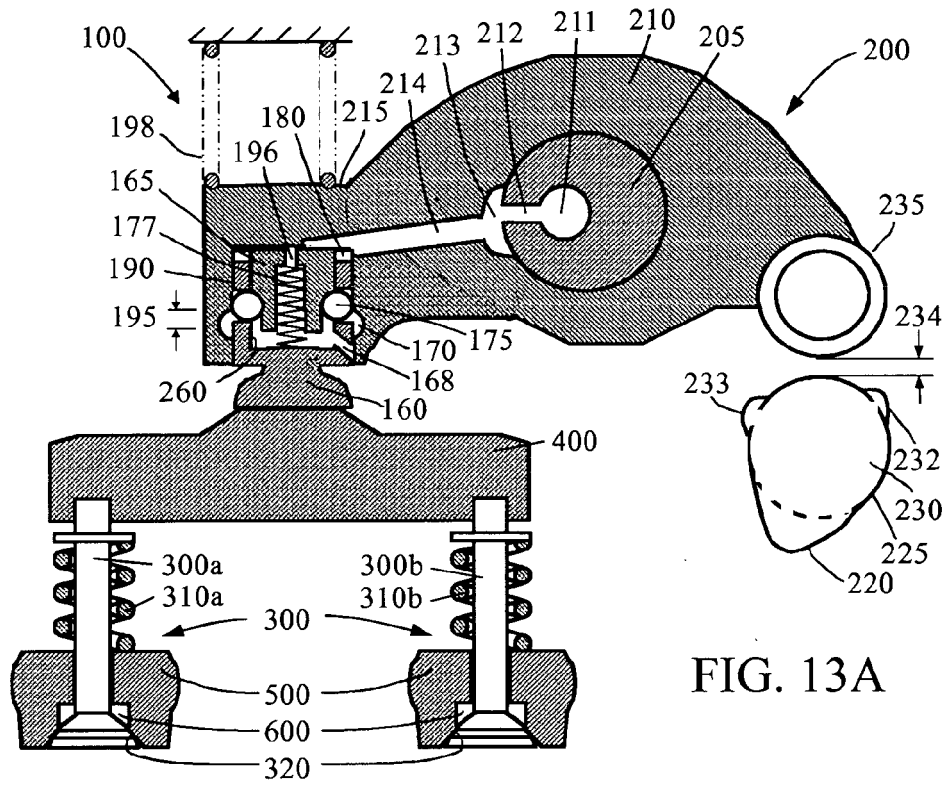
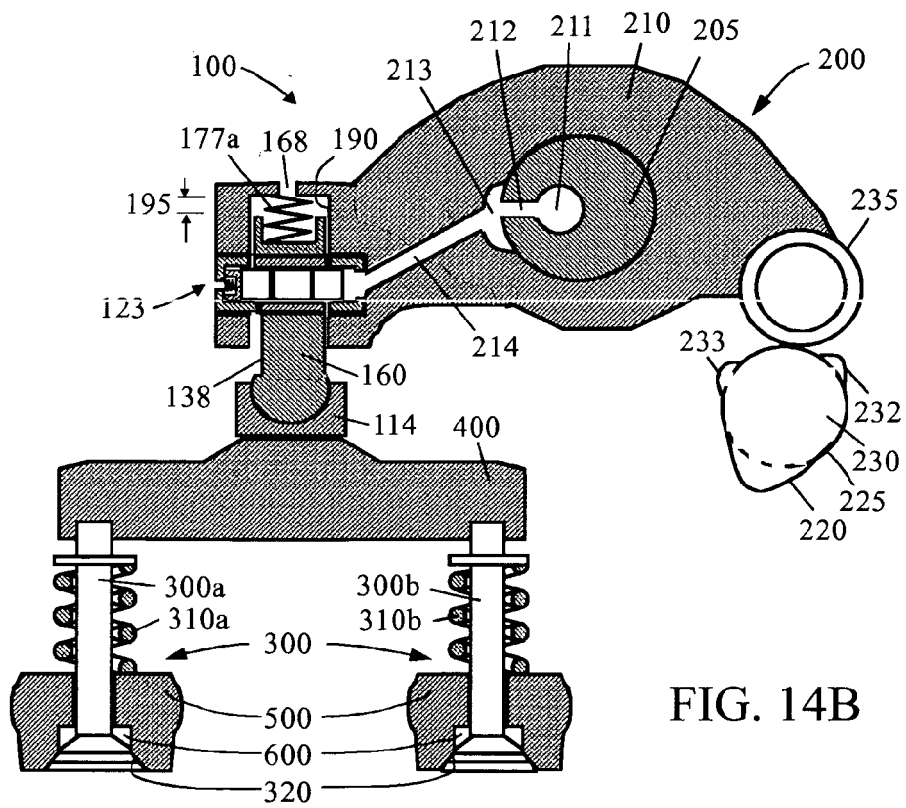
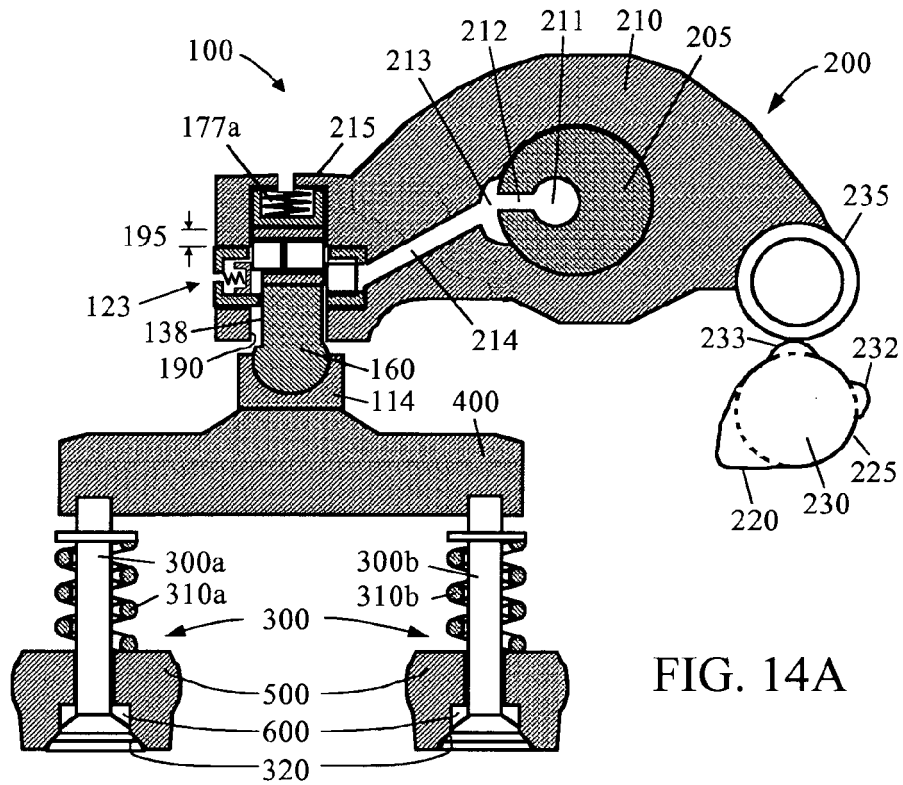
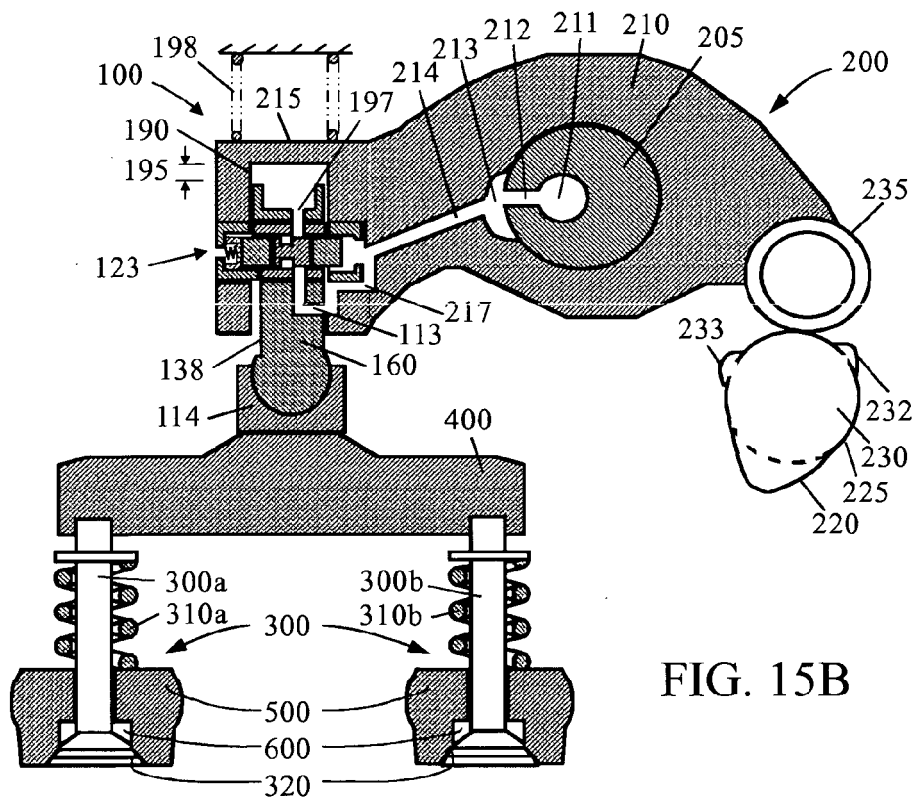
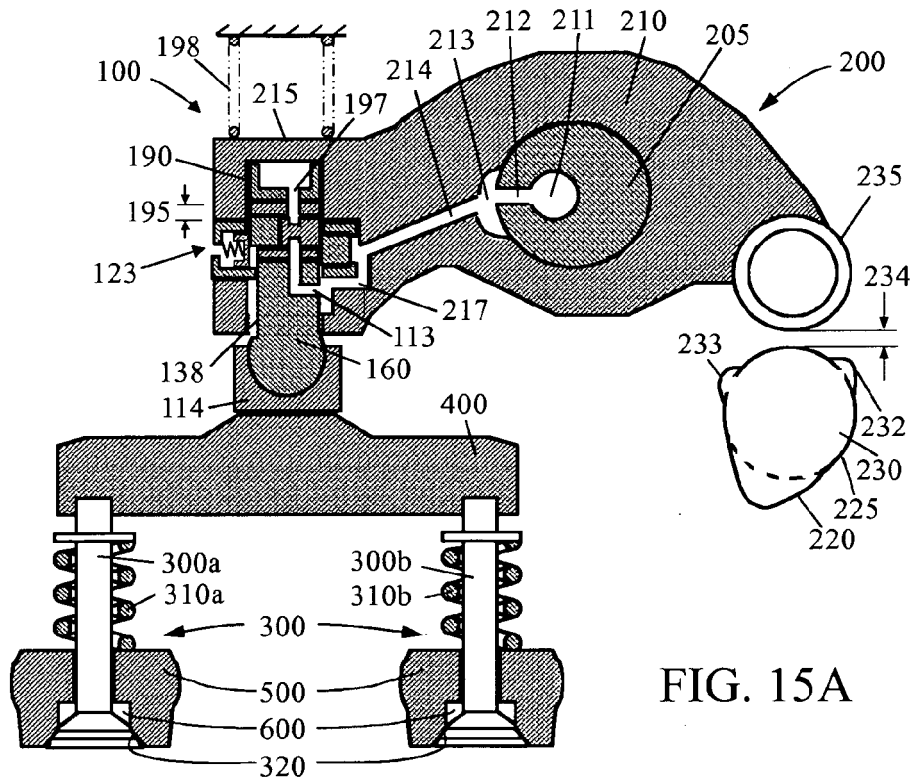


FIG. 11B









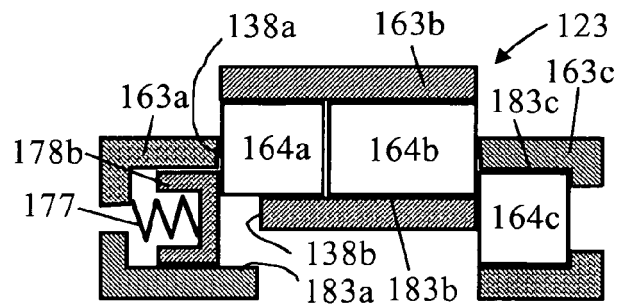


FIG. 14C

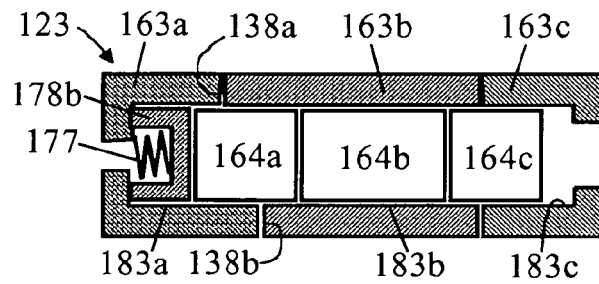


FIG. 14D

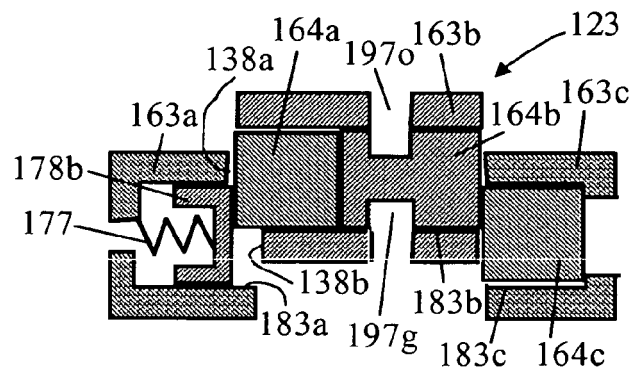


FIG. 15C

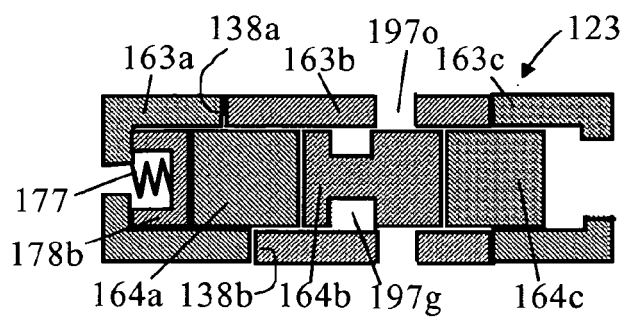
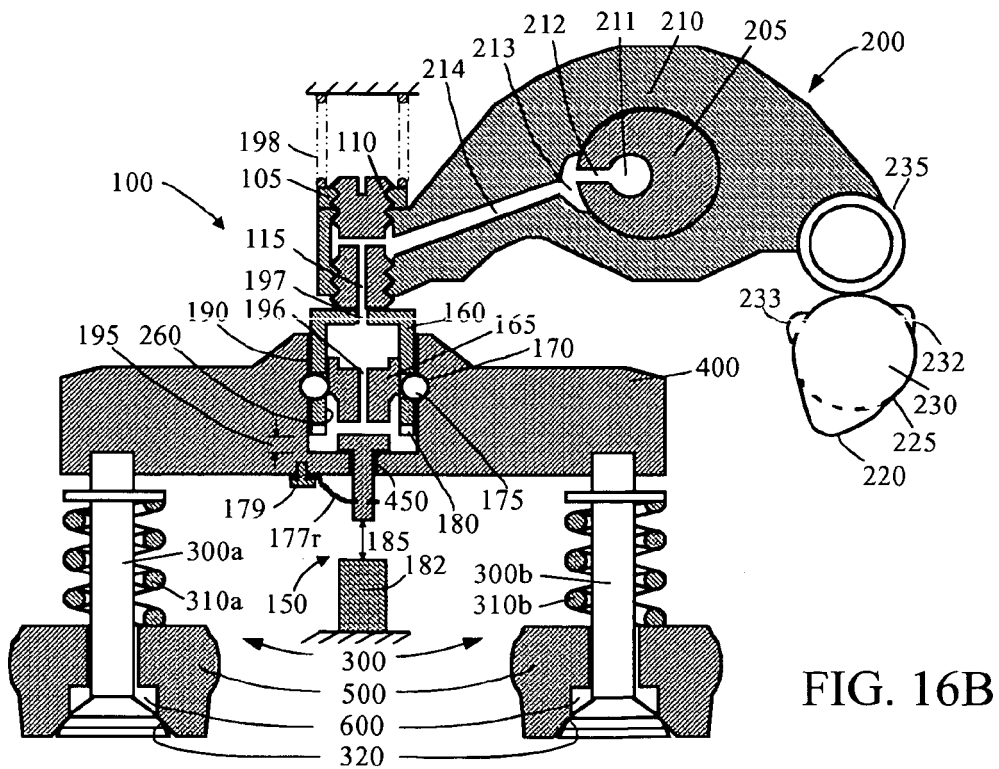
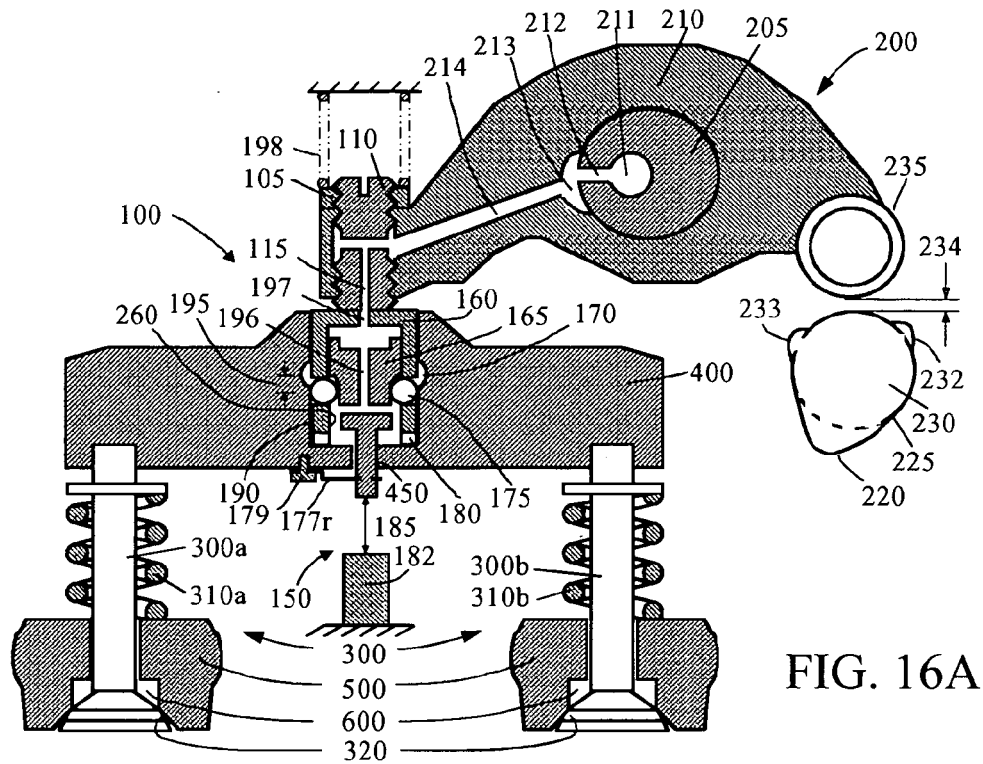
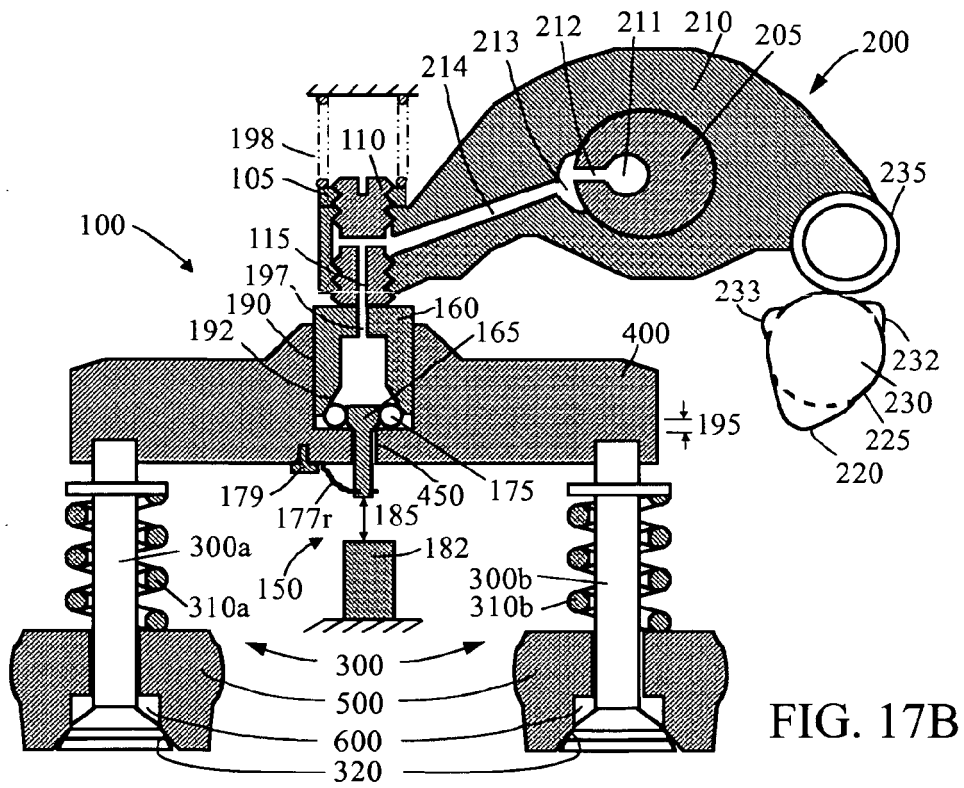
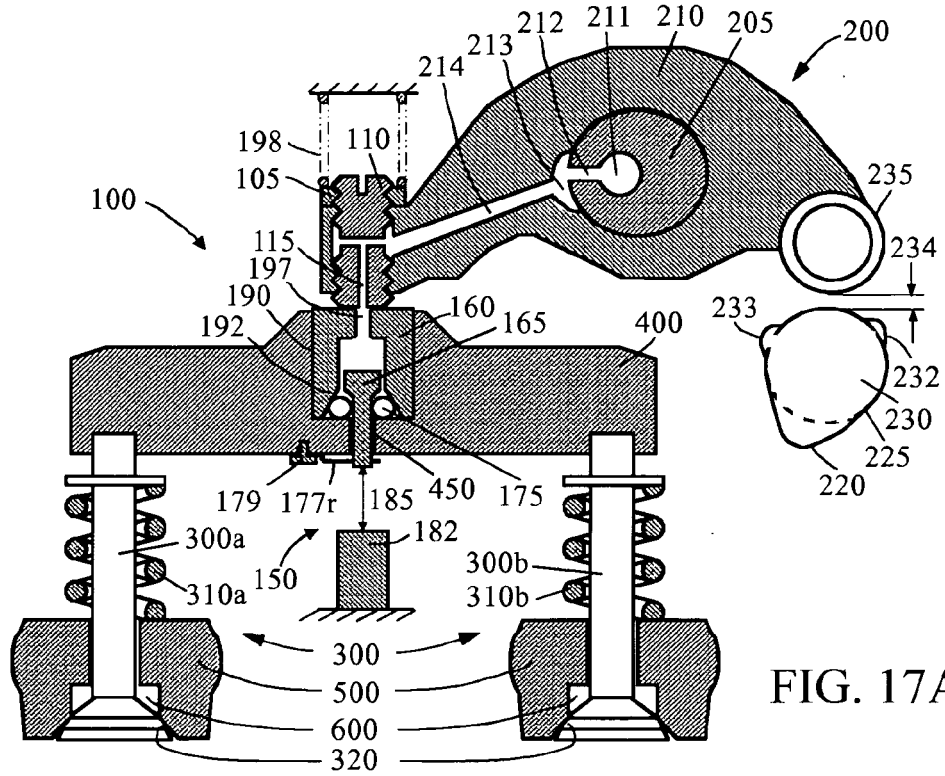


FIG. 15D





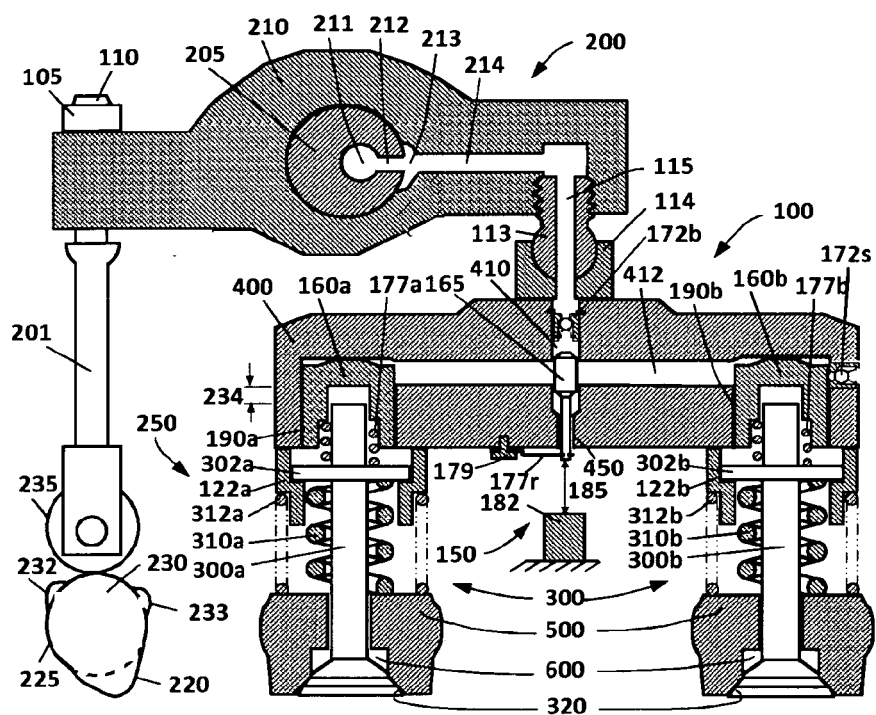


FIG. 18A

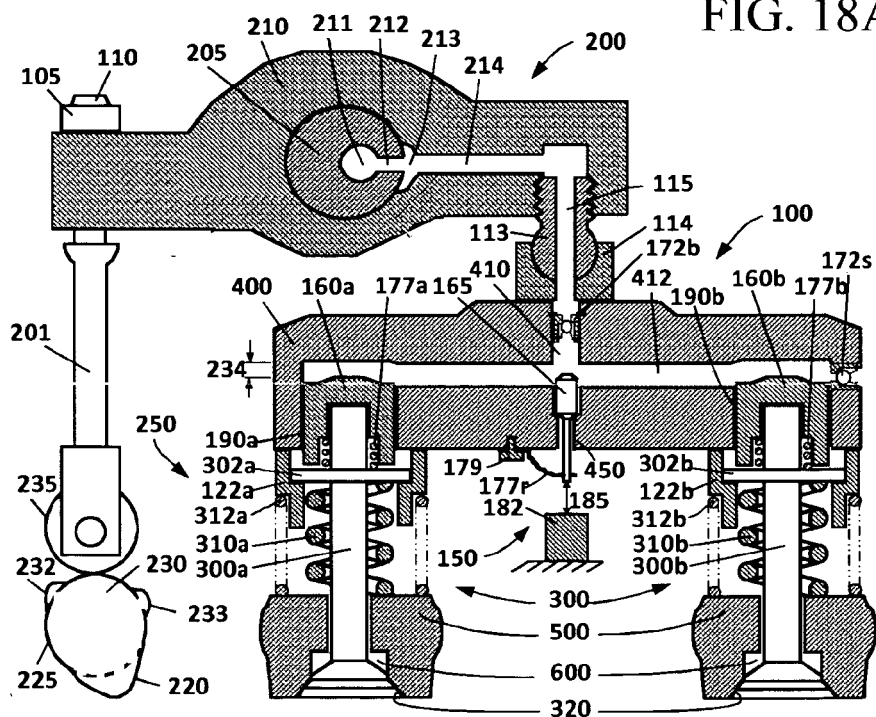


FIG. 18B

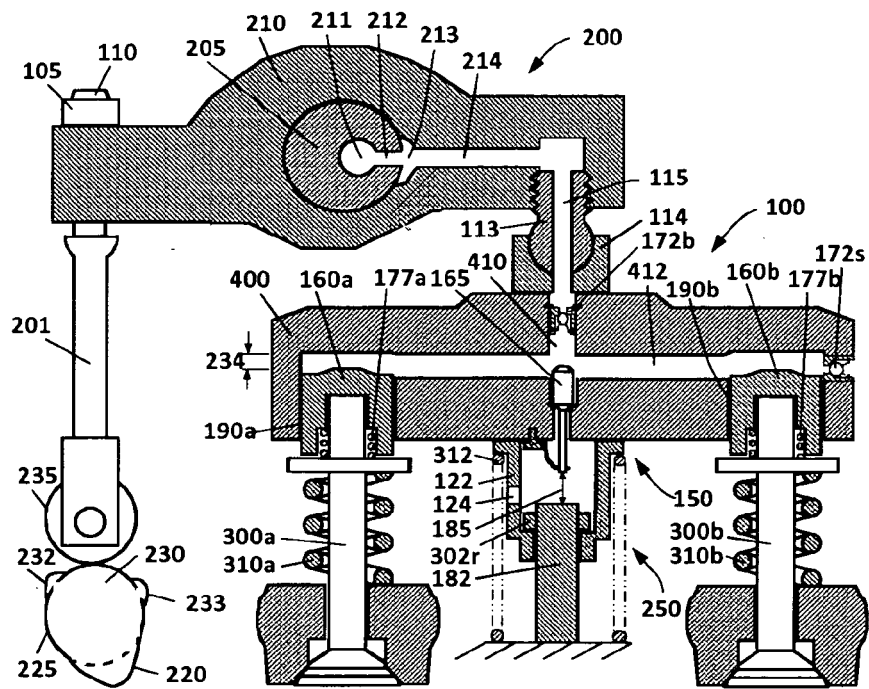


FIG. 19

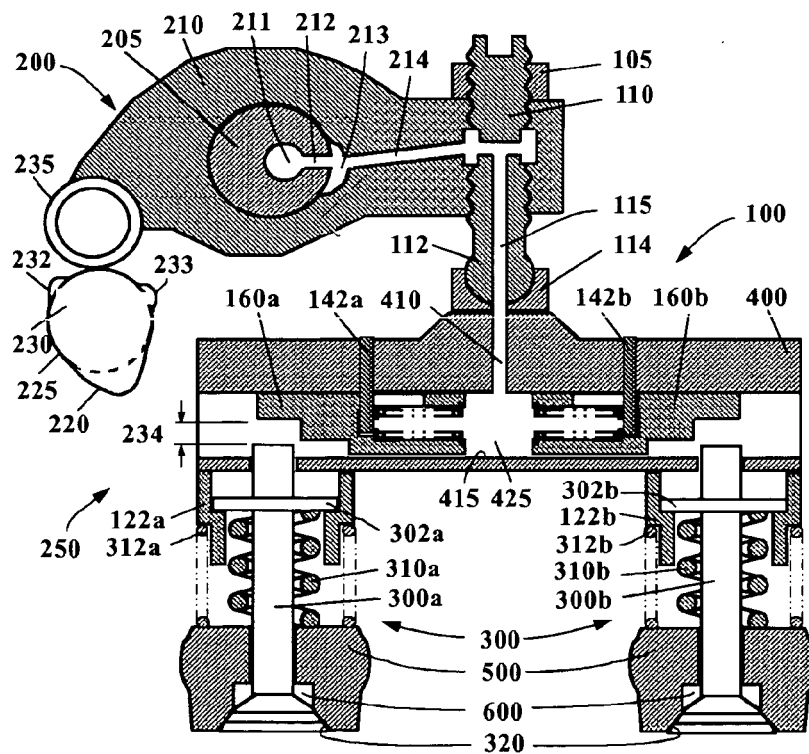


FIG. 20A

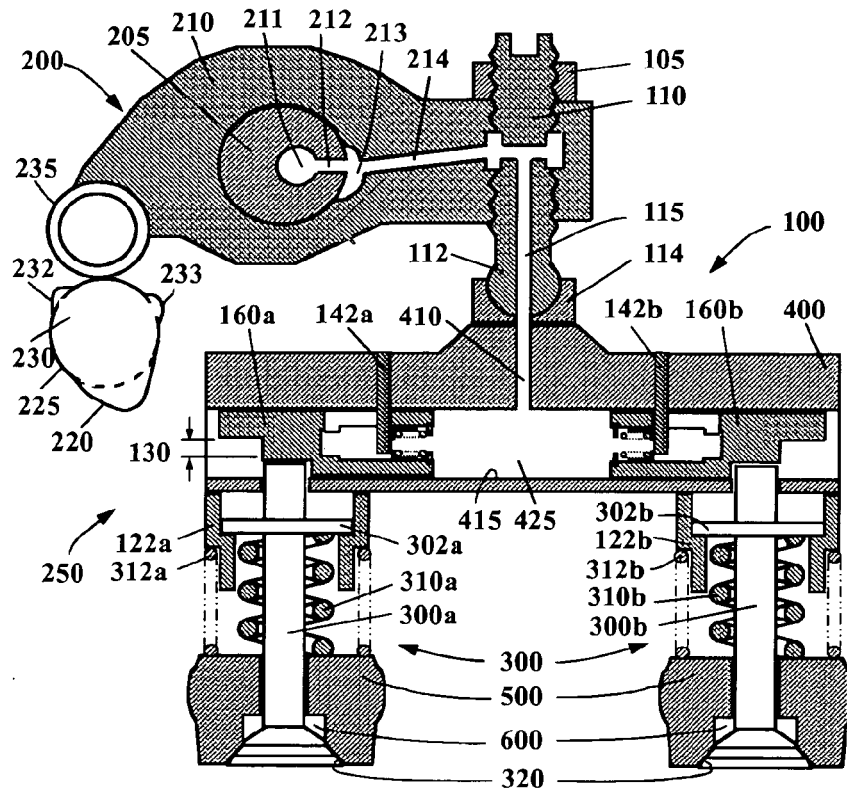


FIG. 20B

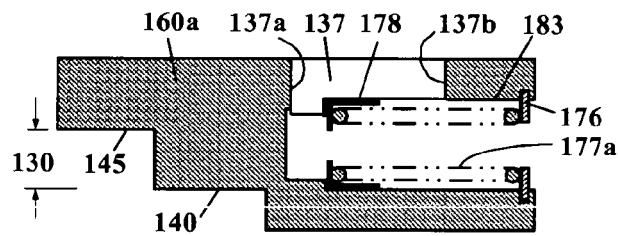


FIG. 20C

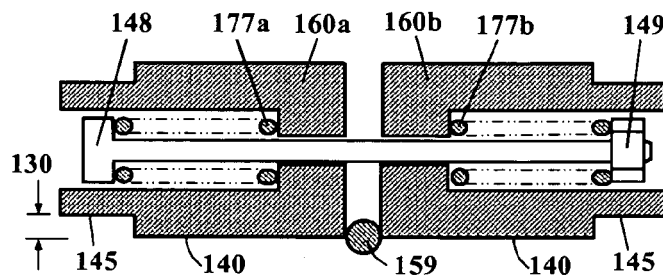


FIG. 20D

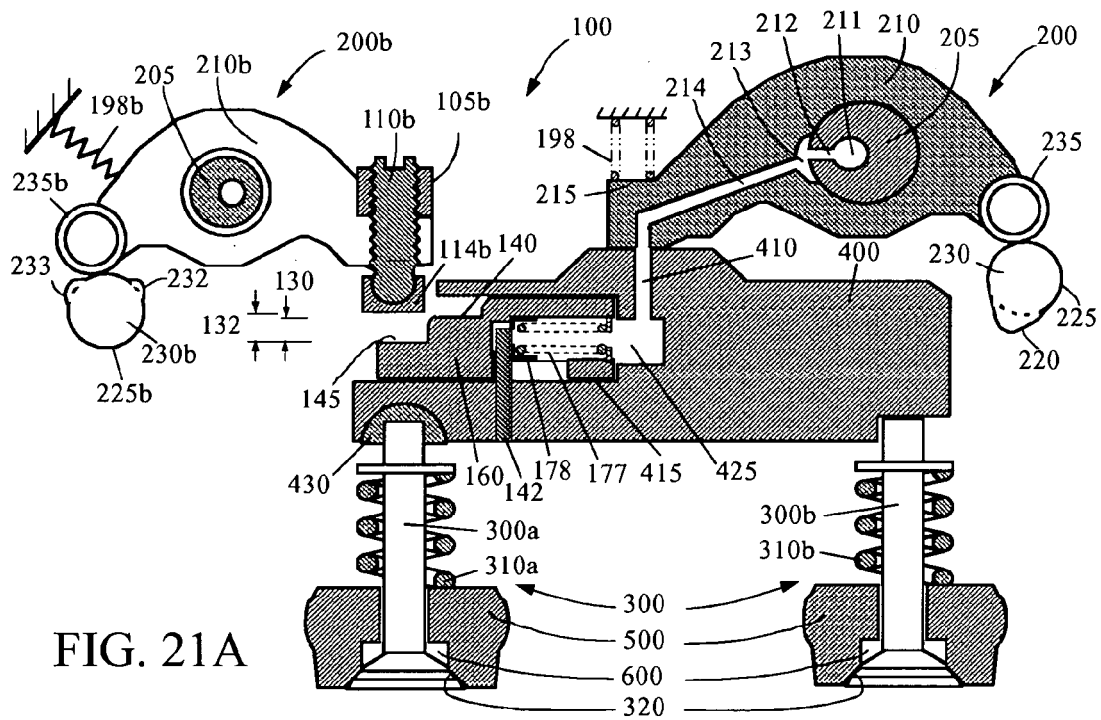


FIG. 21A

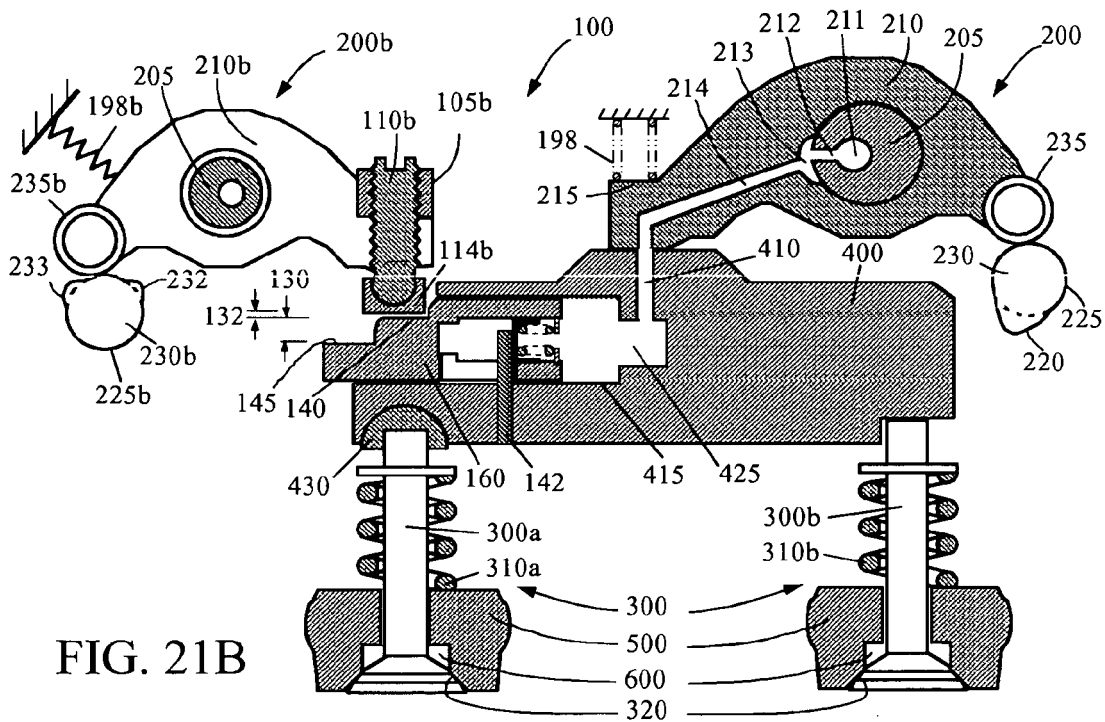
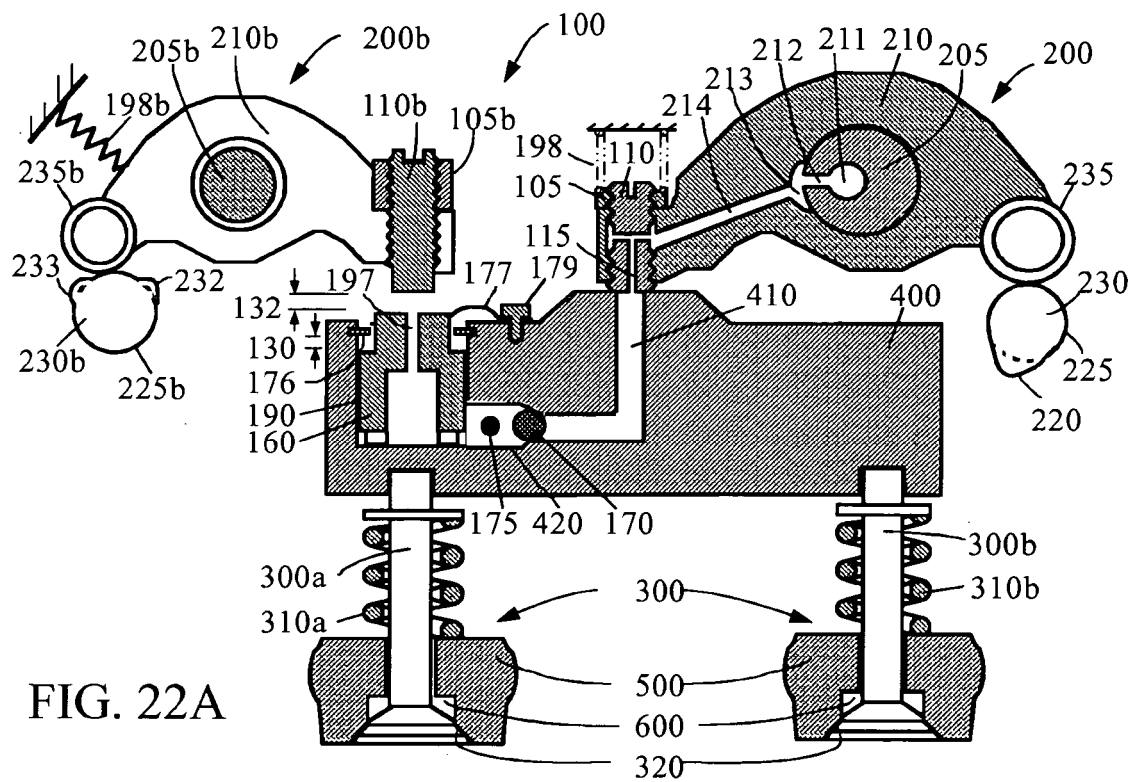
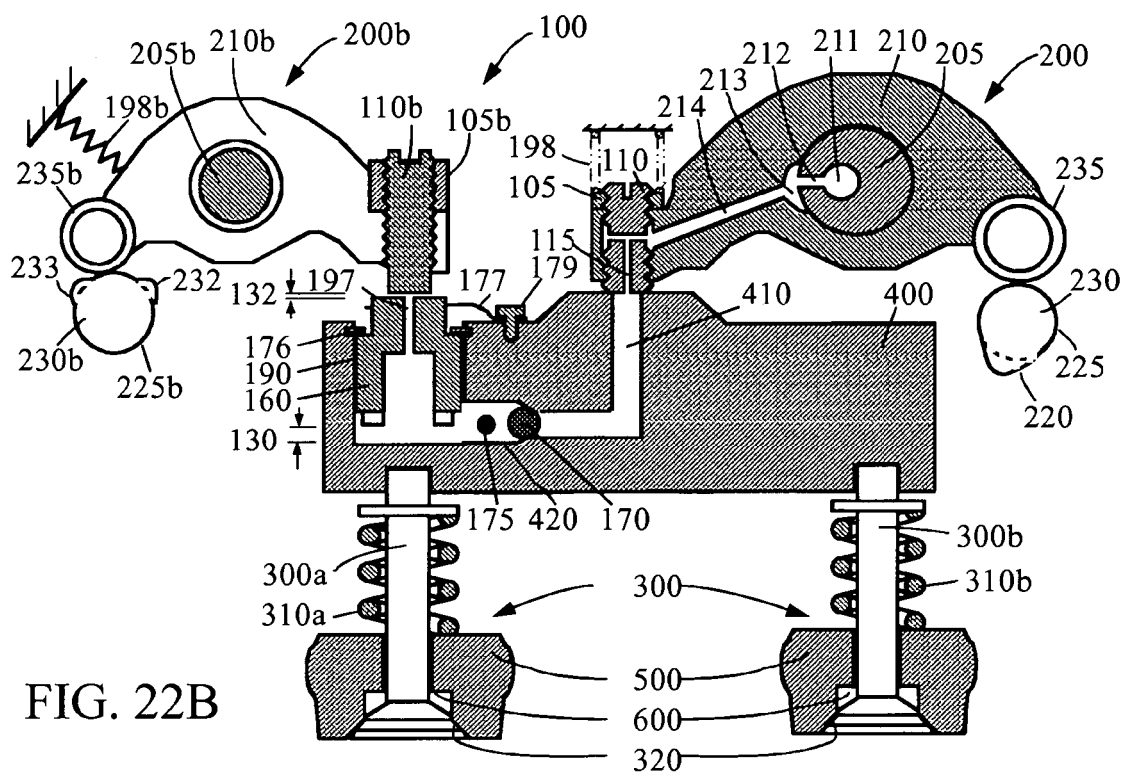
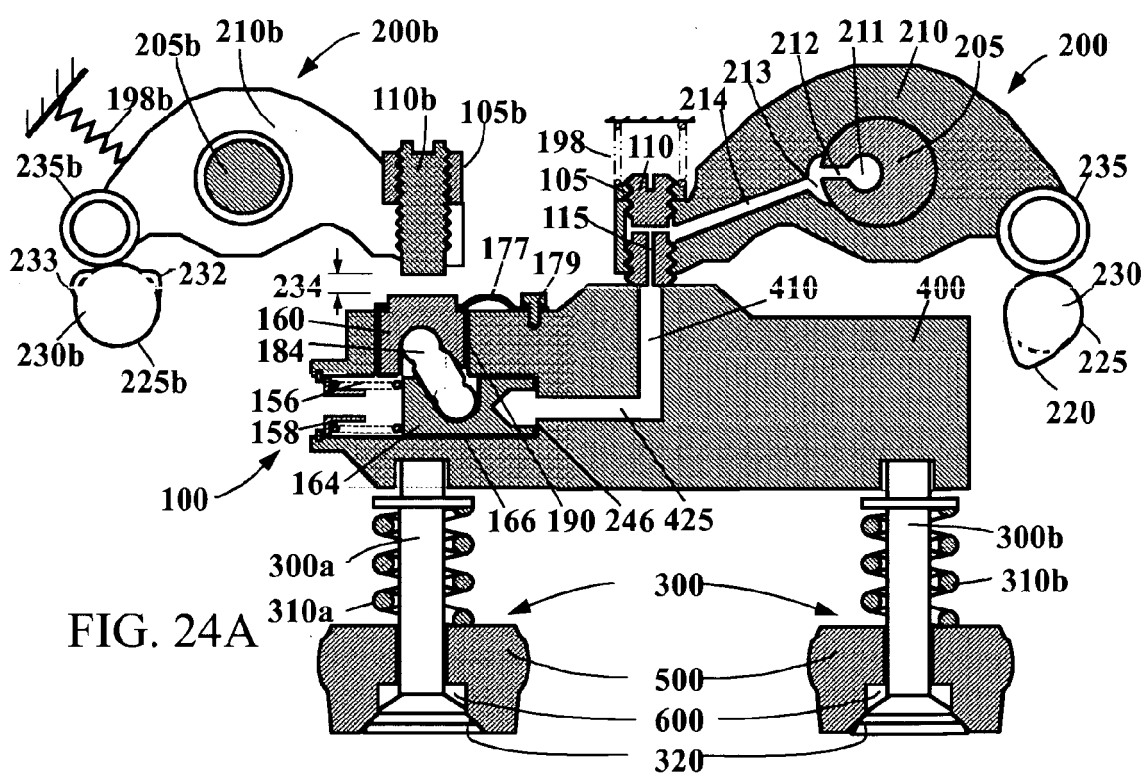
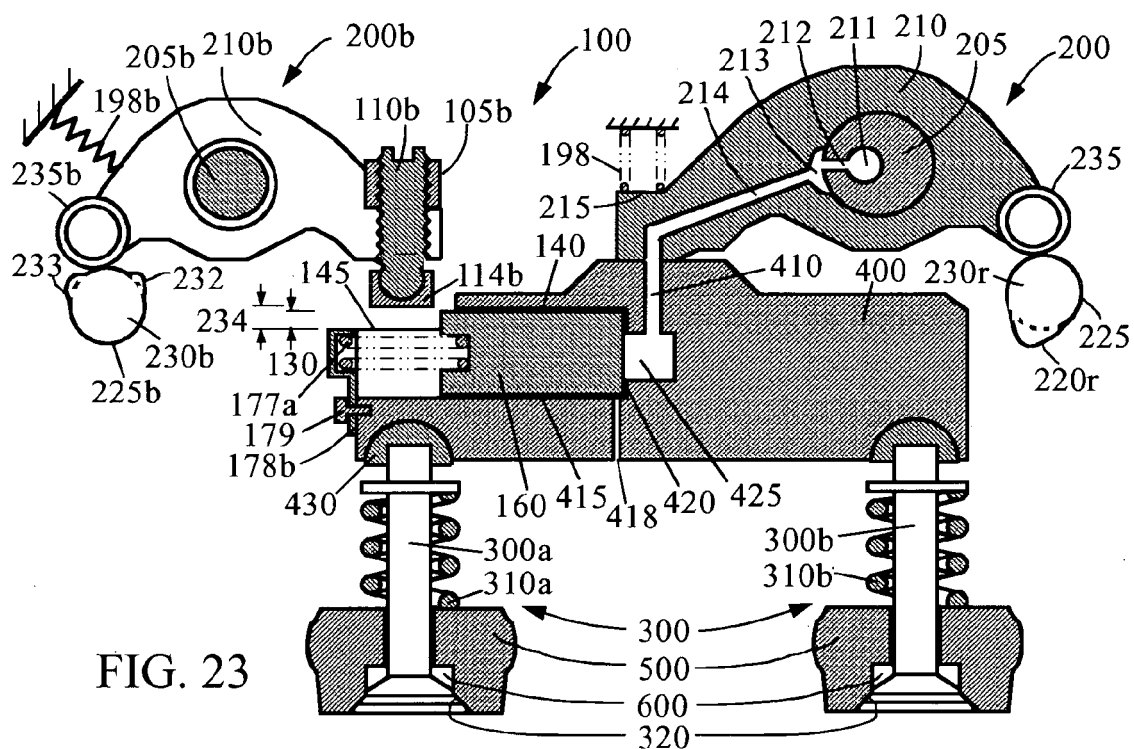
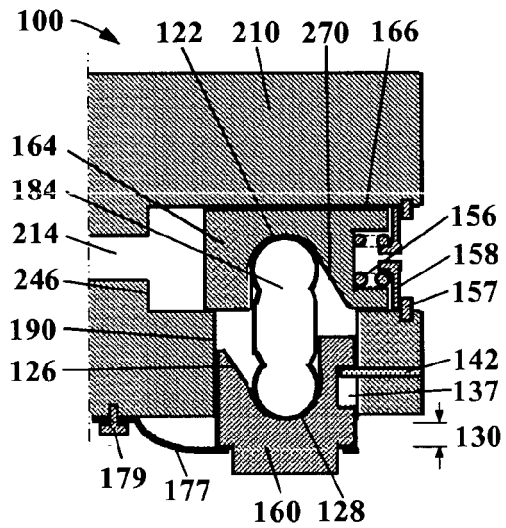
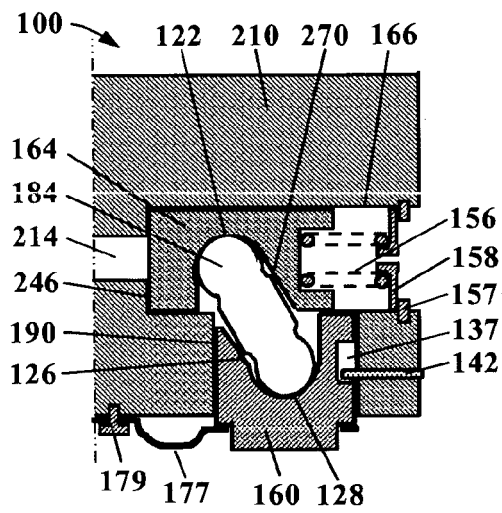
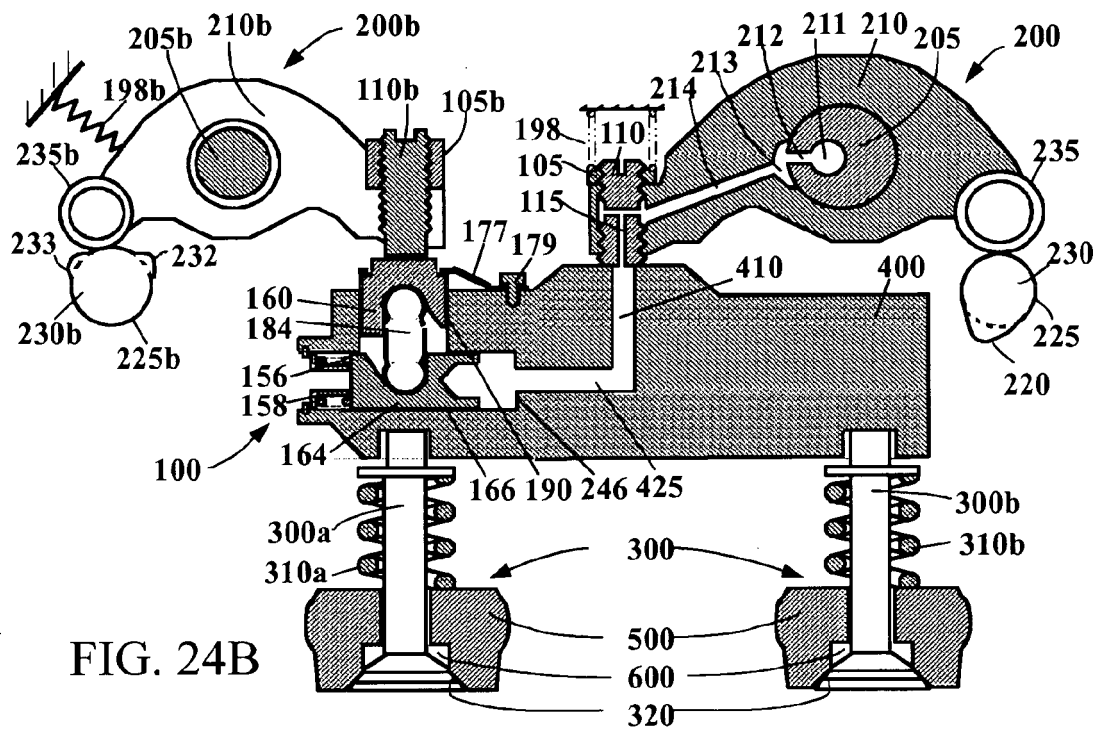


FIG. 21B









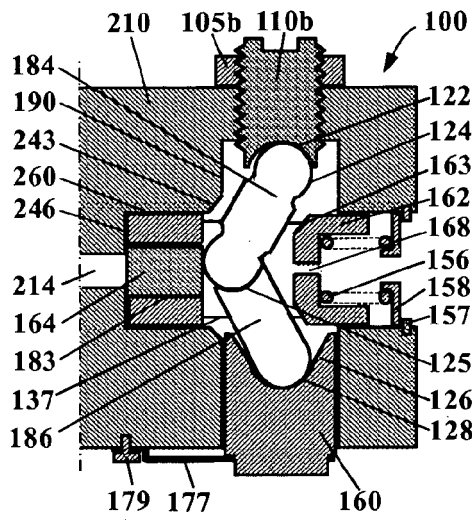


FIG. 25A

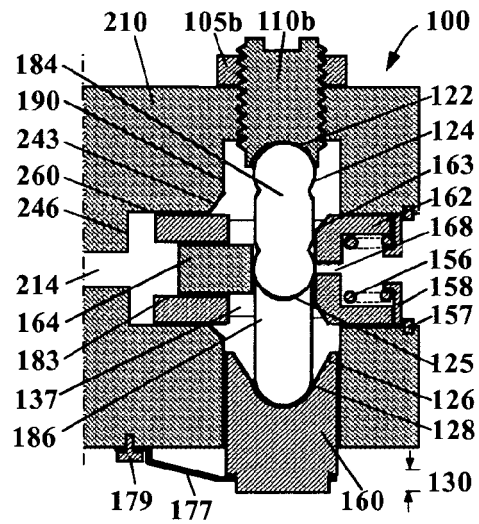


FIG. 25B

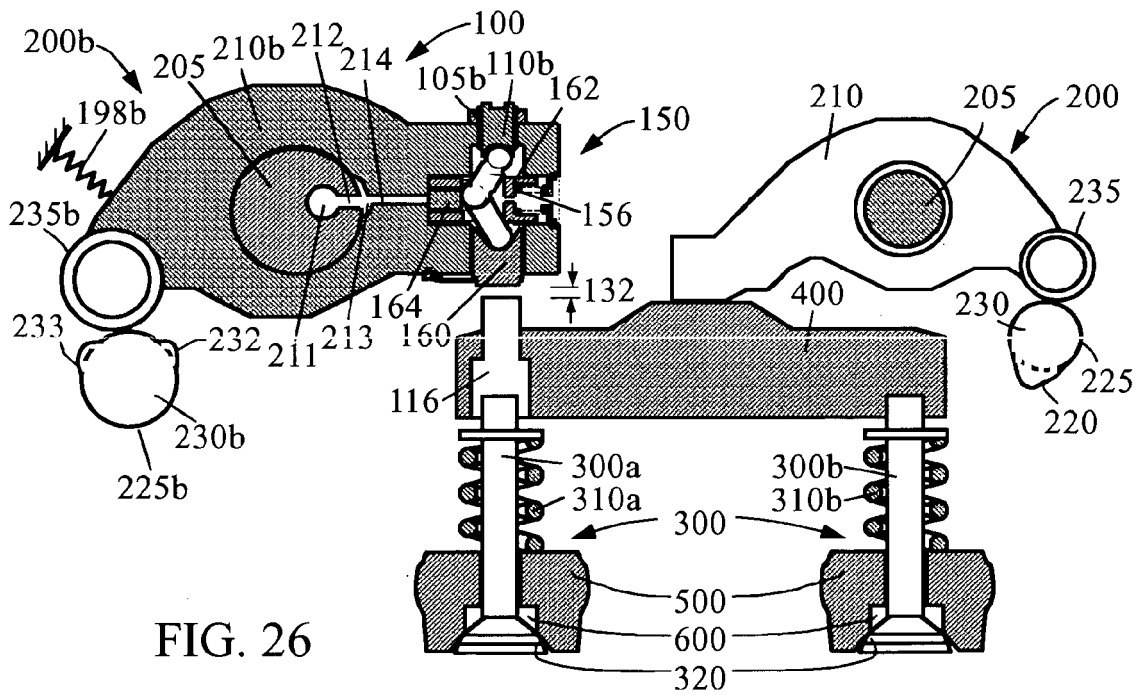


FIG. 26

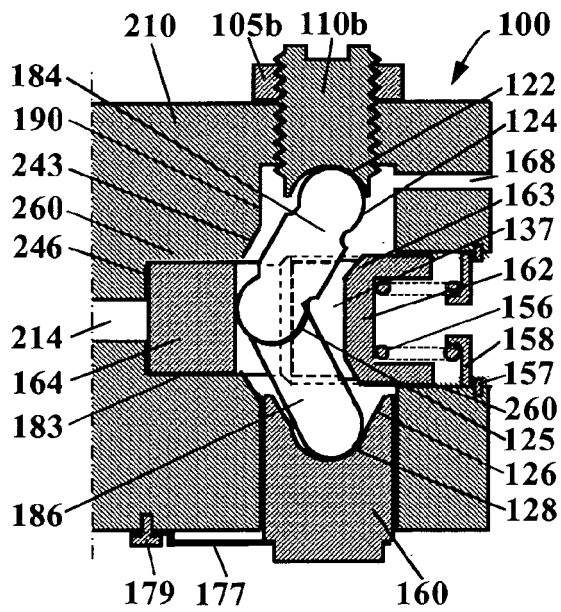


FIG. 27

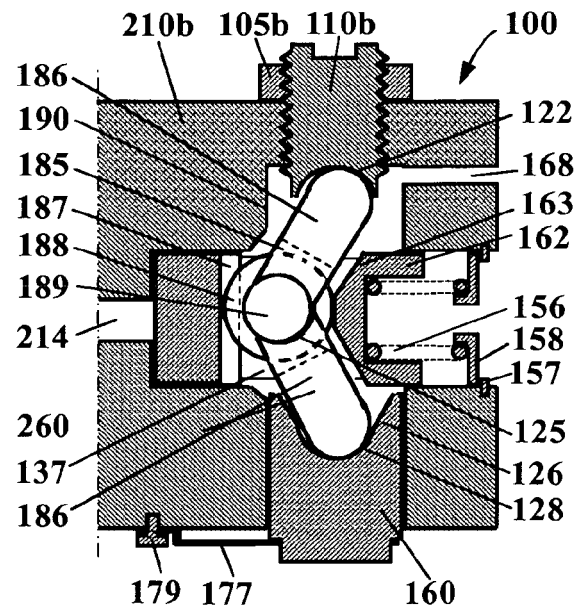


FIG. 28

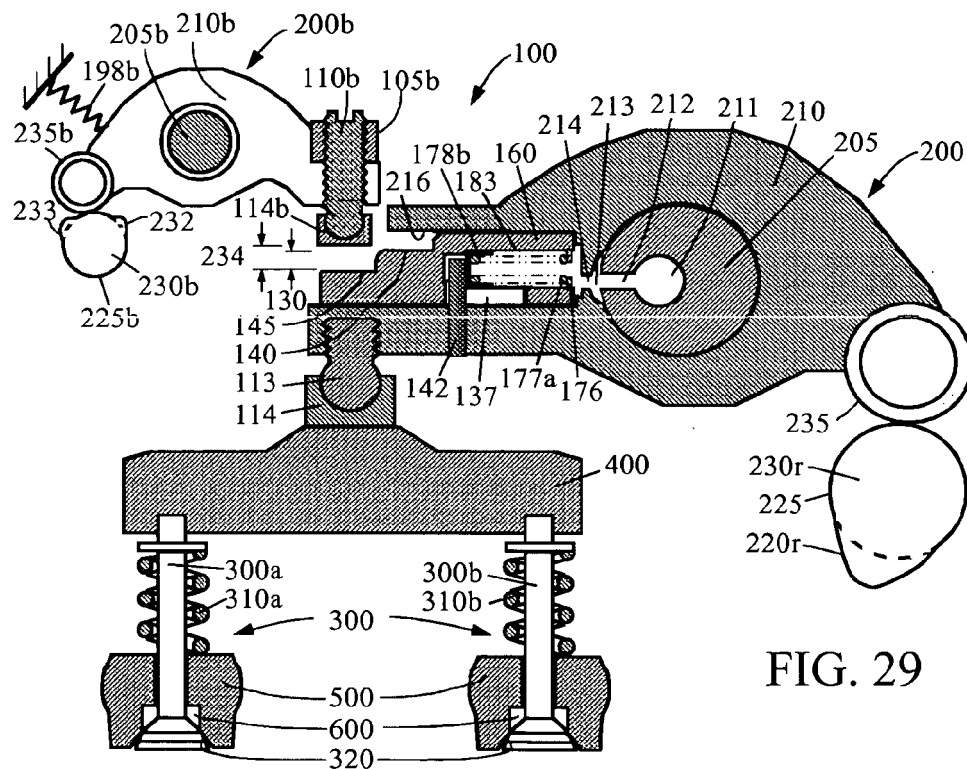


FIG. 29

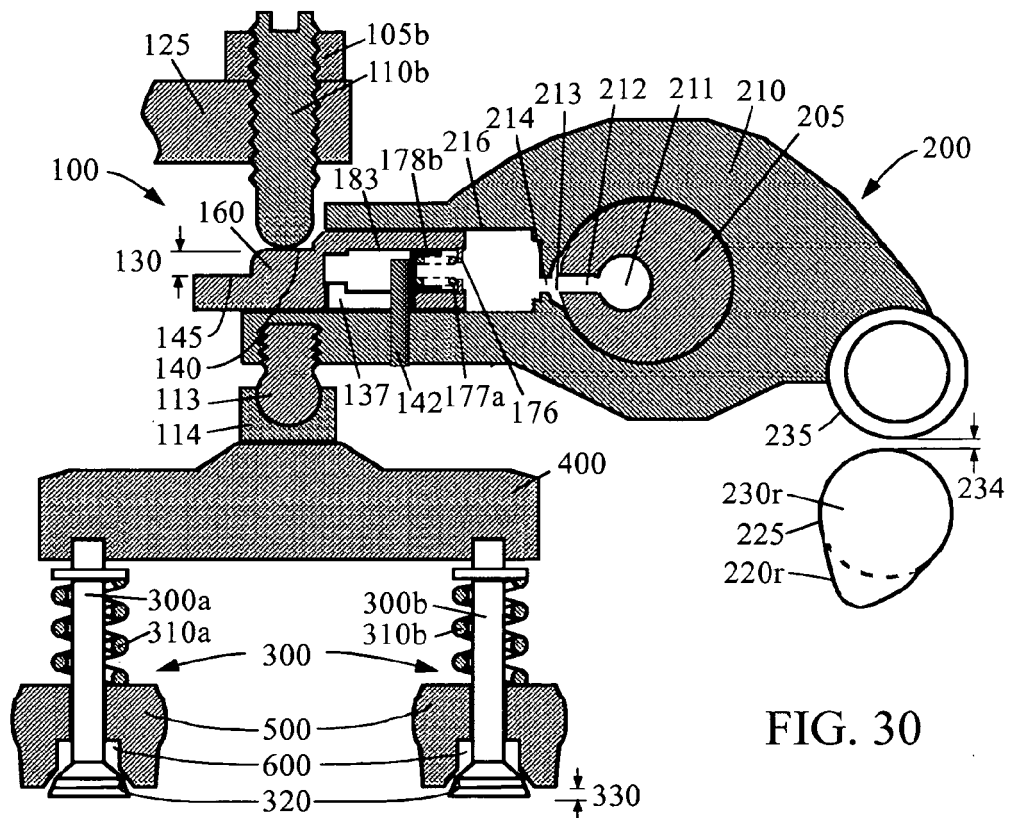


FIG. 30

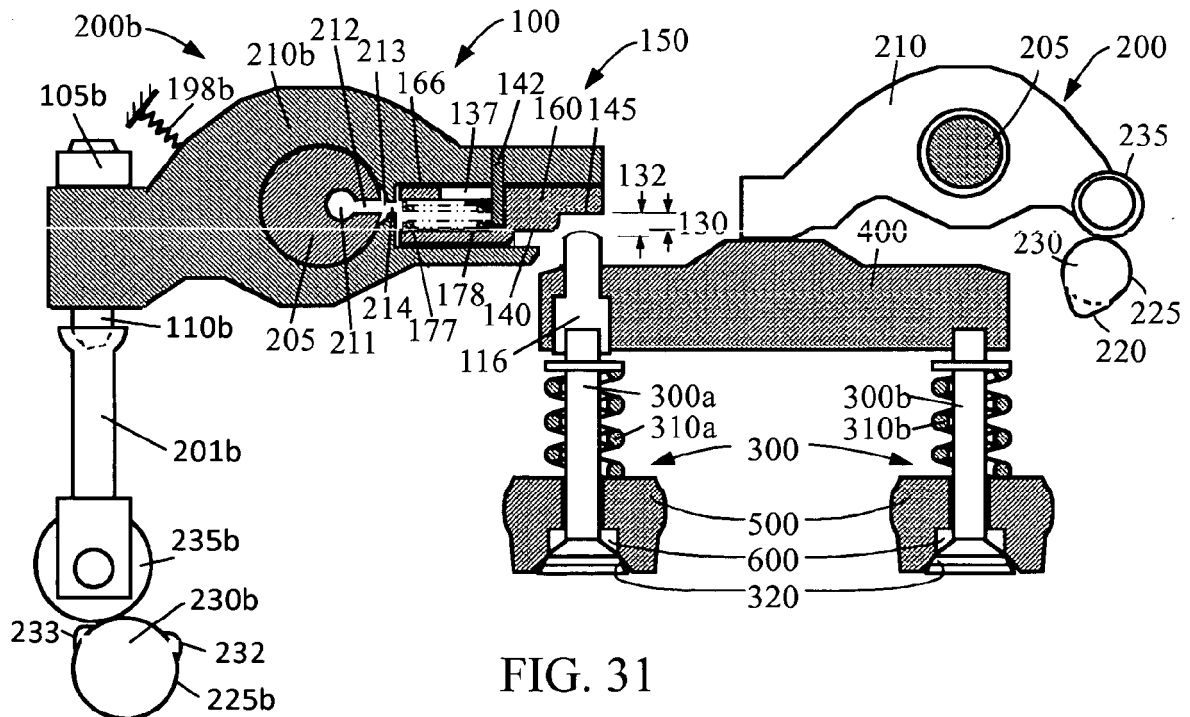


FIG. 31

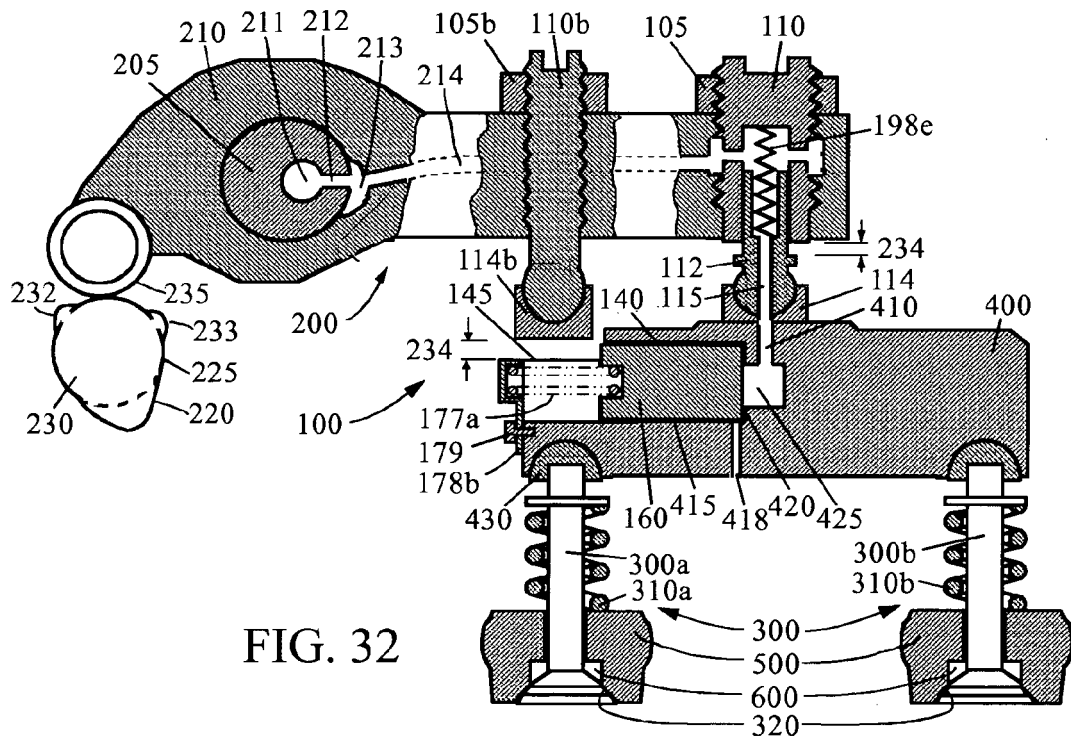


FIG. 32

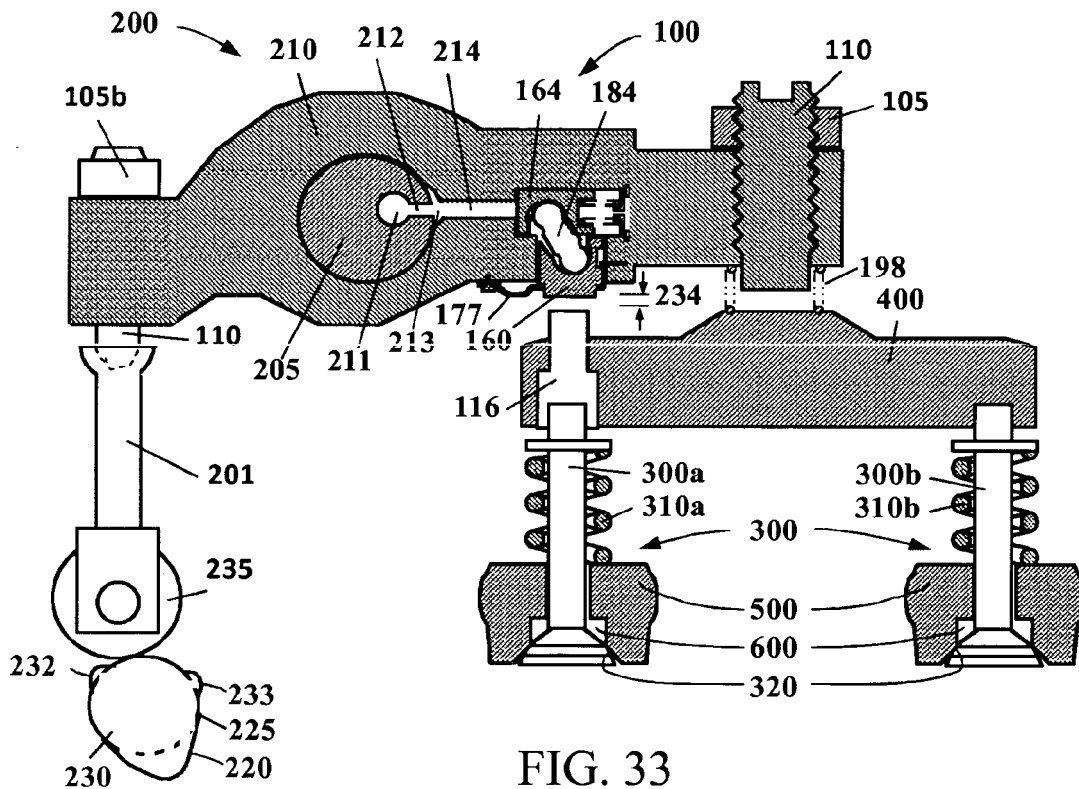


FIG. 33

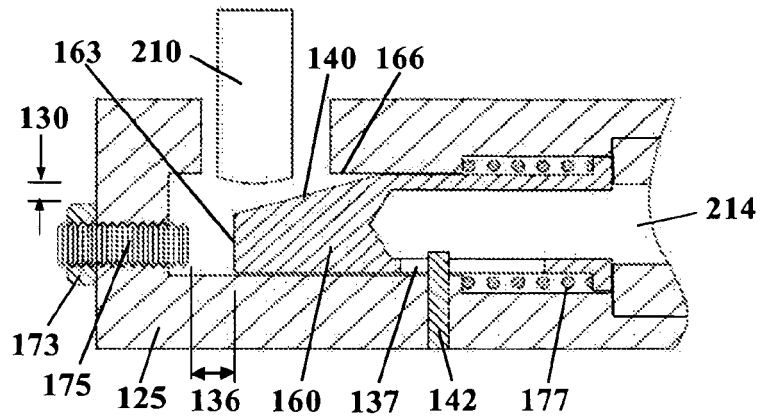


FIG. 34

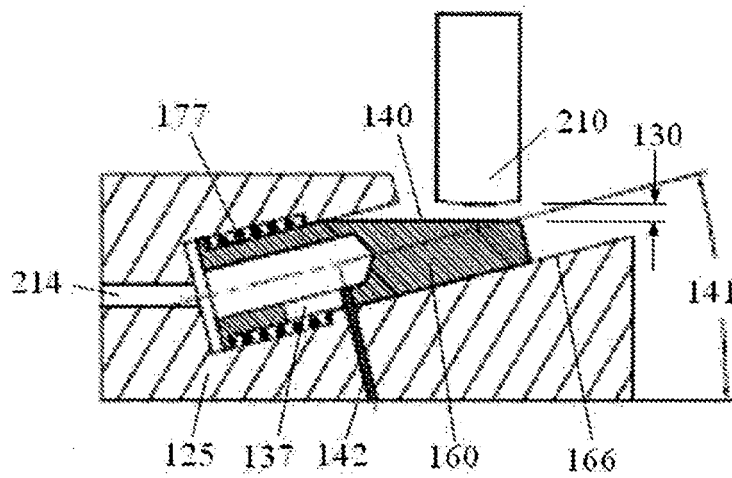


FIG. 35

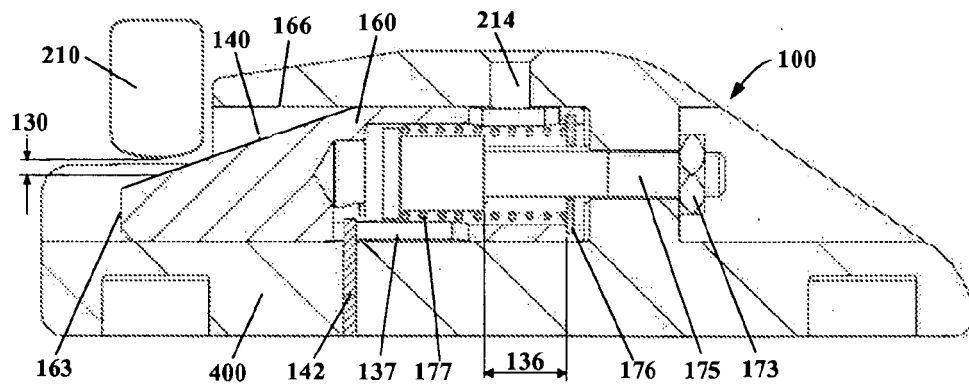


FIG. 36

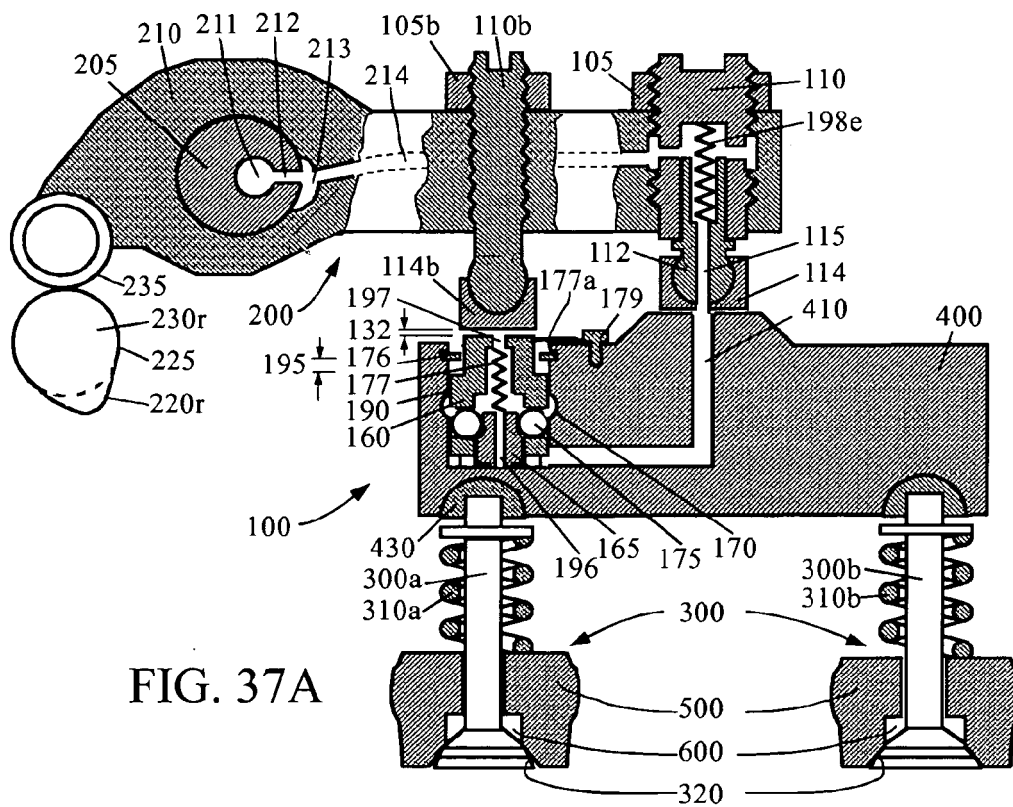


FIG. 37A

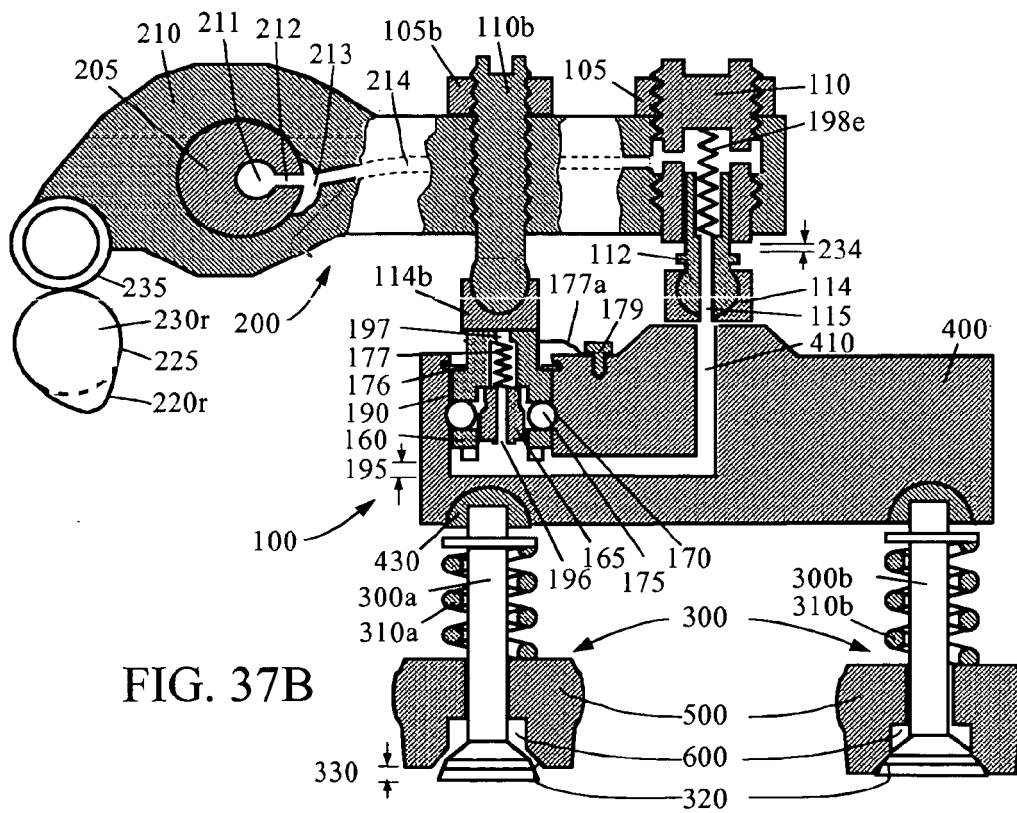


FIG. 37B

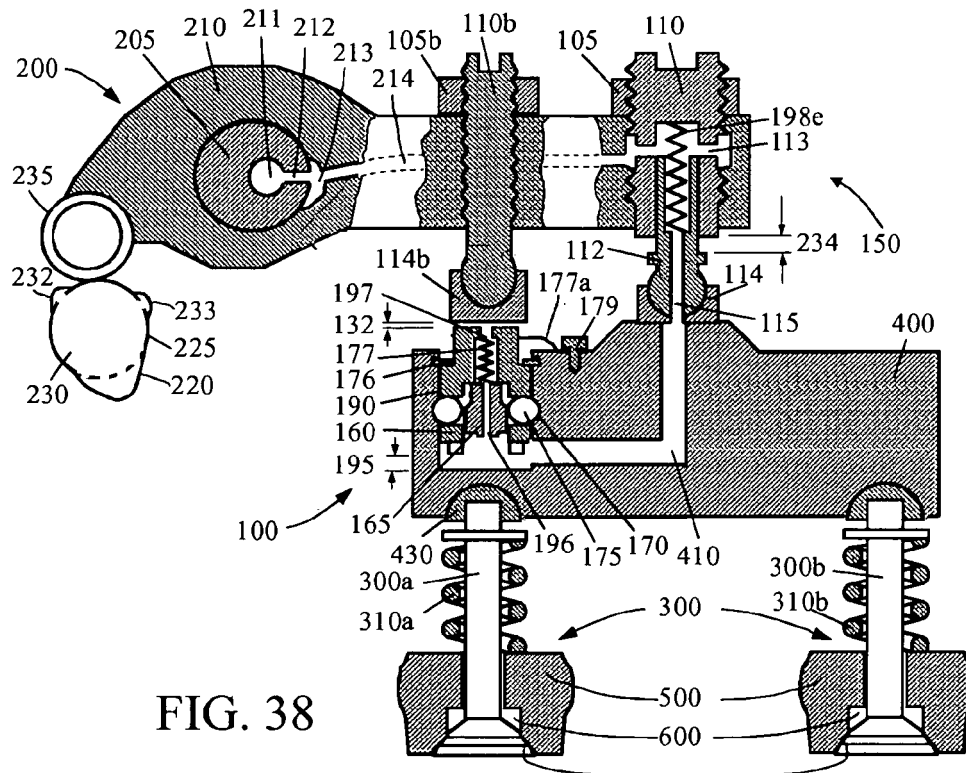


FIG. 38

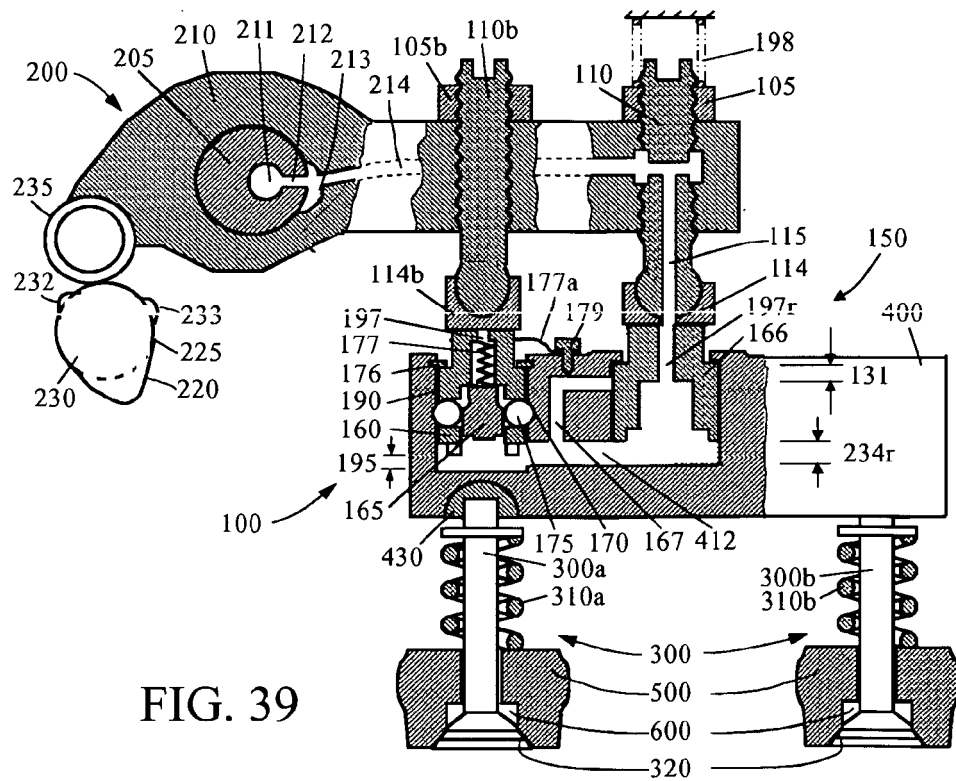


FIG. 39

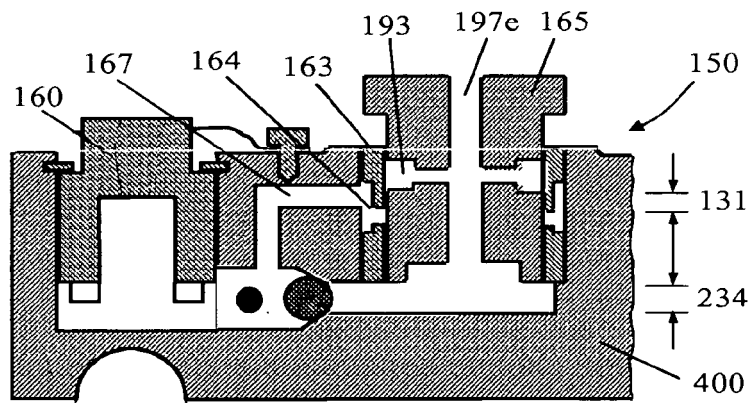
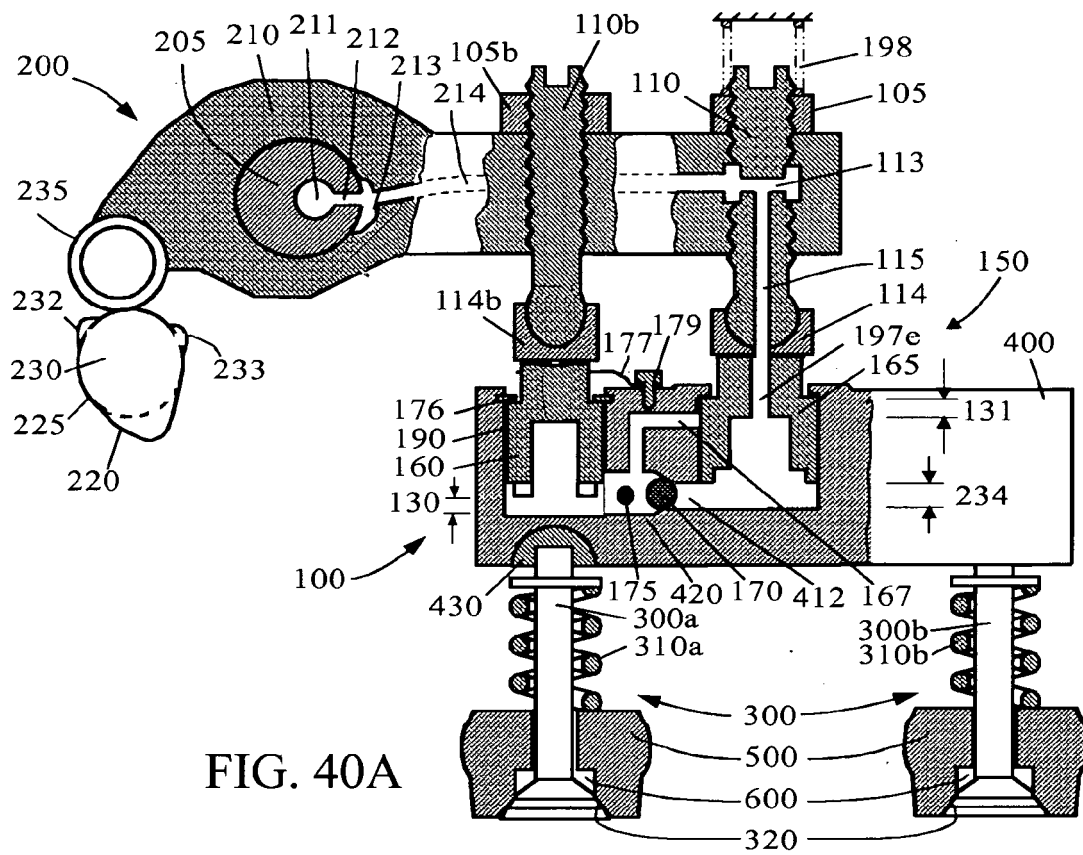


FIG. 40B



EUROPEAN SEARCH REPORT

Application Number
EP 11 00 6451

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
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X	US 2007/144472 A1 (YANG ZHOU [US]) 28 June 2007 (2007-06-28)	13	
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	* figures 12,13 *		

Y,D	US 6 253 730 B1 (GUSTAFSON RICHARD J [US]) 3 July 2001 (2001-07-03)	5,6	
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	* figures 8,9,10,11 *		

	-/--		
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 8 March 2012	Examiner Clot, Pierre
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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EPO FORM 1503 03.82 (P04C01)



EUROPEAN SEARCH REPORT

Application Number
EP 11 00 6451

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	US 2005/000499 A1 (RUGGIERO BRIAN [US] ET AL) 6 January 2005 (2005-01-06) * figure 10 *	1,10,13	
A	US 6 234 143 B1 (BARTEL JOHN B [US] ET AL) 22 May 2001 (2001-05-22) * the whole document *	1,10,13	
			TECHNICAL FIELDS SEARCHED (IPC)
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 8 March 2012	Examiner Clot, Pierre
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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EPO FORM 1503 03-82 (P04C01)



Application Number

EP 11 00 6451

CLAIMS INCURRING FEES

The present European patent application comprised at the time of filing claims for which payment was due.

☐ Only part of the claims have been paid within the prescribed time limit. The present European search report has been drawn up for those claims for which no payment was due and for those claims for which claims fees have been paid, namely claim(s):

☐ No claims fees have been paid within the prescribed time limit. The present European search report has been drawn up for those claims for which no payment was due.

LACK OF UNITY OF INVENTION

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

see sheet B

☐ All further search fees have been paid within the fixed time limit. The present European search report has been drawn up for all claims.

☒ As all searchable claims could be searched without effort justifying an additional fee, the Search Division did not invite payment of any additional fee.

☐ Only part of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the inventions in respect of which search fees have been paid, namely claims:

☐ None of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims, namely claims:

☐ The present supplementary European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims (Rule 164 (1) EPC).



**LACK OF UNITY OF INVENTION
SHEET B**

Application Number

EP 11 00 6451

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

1. claims: 1-12

Apparatus for converting an ICE from normal engine operation to an engine braking operation and Method of modifying engine valve lift in an ICE to produce an auxiliary engine valve event that is different from a normal engine operation, whereby in an inoperative position, a braking piston, resp. a hydraulic piston which is slidably disposed in a valve bridge is retracted and disengaged from the normal engine operation

2. claims: 13-15

Method of modifying engine valve lift in an ICE to produce an auxiliary engine valve event that is different from a normal engine operation, whereby a valve lash adjusting device is integrated into an engine valve lifter

**ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.**

EP 11 00 6451

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
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08-03-2012

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EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

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- US 3367312 A, Jonson [0009]
- US 5564385 A [0011]
- US 6234143 B [0016]
- US 6253730 B [0018]
- US 20050211206 A [0088]