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(54) VALVE ACTIVATION IN COMPRESSED-GAS ENERGY STORAGE AND RECOVERY SYSTEMS

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(21) Appl. No.: **13/715,039**

(22) Filed: **Dec. 14, 2012**

Related U.S. Application Data

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Publication Classification

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F01B 29/00 (2006.01)

(52) **U.S. Cl.**
CPC ***F01B 29/00*** (2013.01)
USPC **60/327**

(57) **ABSTRACT**

In various embodiments, valve efficiency and reliability are enhanced via use of hydraulic or magnetic valve actuation, valves configured for increased actuation speed, and/or valves controlled to reduce collision forces during actuation.

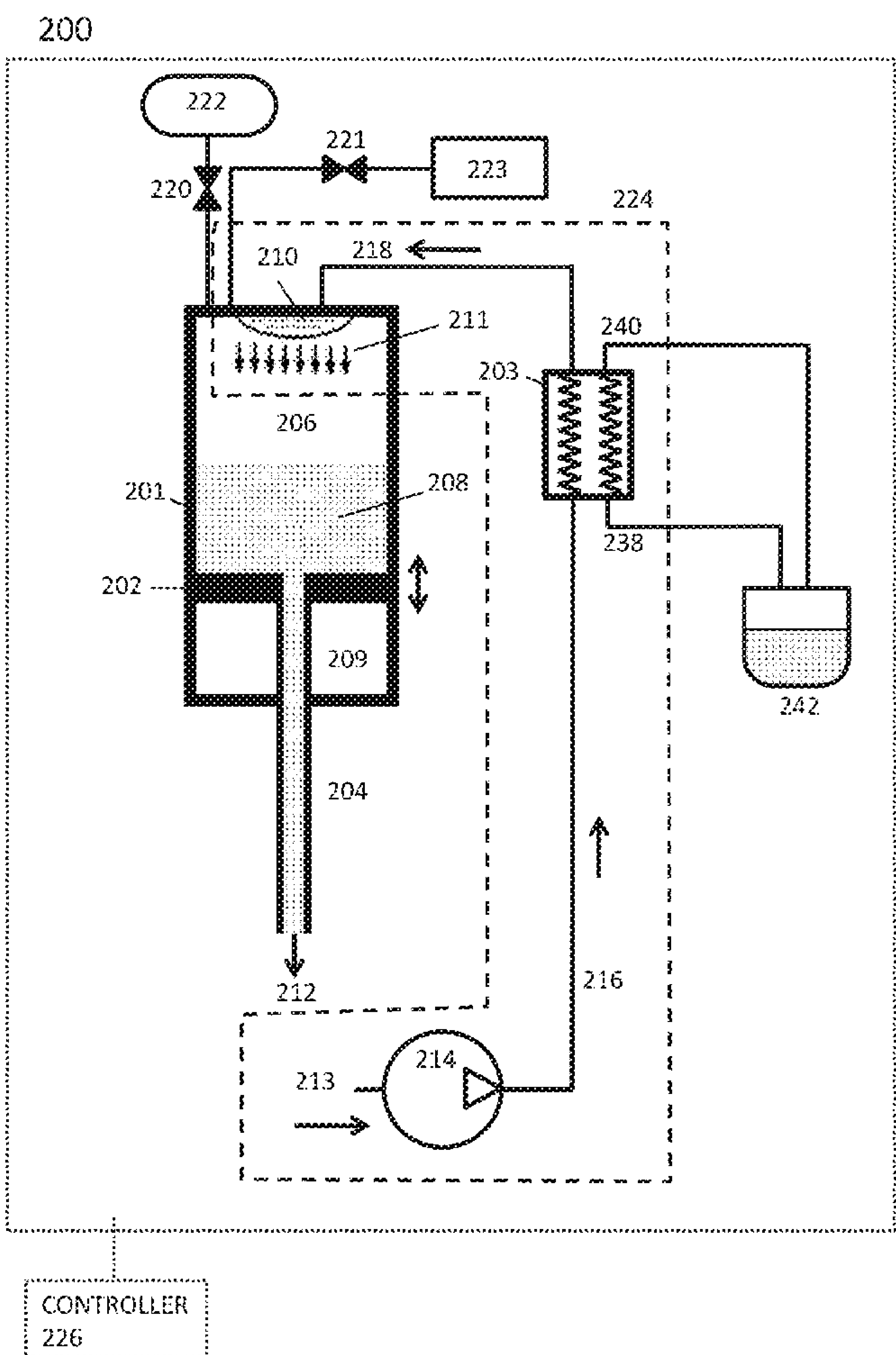


FIG. 1

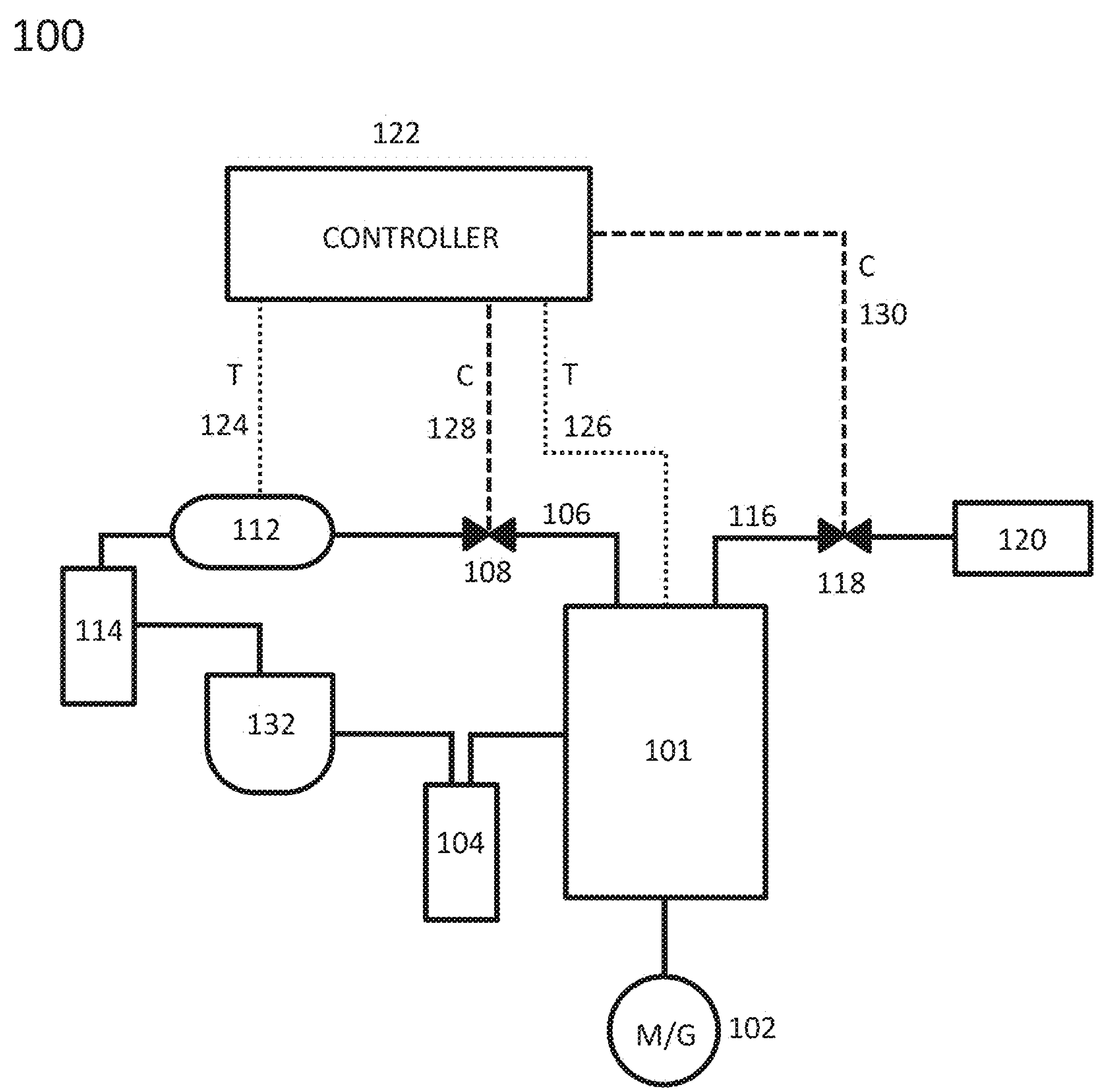
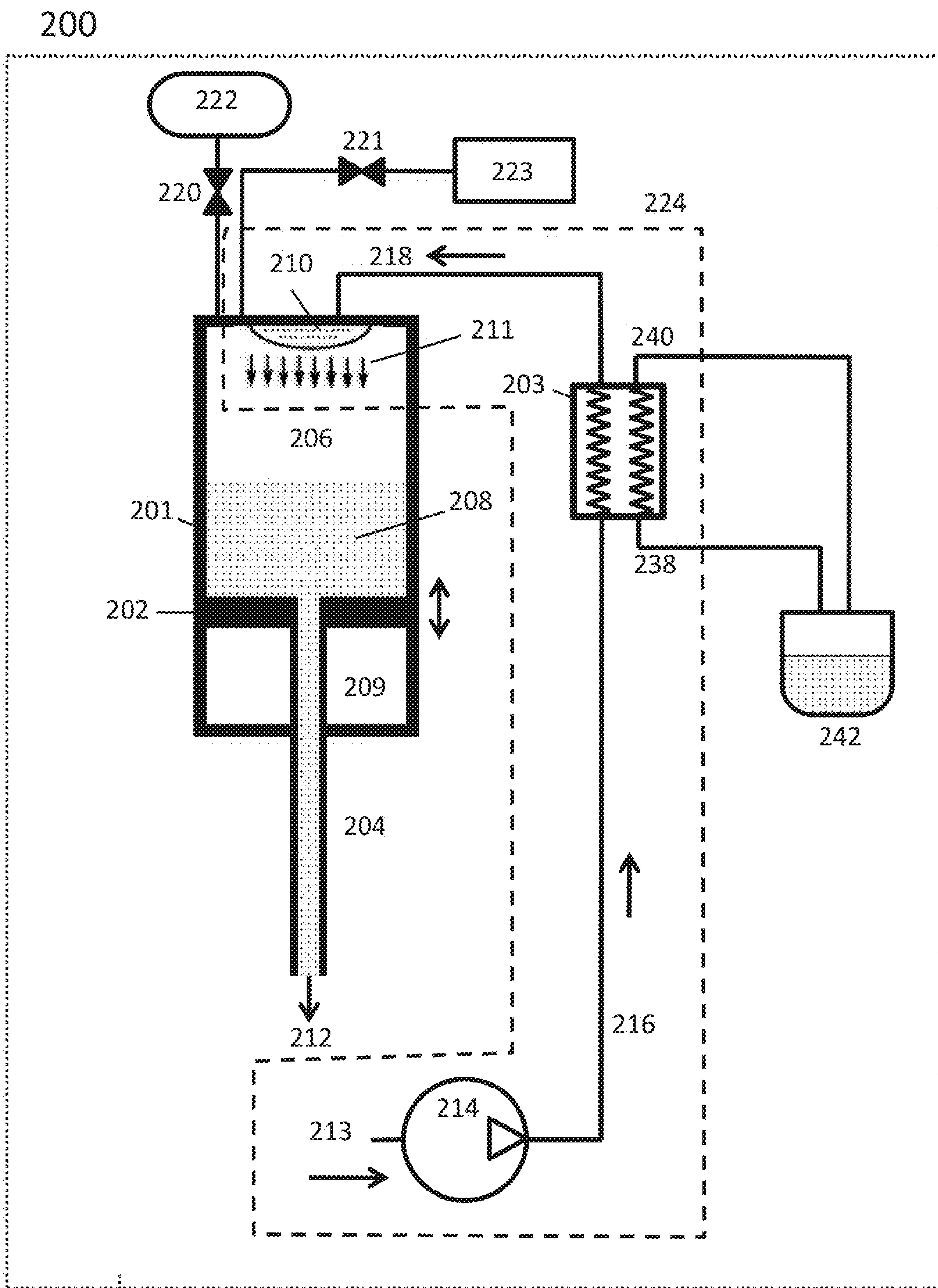

100
122
CONTROLLER
T
124
C
128
T
126
C
130
112
108
106
116
118
120
114
132
104
101
M/G
102

FIG. 2



CONTROLLER
226

FIG. 3

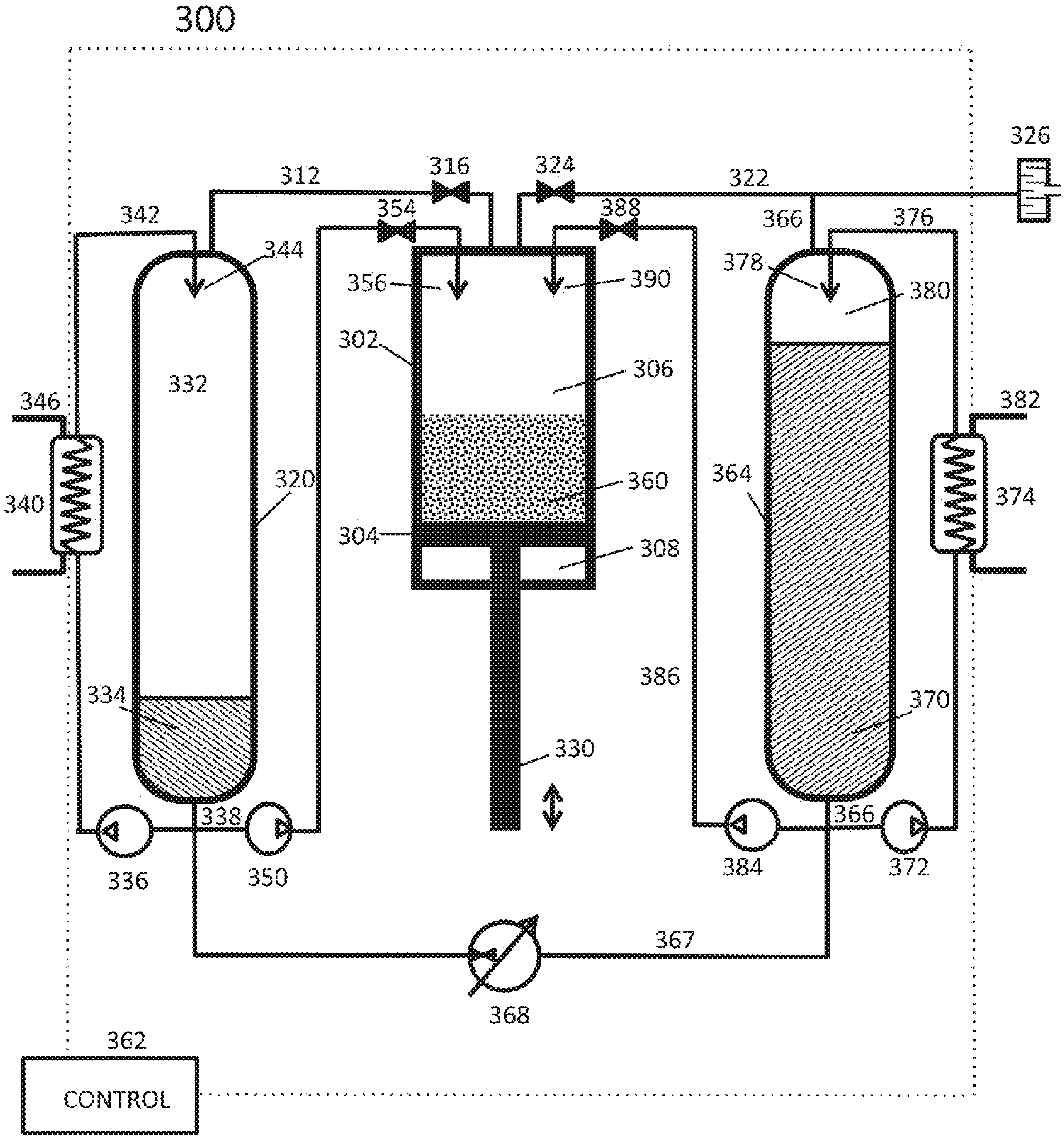

300
326
312
316
324
322
342
354
388
366
376
344
356
390
378
380
302
332
306
346
382
340
320
360
364
374
304
308
386
334
370
330
338
366
336
350
384
372
367
368
362
CONTROL

FIG. 4

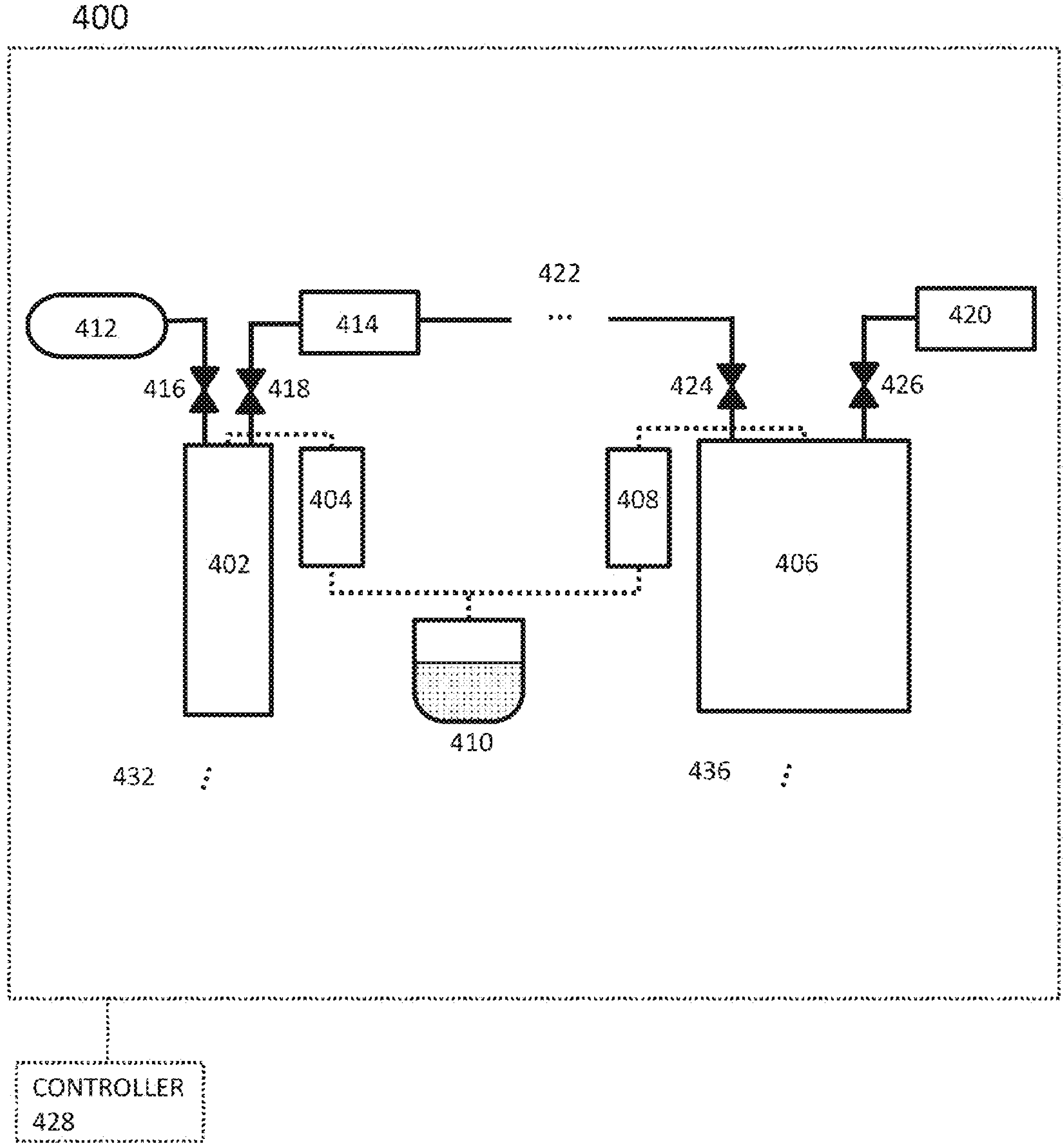

400
412
414
422
...
420
416
418
424
426
404
408
402
406
410
432
436
CONTROLLER
428

FIG. 5

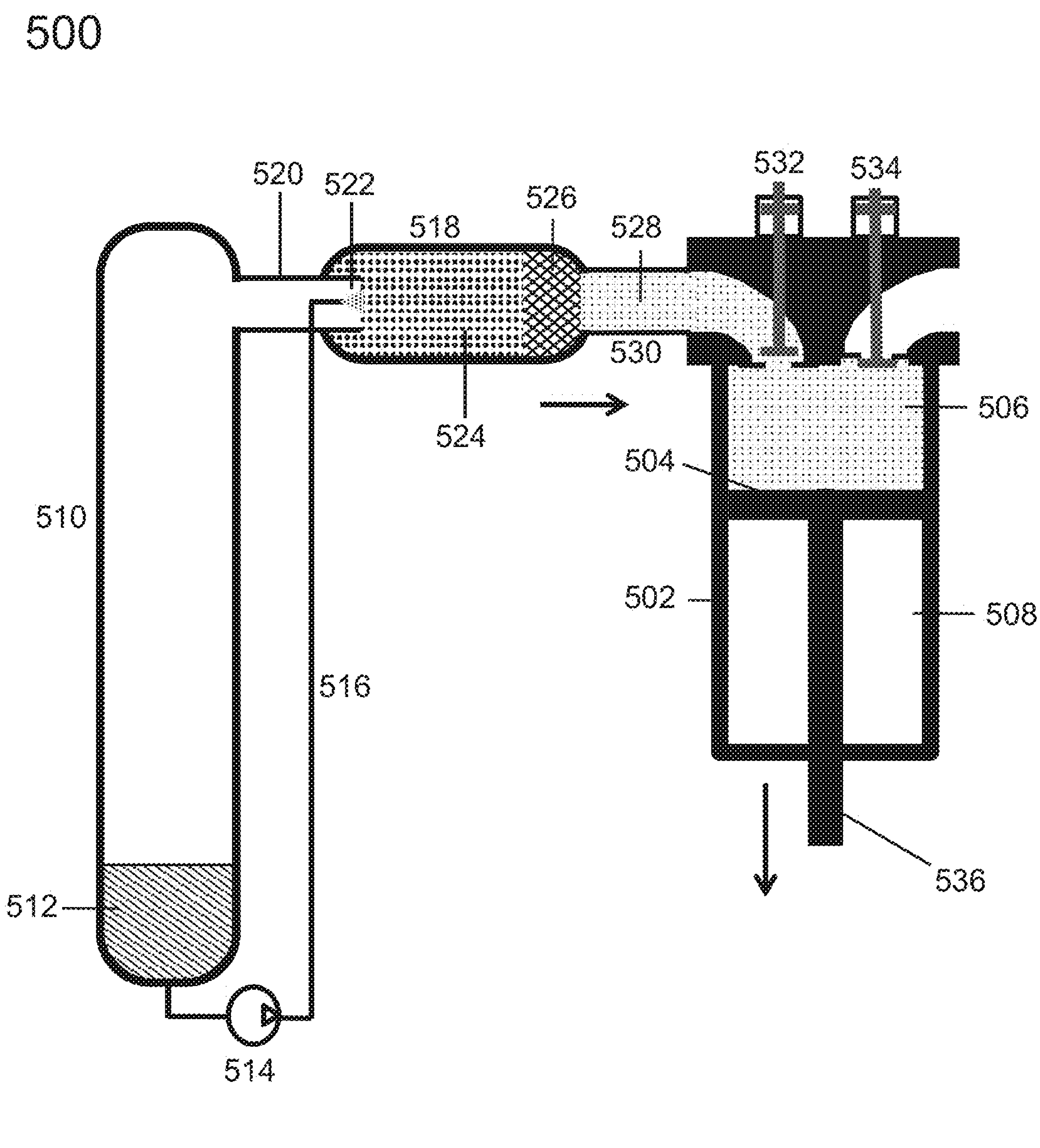

500
520
522
518
526
528
532
534
530
524
506
504
510
502
508
516
512
514
536

FIG. 6

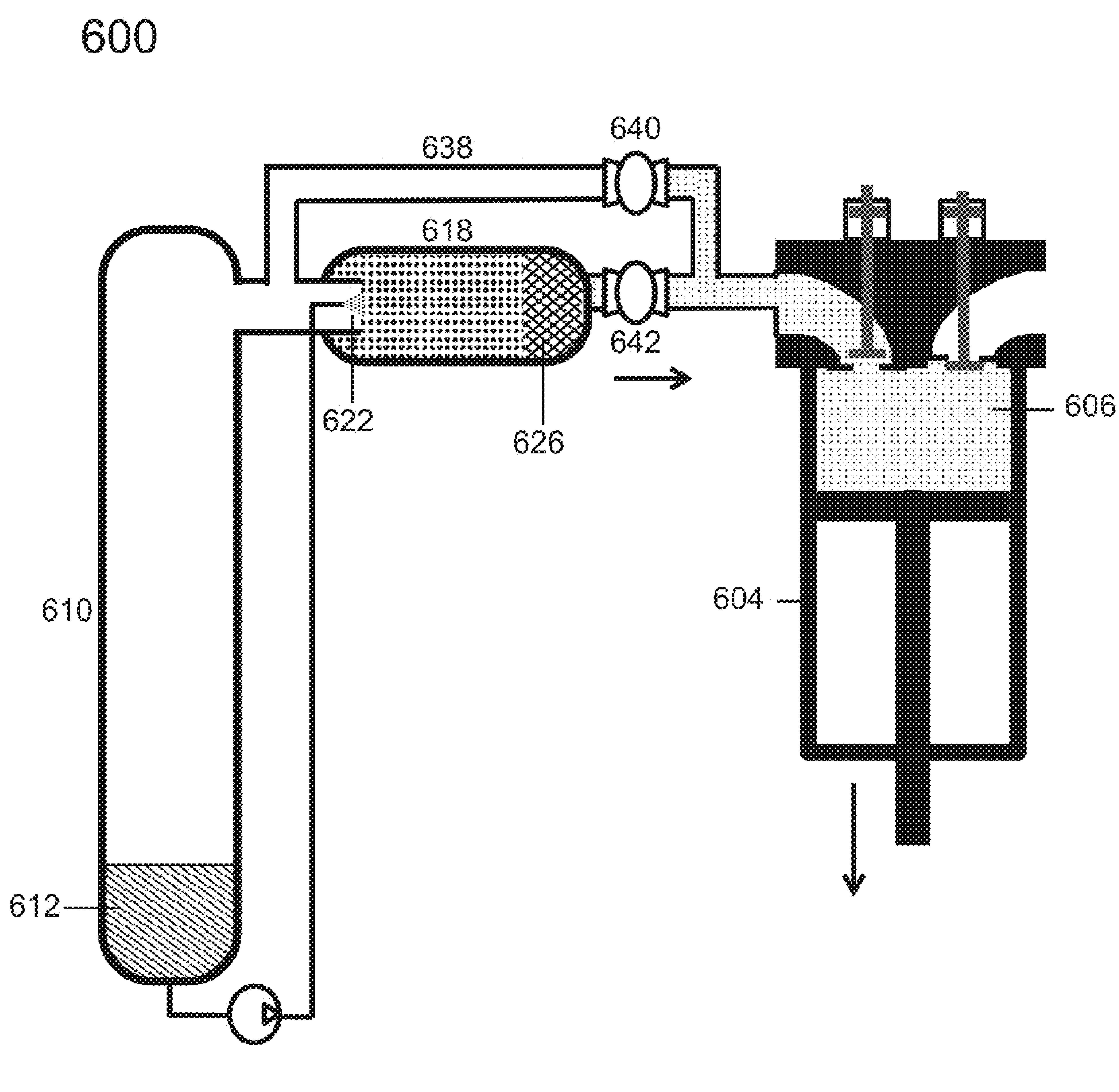


FIG. 7

700

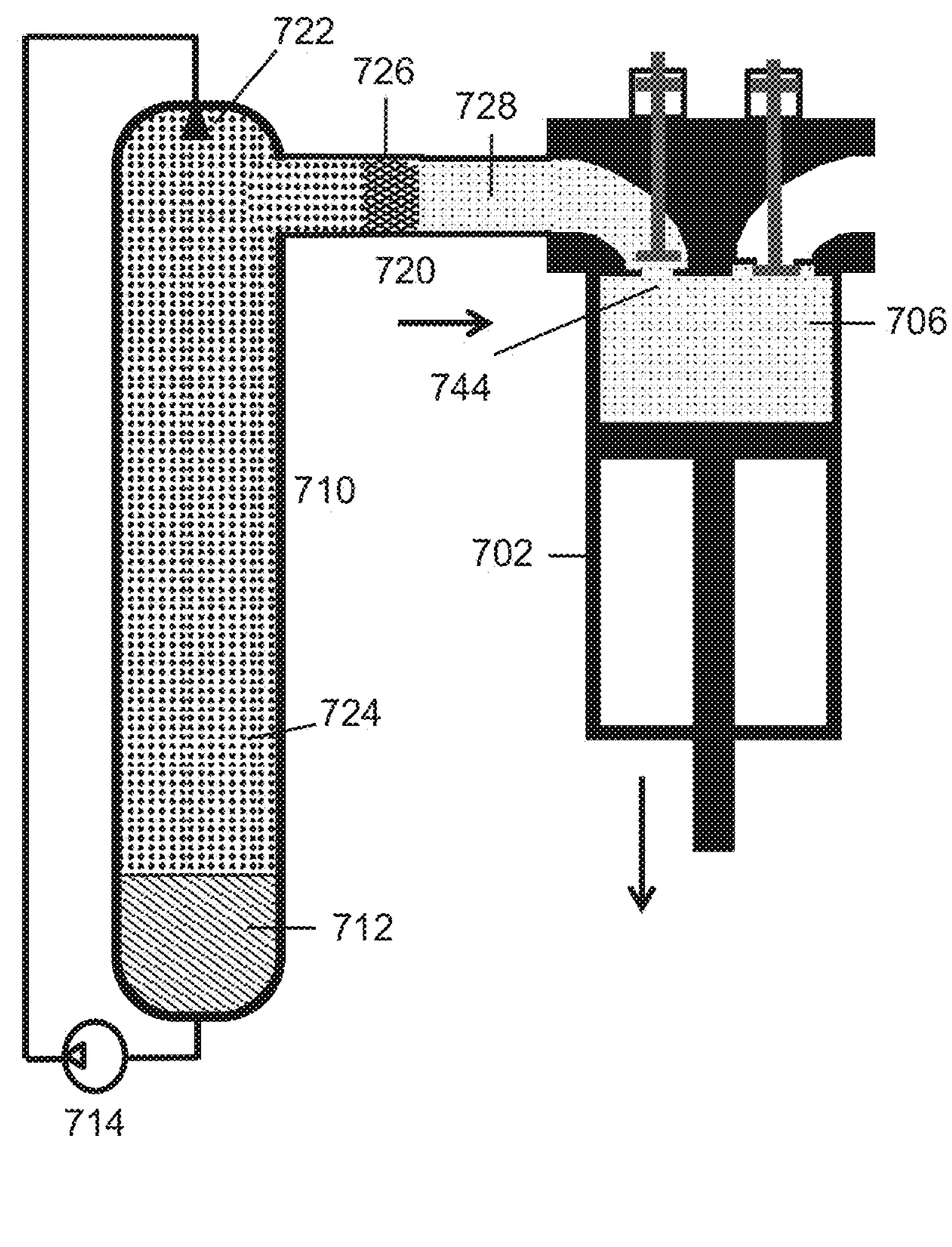


FIG. 8

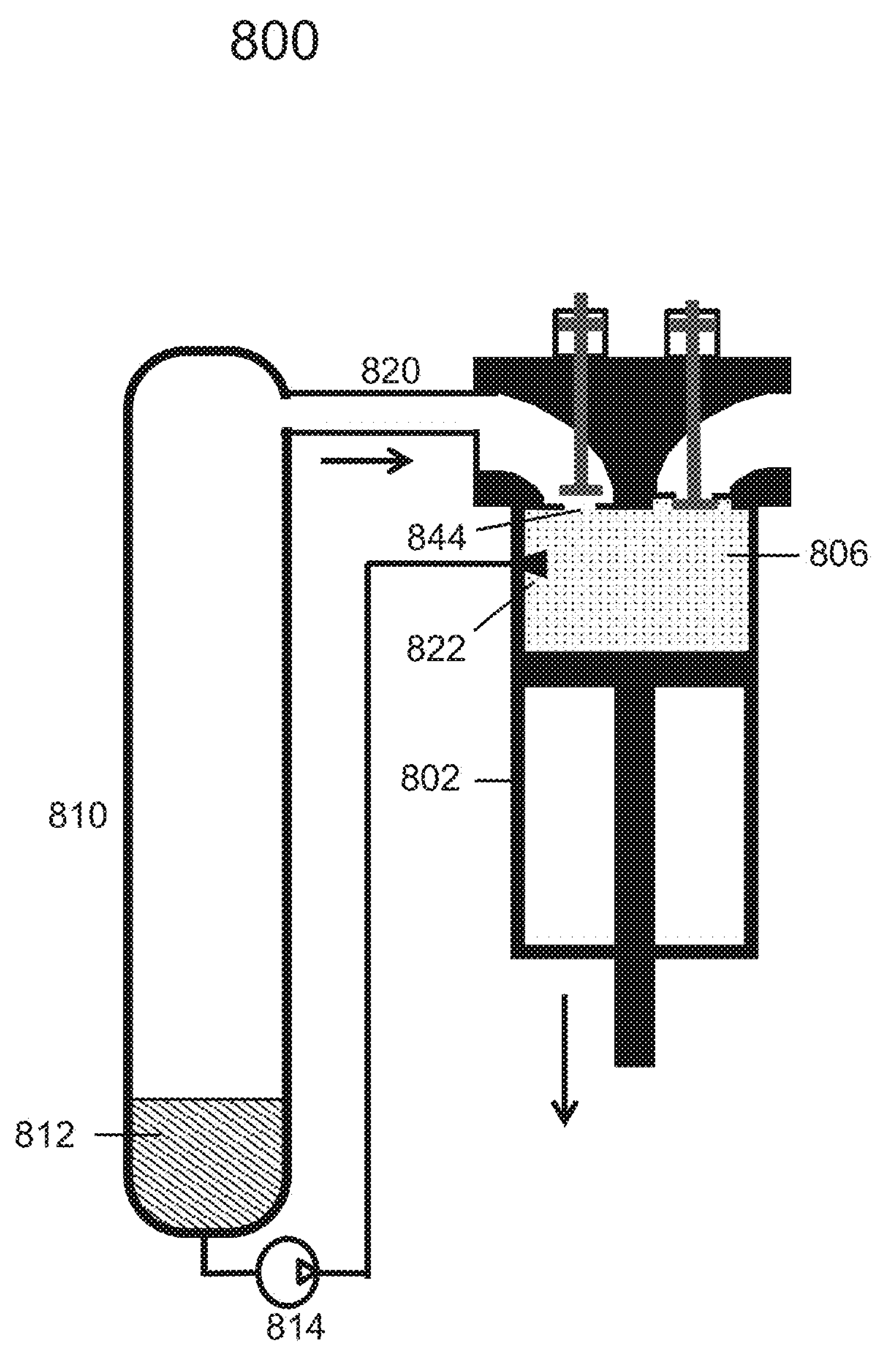


FIG. 9

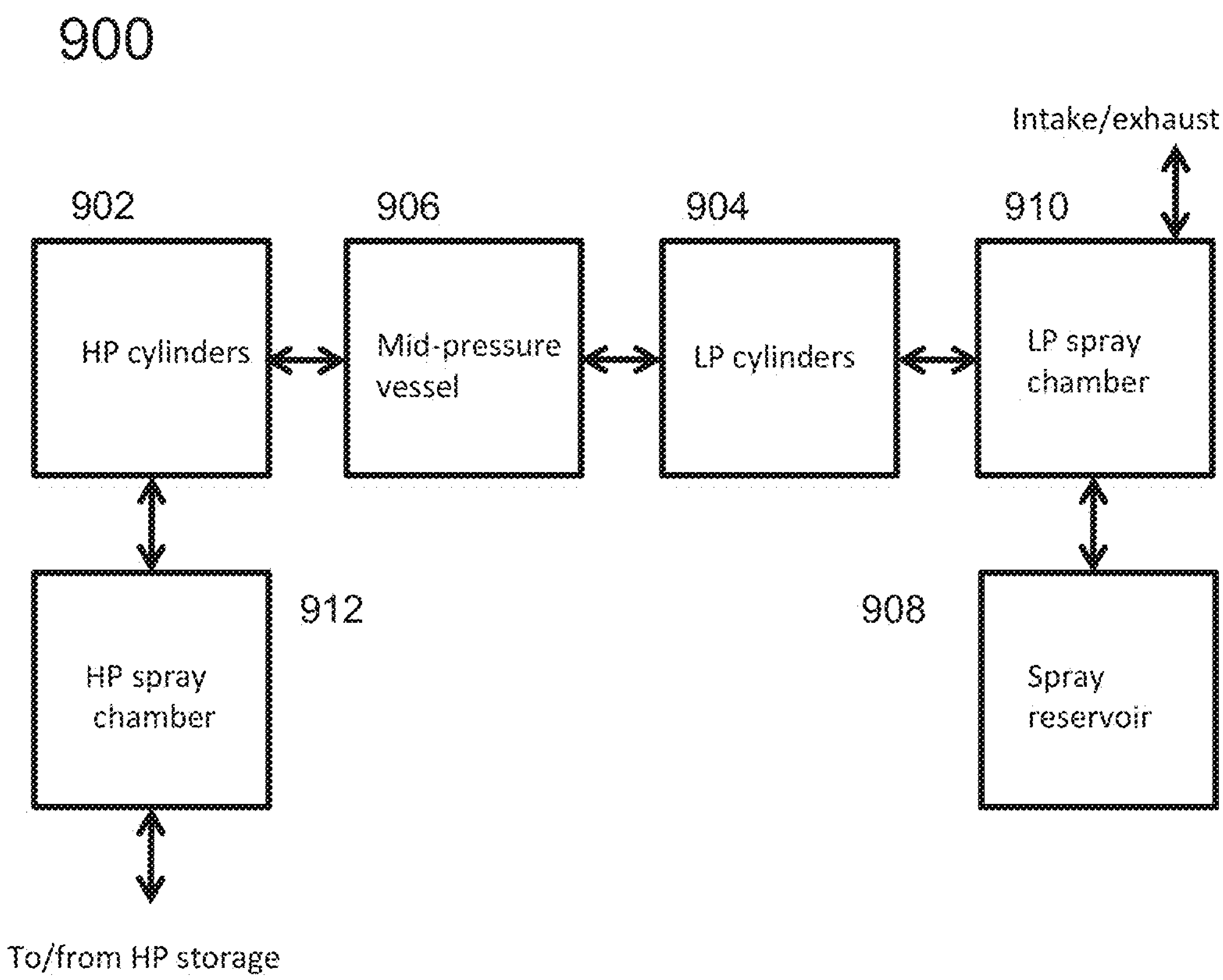

900
Intake/exhaust
902
906
904
910
HP cylinders
Mid-pressure vessel
LP cylinders
LP spray chamber
912
908
HP spray chamber
Spray reservoir
To/from HP storage

FIG. 10

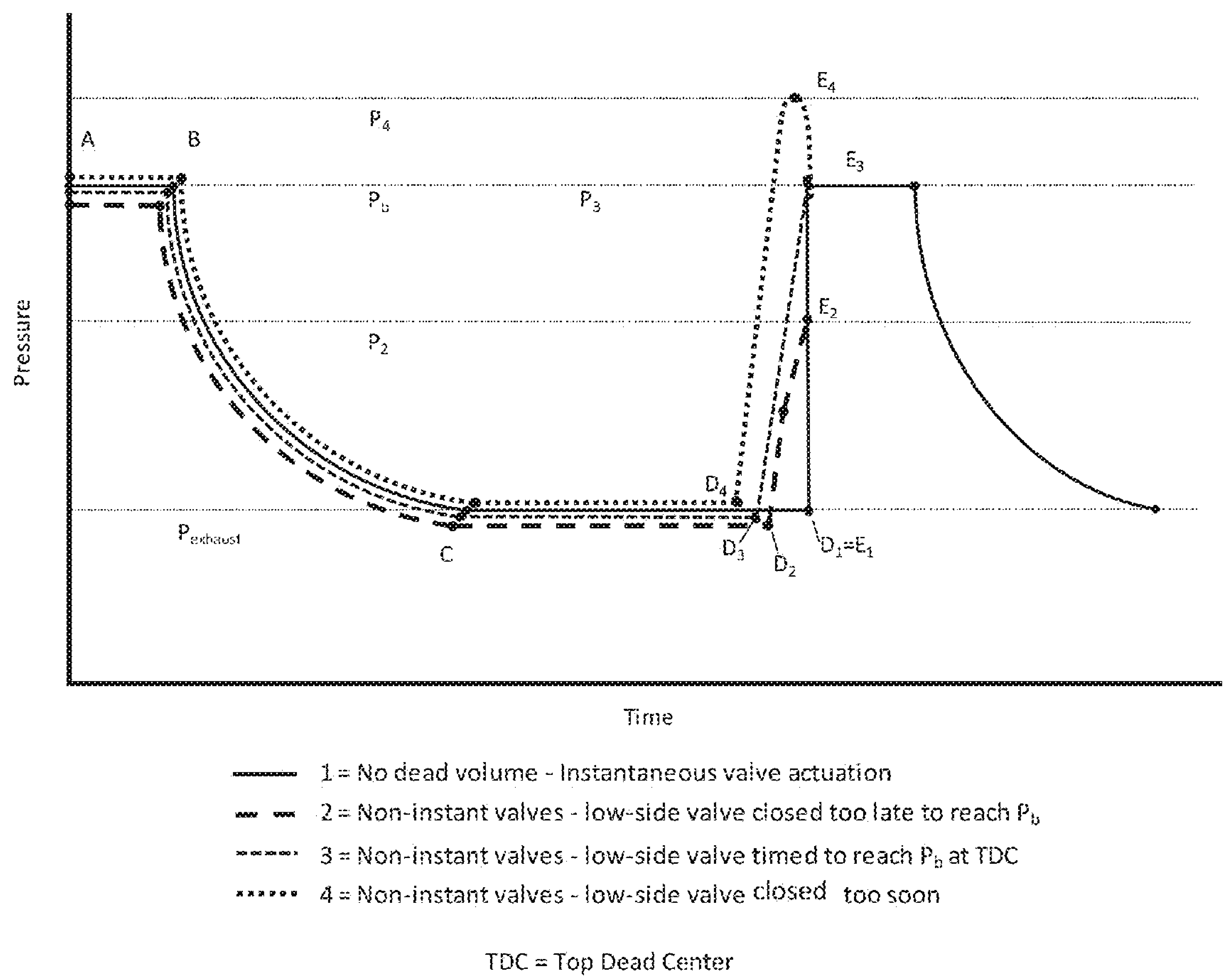
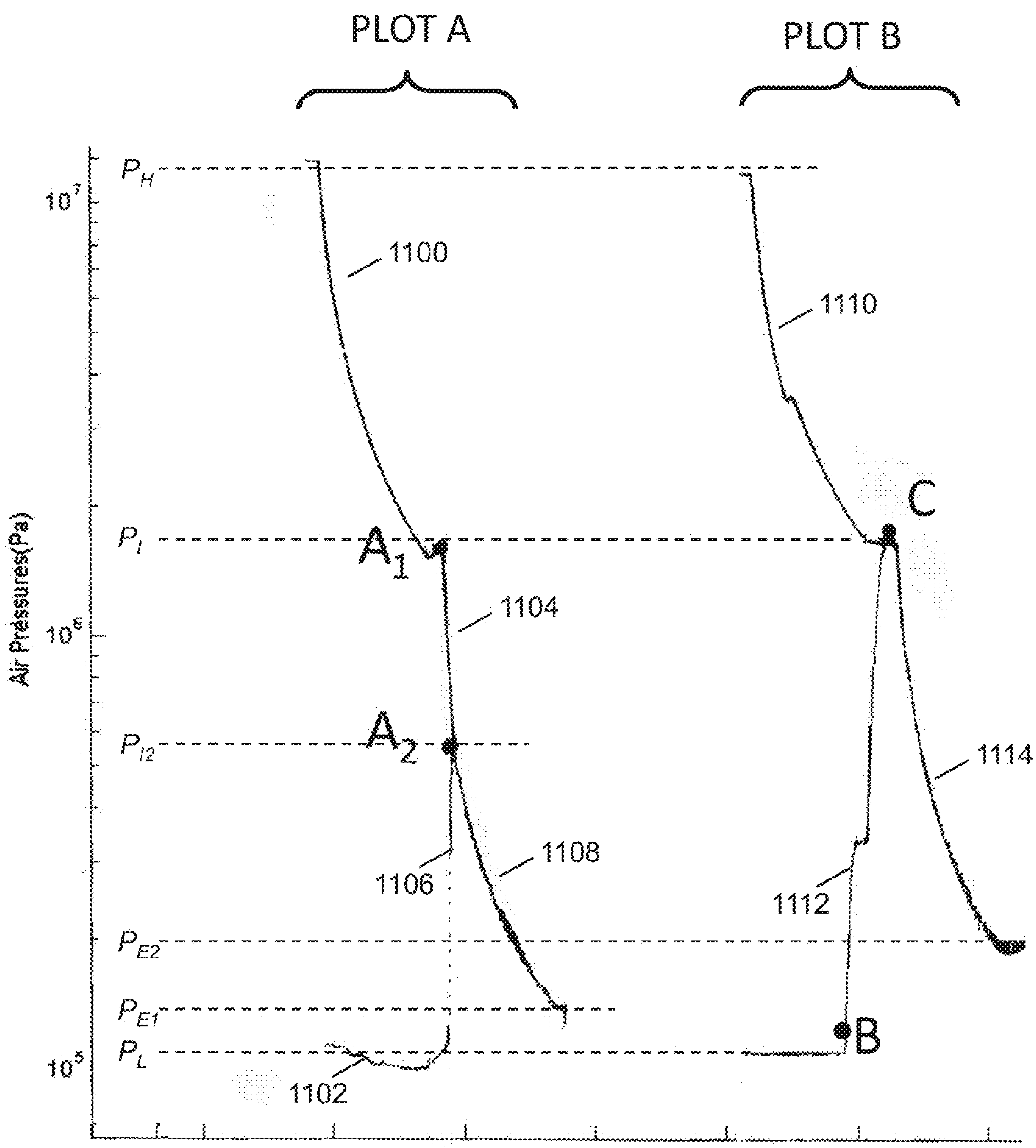

E4
P4
A
B
E3
Pb
P3
Pressure
E2
P2
D4
Pexhaust
C
D3
D2
D1=E1
Time
1 = No dead volume - Instantaneous valve actuation
2 = Non-instant valves - low-side valve closed too late to reach Pb
3 = Non-instant valves - low-side valve timed to reach Pb at TDC
4 = Non-instant valves - low-side valve closed too soon
TDC = Top Dead Center

FIG. 11



PLOT A
PLOT B
Air Pressures(Pa)
P_H
10^7
1100
1110
P_I
A_1
C
1104
10^6
P_{I2}
A_2
1114
1106
1108
1112
P_{E2}
P_{E1}
P_L
B
10^5
1102

FIG. 12

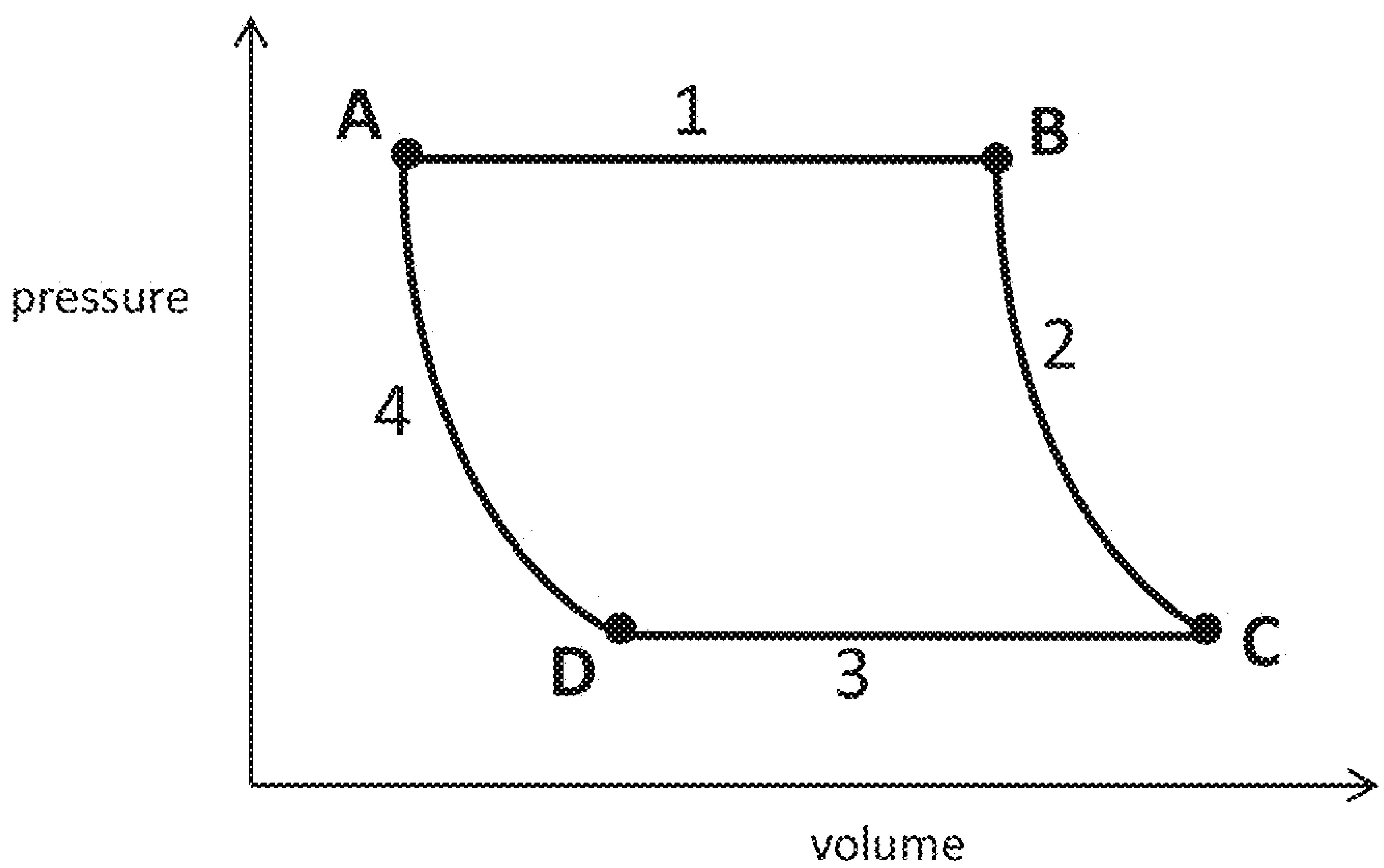


FIG. 13

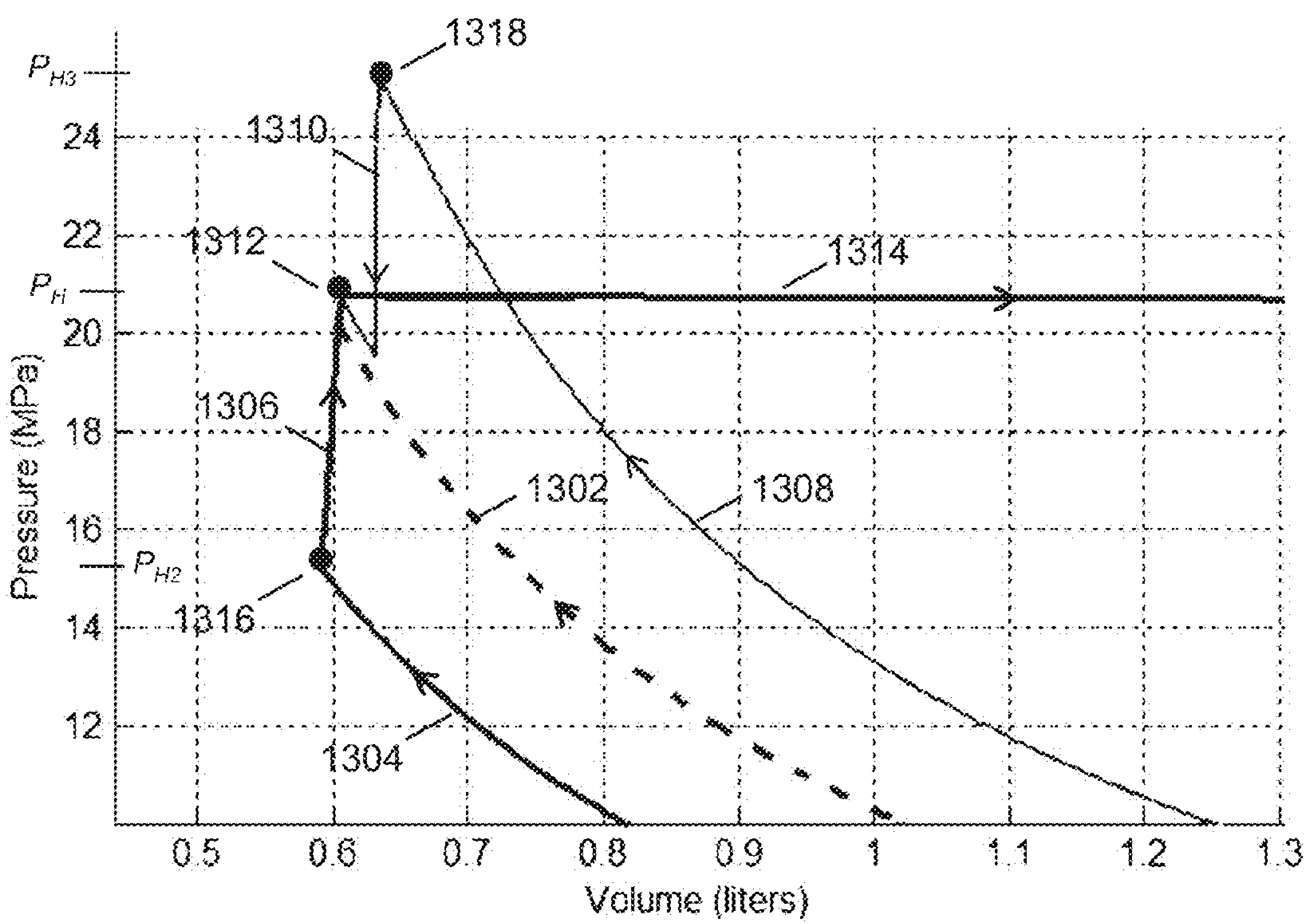

1318
P_{H3}
24
1310
22
1312
1314
P_{H}
20
1306
18
Pressure (MPa)
1302
1308
16
P_{H2}
14
1316
12
1304
0.5
0.6
0.7
0.8
0.9
1
1.1
1.2
1.3
Volume (liters)

FIG. 14A

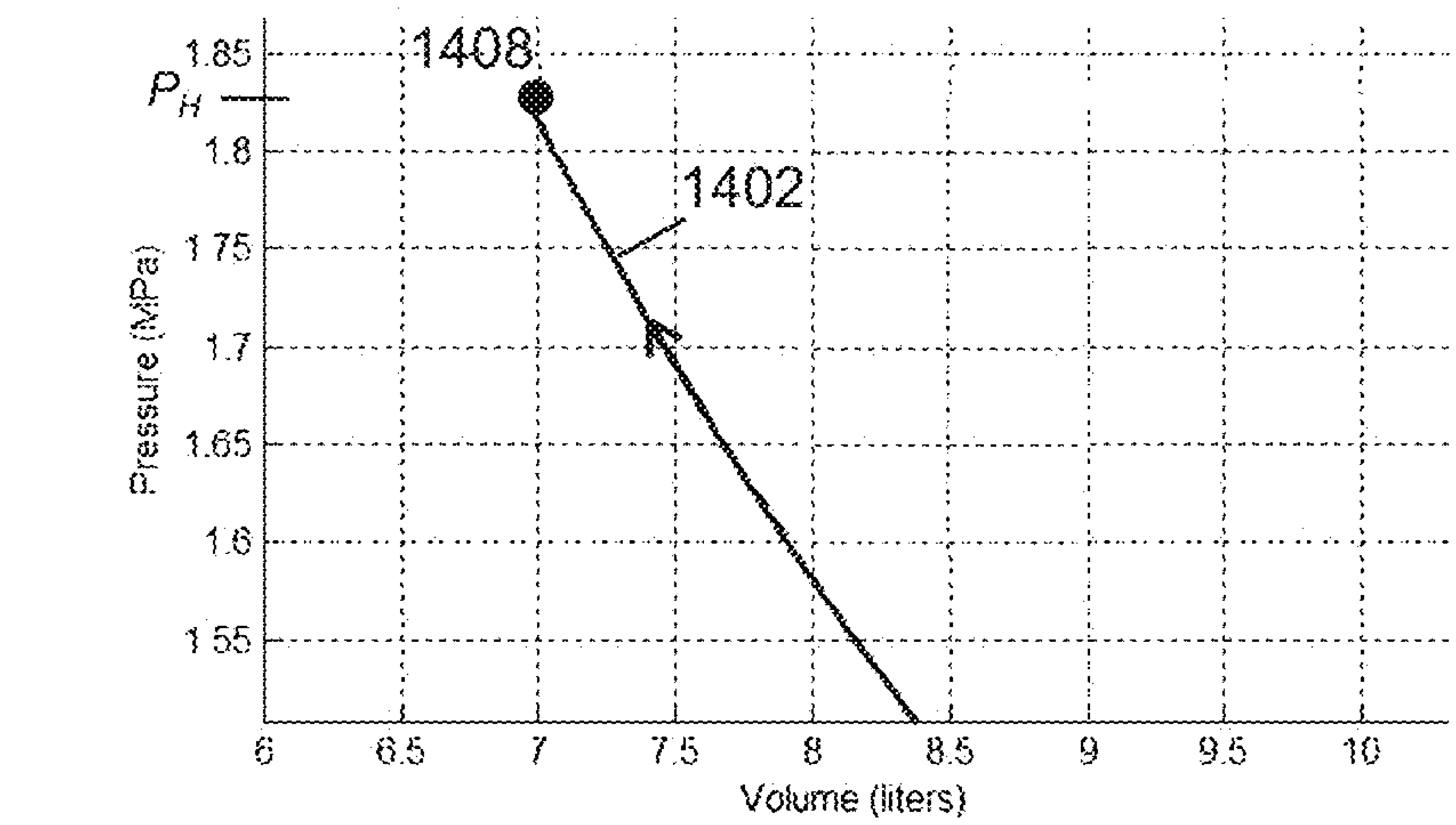


FIG. 14B

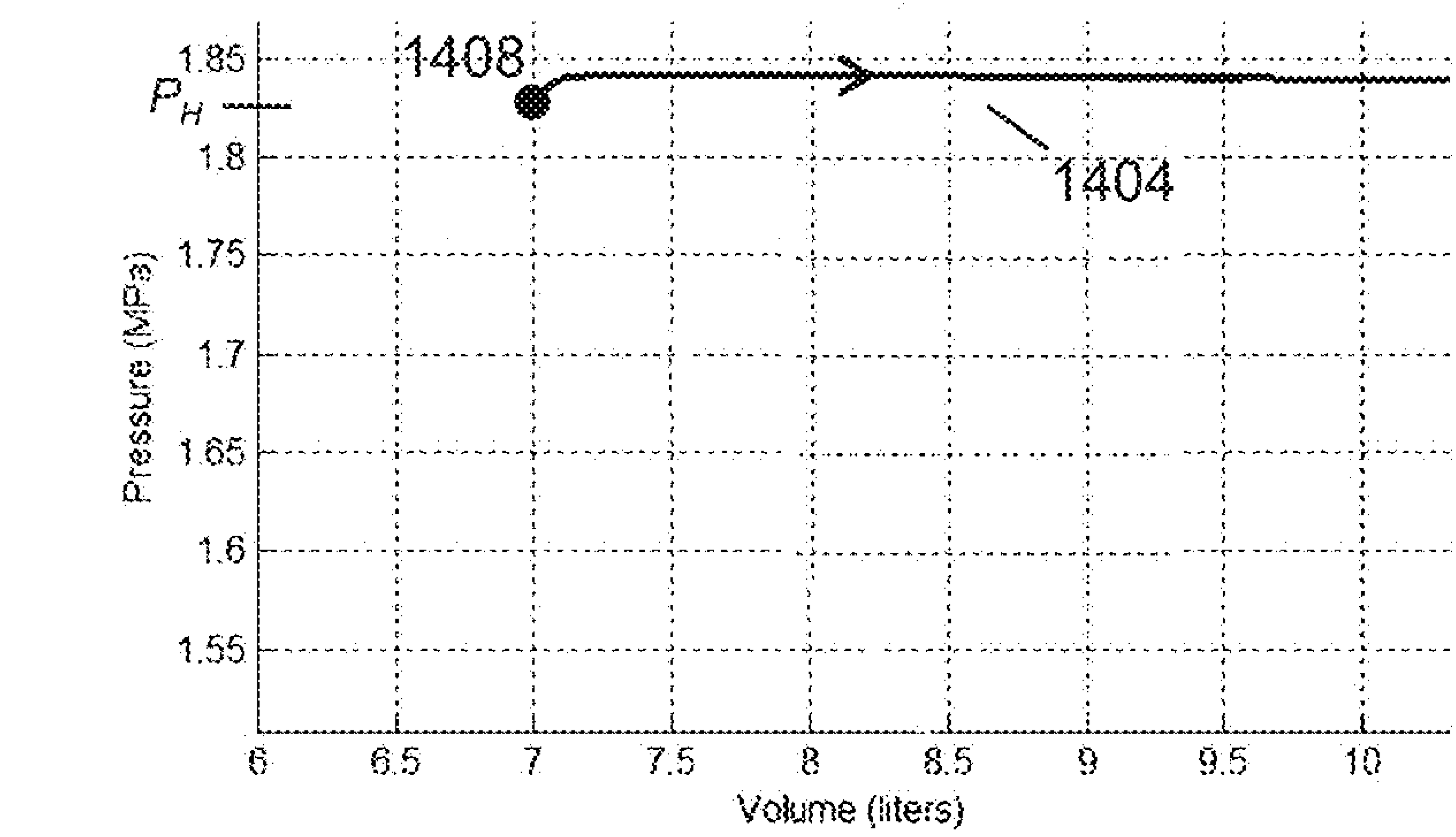


FIG. 14C

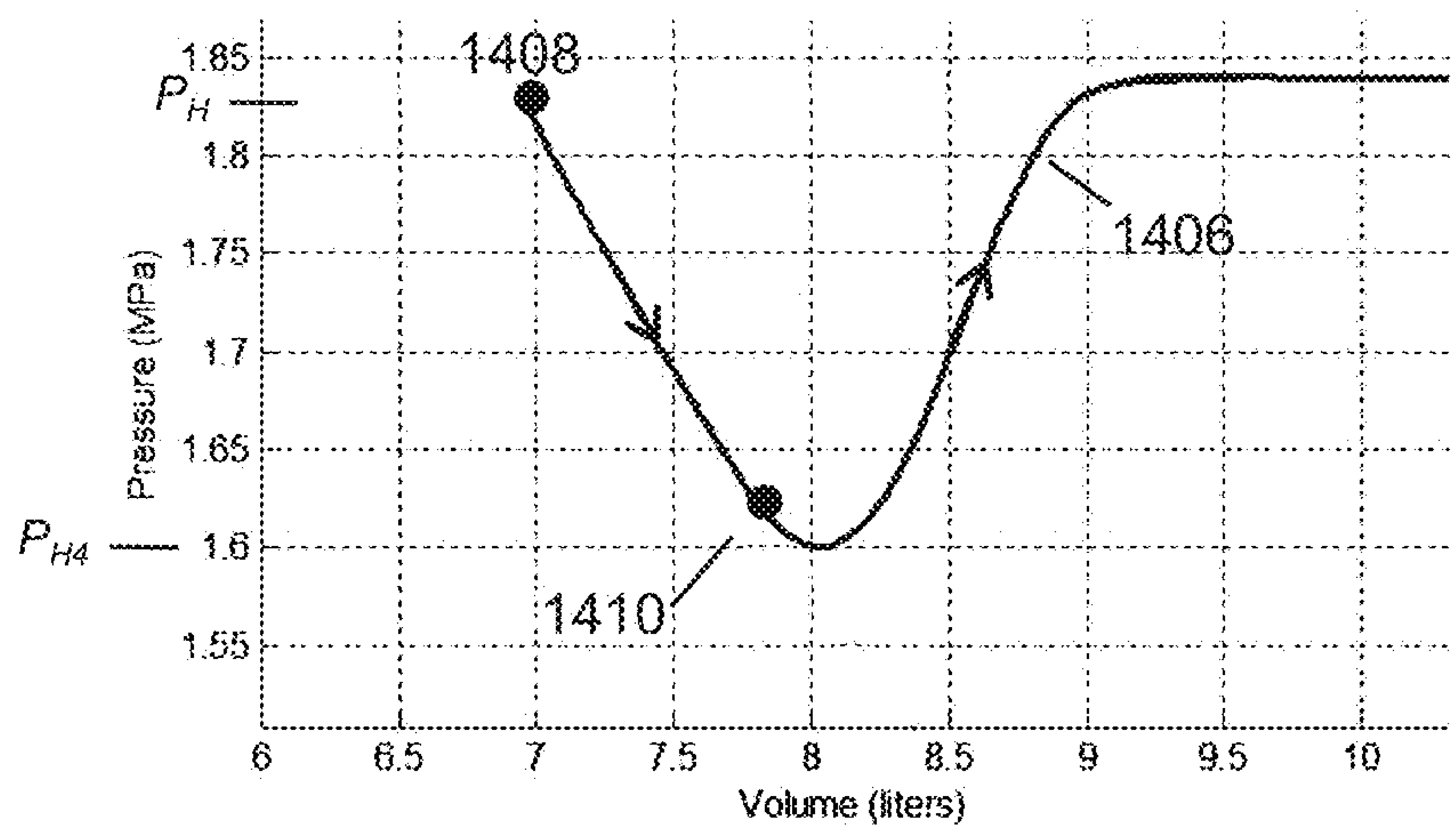


FIG. 15

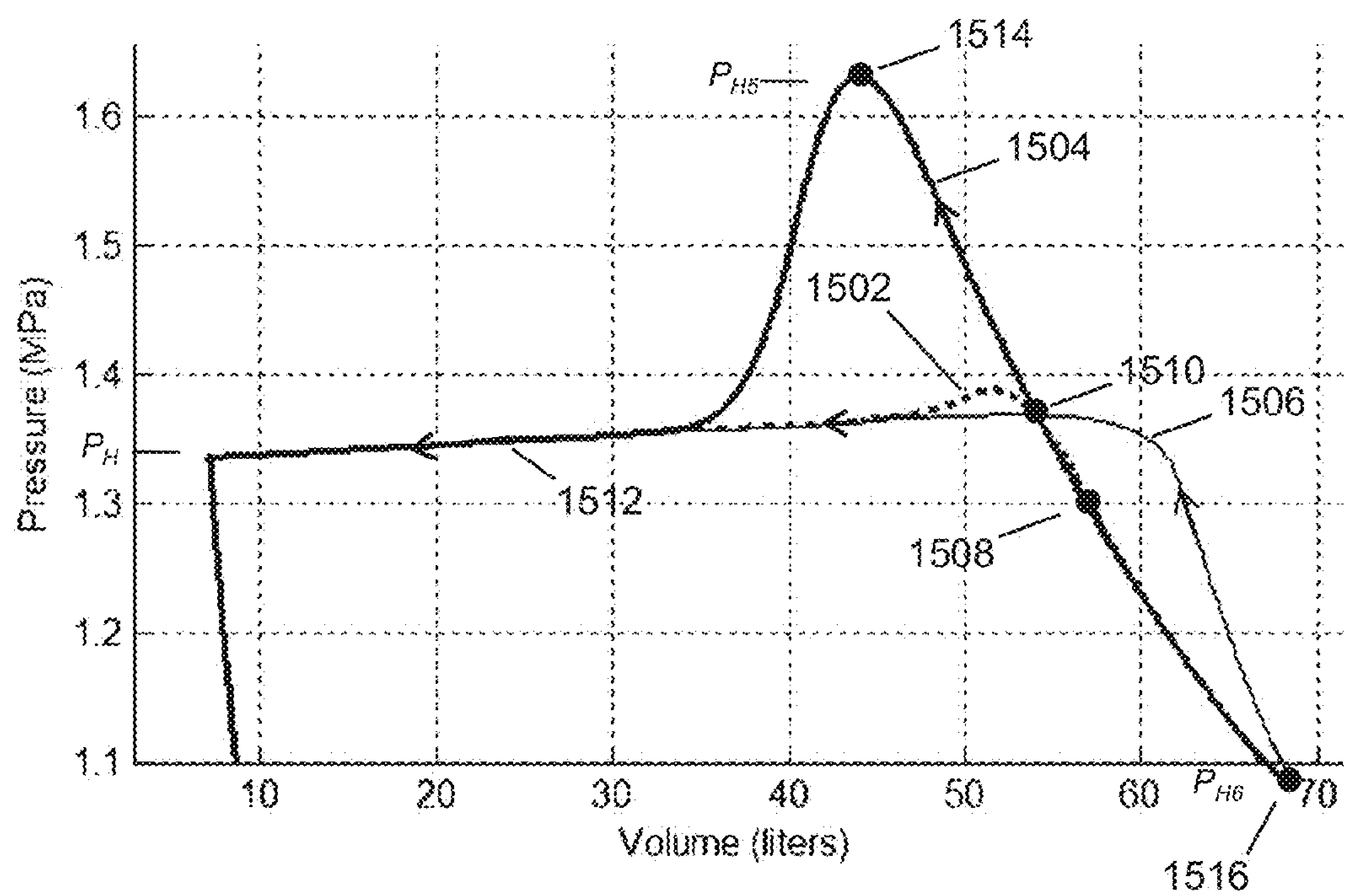
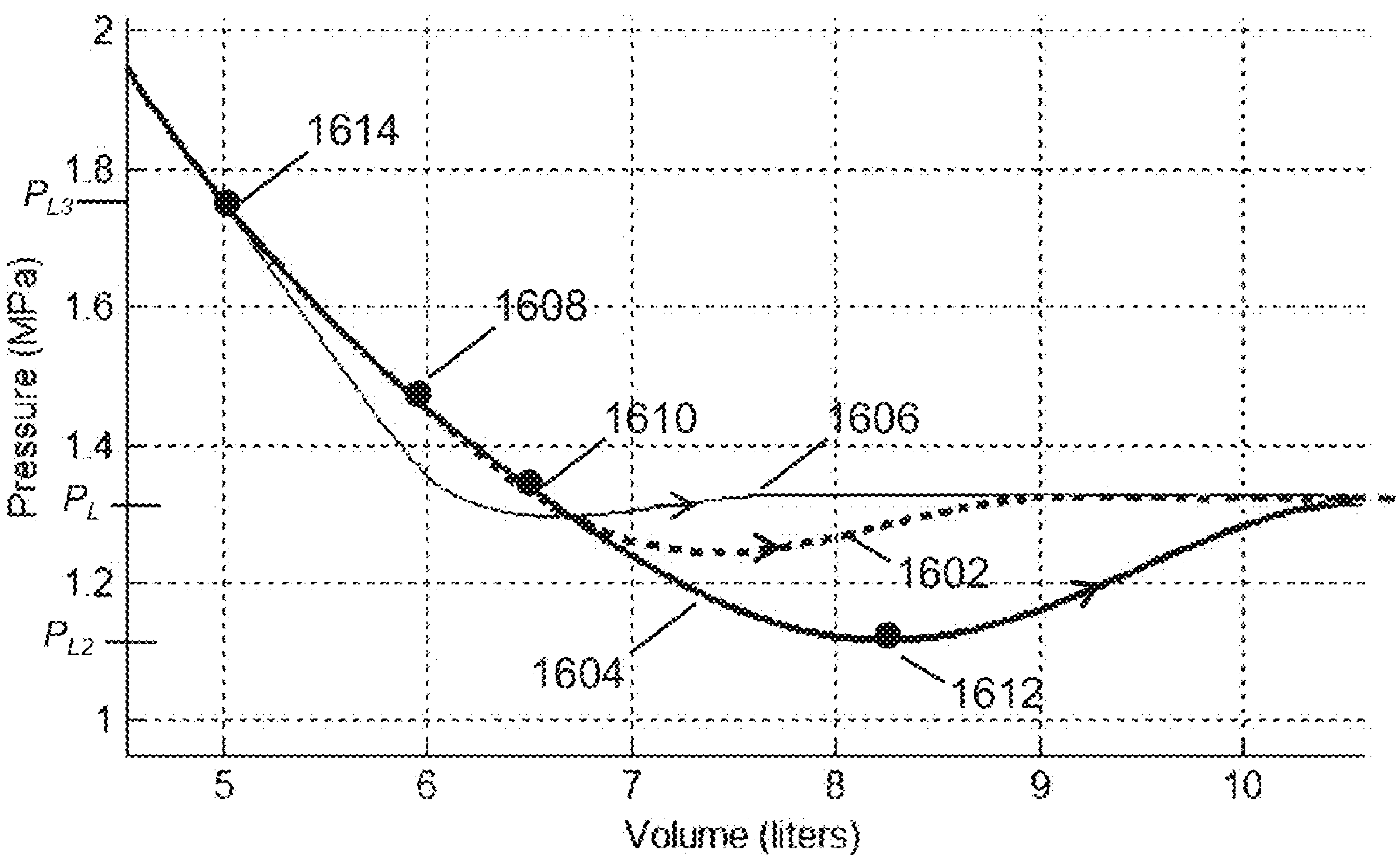

1514
P_{H5}
1504
1502
1510
1506
P_H
1512
1508
P_{H6}
1516
1.6
1.5
1.4
1.3
1.2
1.1
Pressure (MPa)
10
20
30
40
50
60
70
Volume (liters)

FIG. 16



2
1.8
1.6
1.4
1.2
1
P_{L3}
P_L
P_{L2}
Pressure (MPa)
1614
1608
1610
1606
1602
1604
1612
5
6
7
8
9
10
Volume (liters)

FIG. 17A

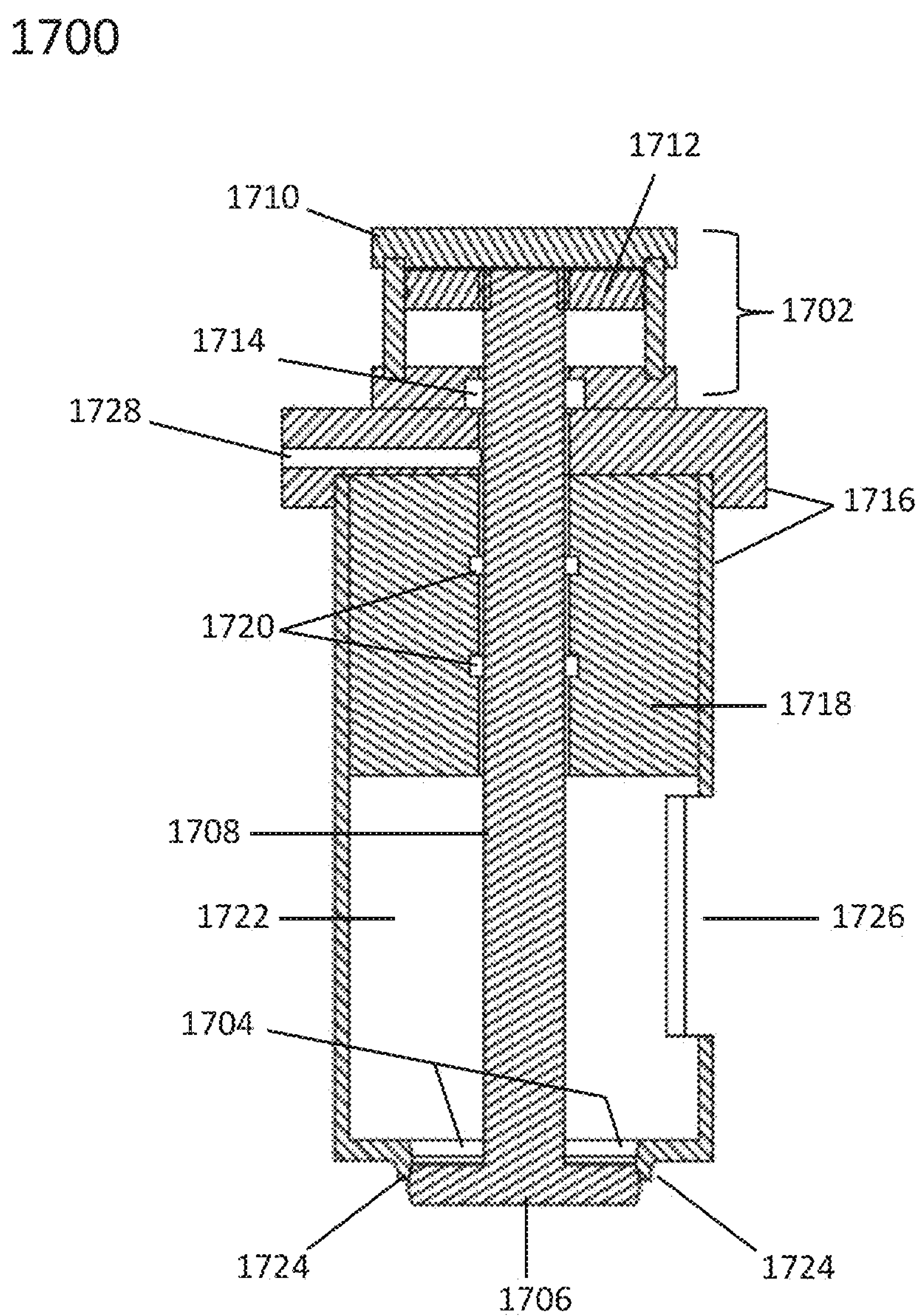

1700
1712
1710
1702
1714
1728
1716
1720
1718
1708
1722
1726
1704
1724
1706
1724

FIG. 17B

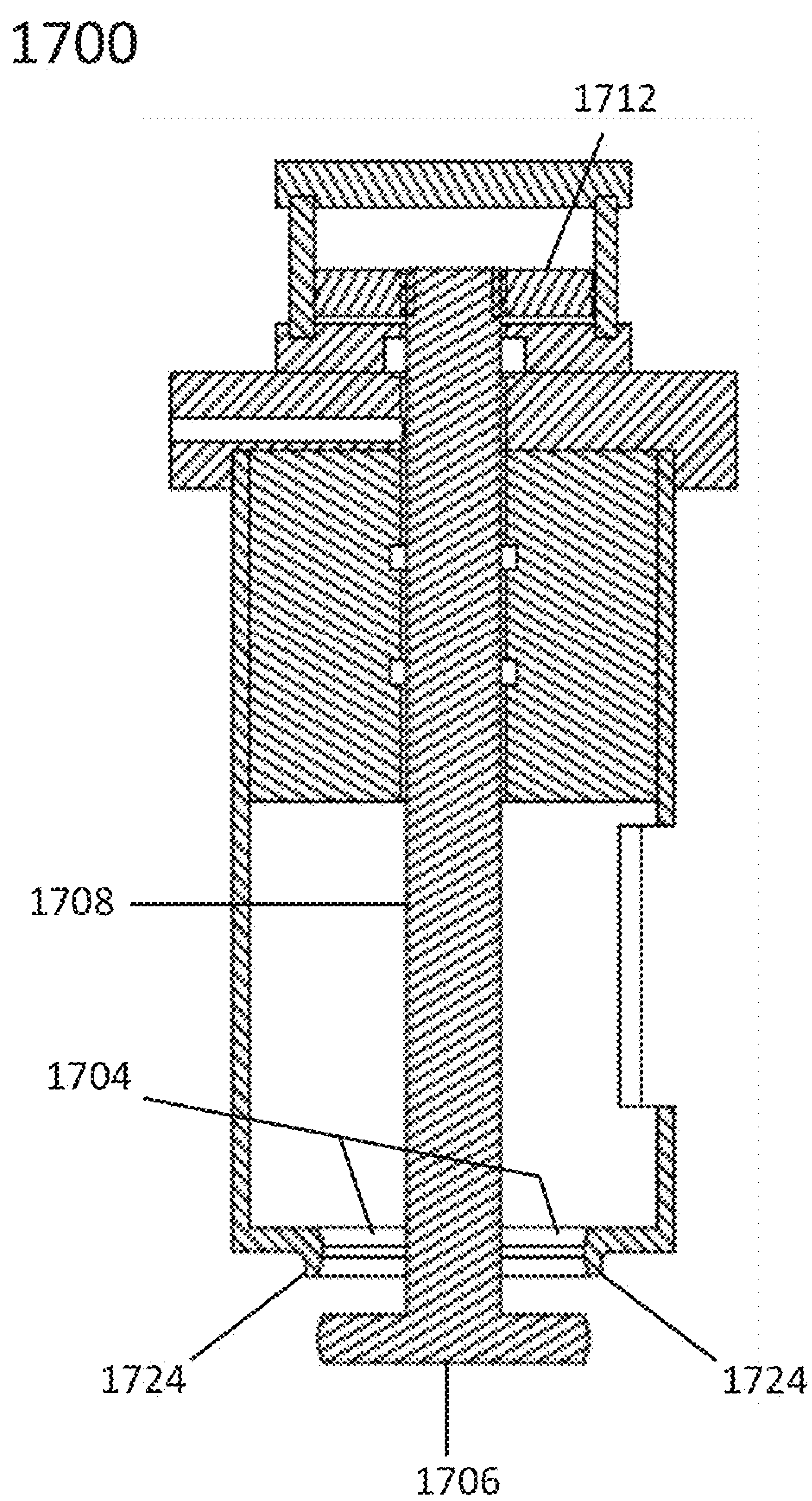


FIG. 18A

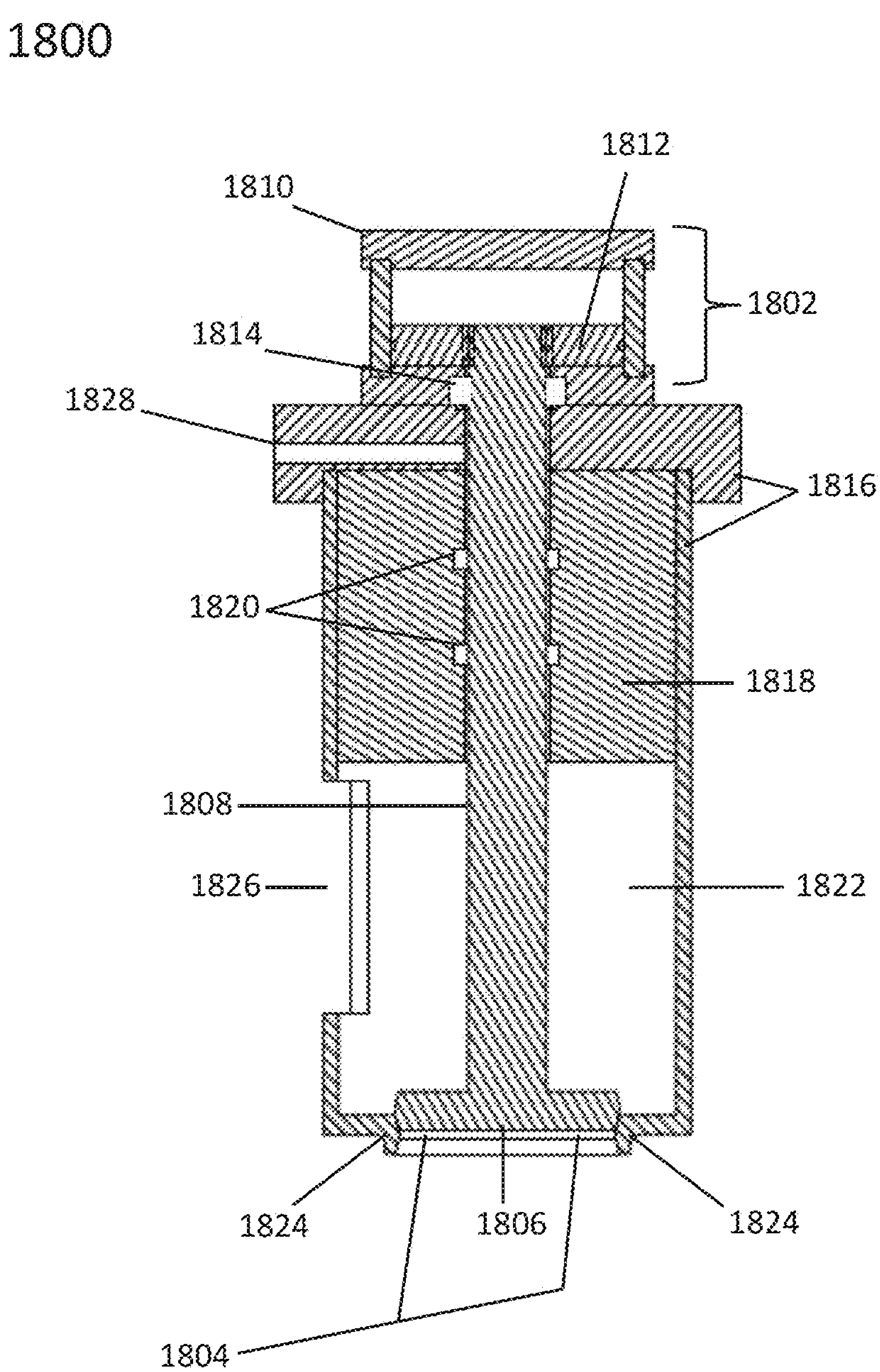

1800
1810
1812
1802
1814
1828
1816
1820
1818
1808
1826
1822
1824
1806
1824
1804

FIG. 18B

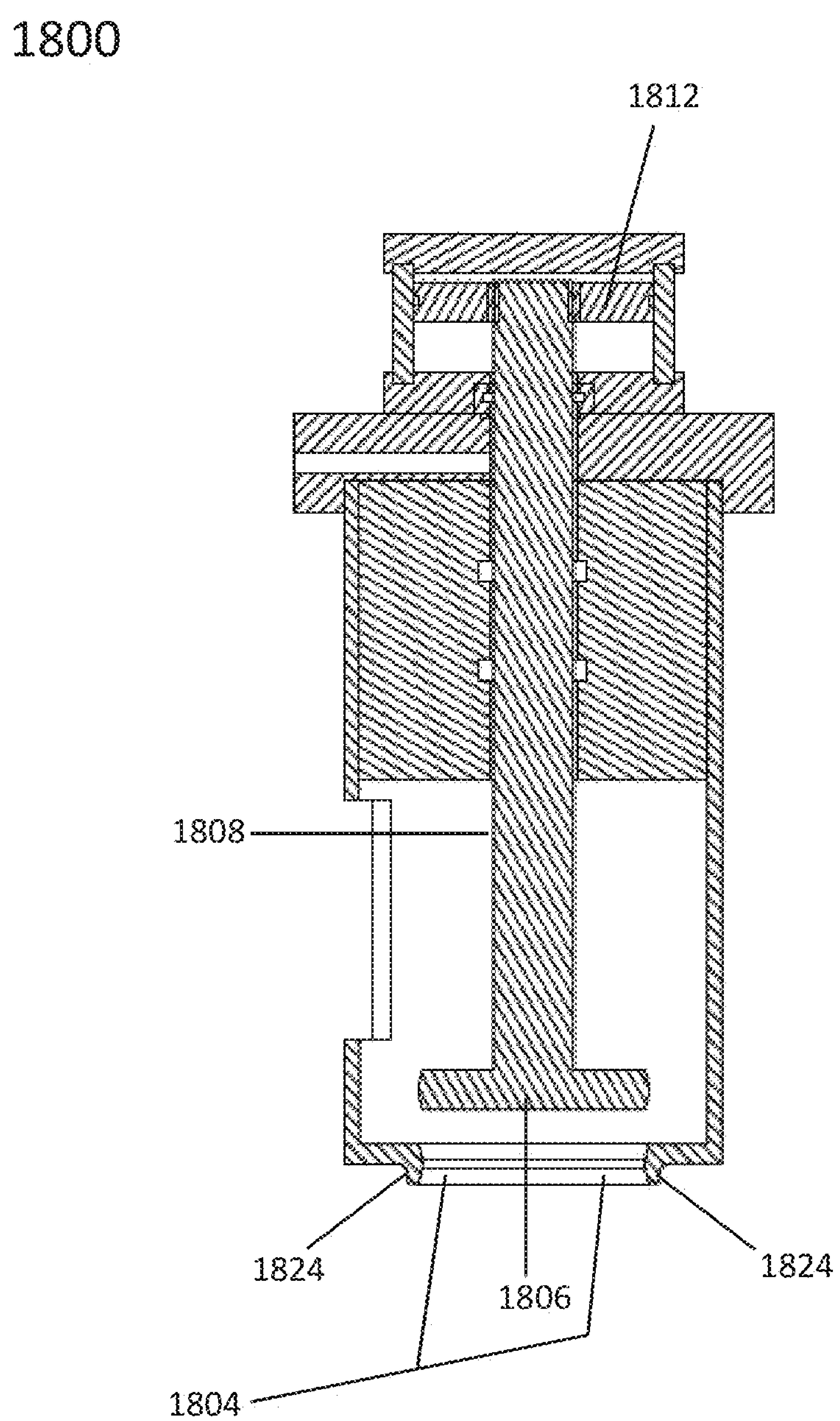


FIG. 19A

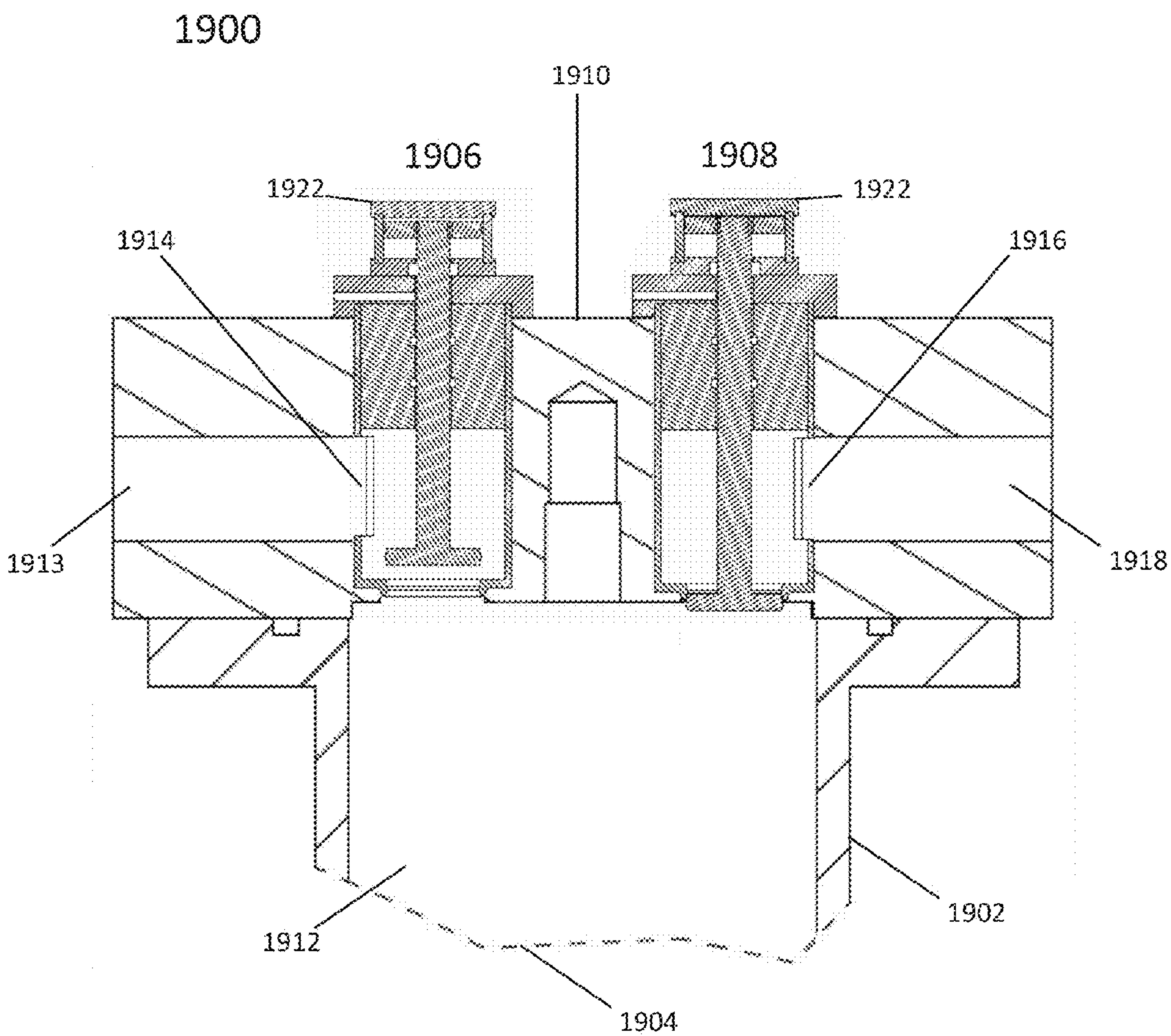

1900
1910
1906
1908
1922
1922
1914
1916
1913
1918
1912
1902
1904

FIG. 19B

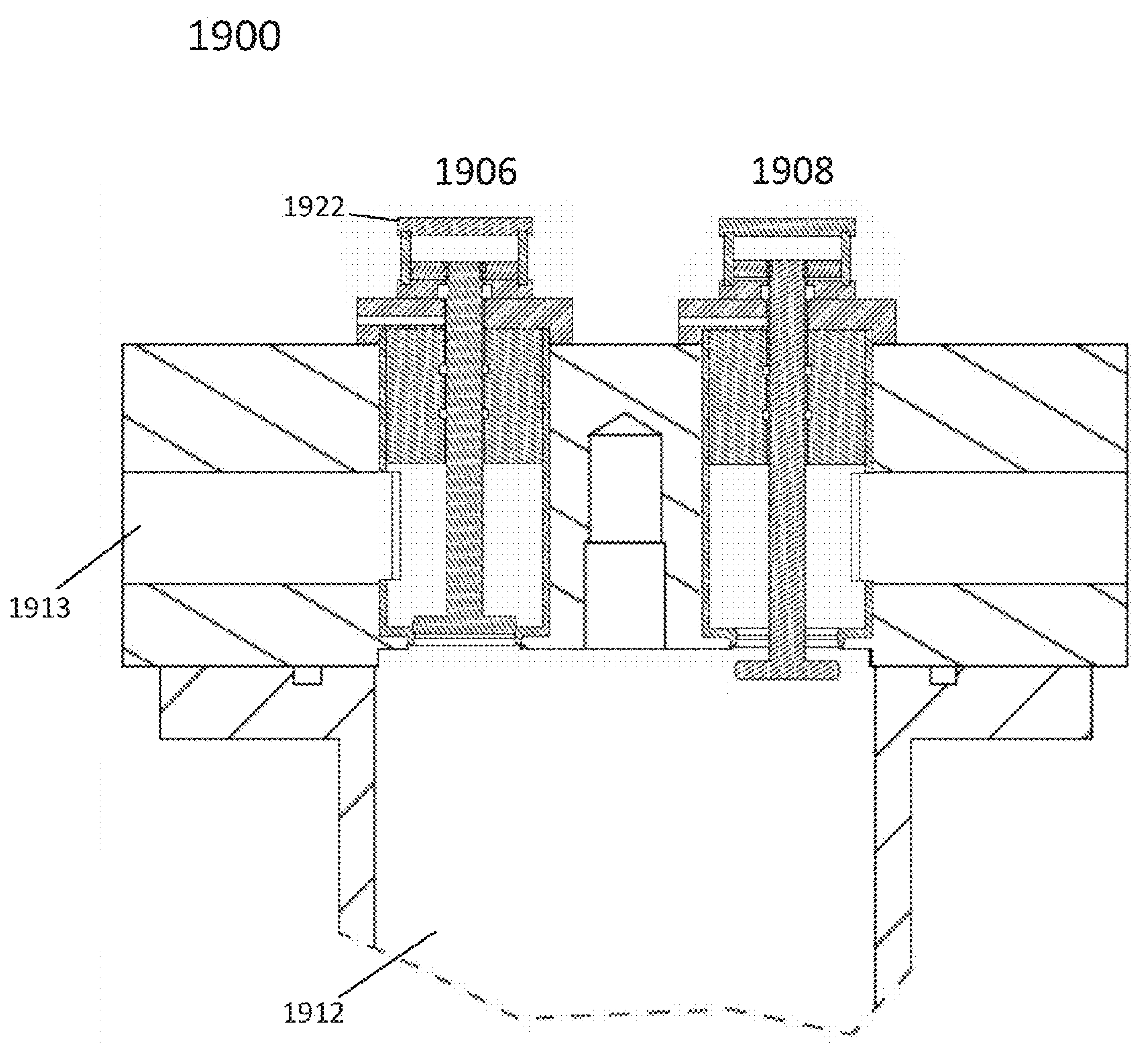


FIG. 20

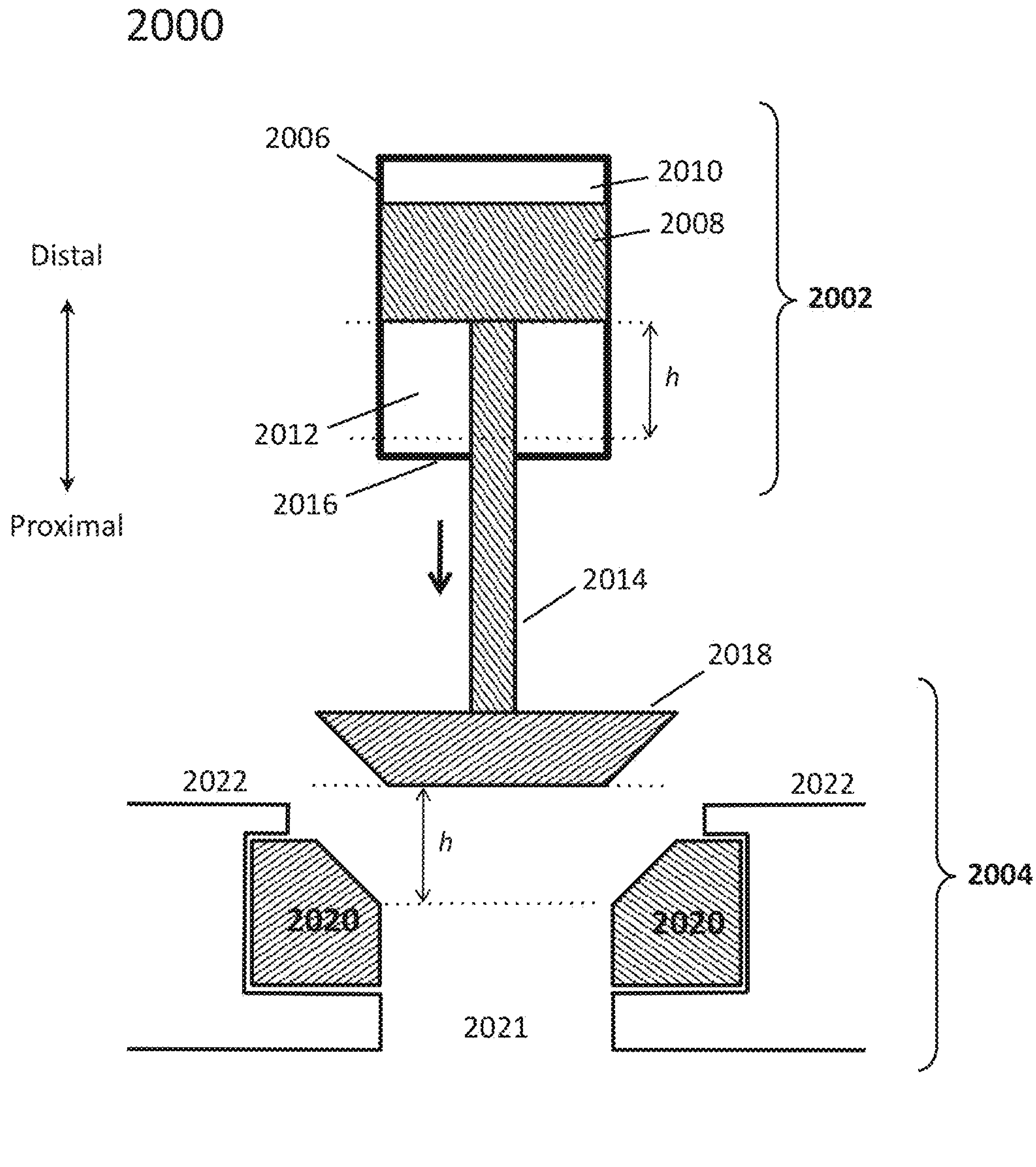

2000
2006
2010
2008
Distal
2002
h
2012
2016
Proximal
2014
2018
2022
2022
h
2004
2020
2020
2021

FIG. 21

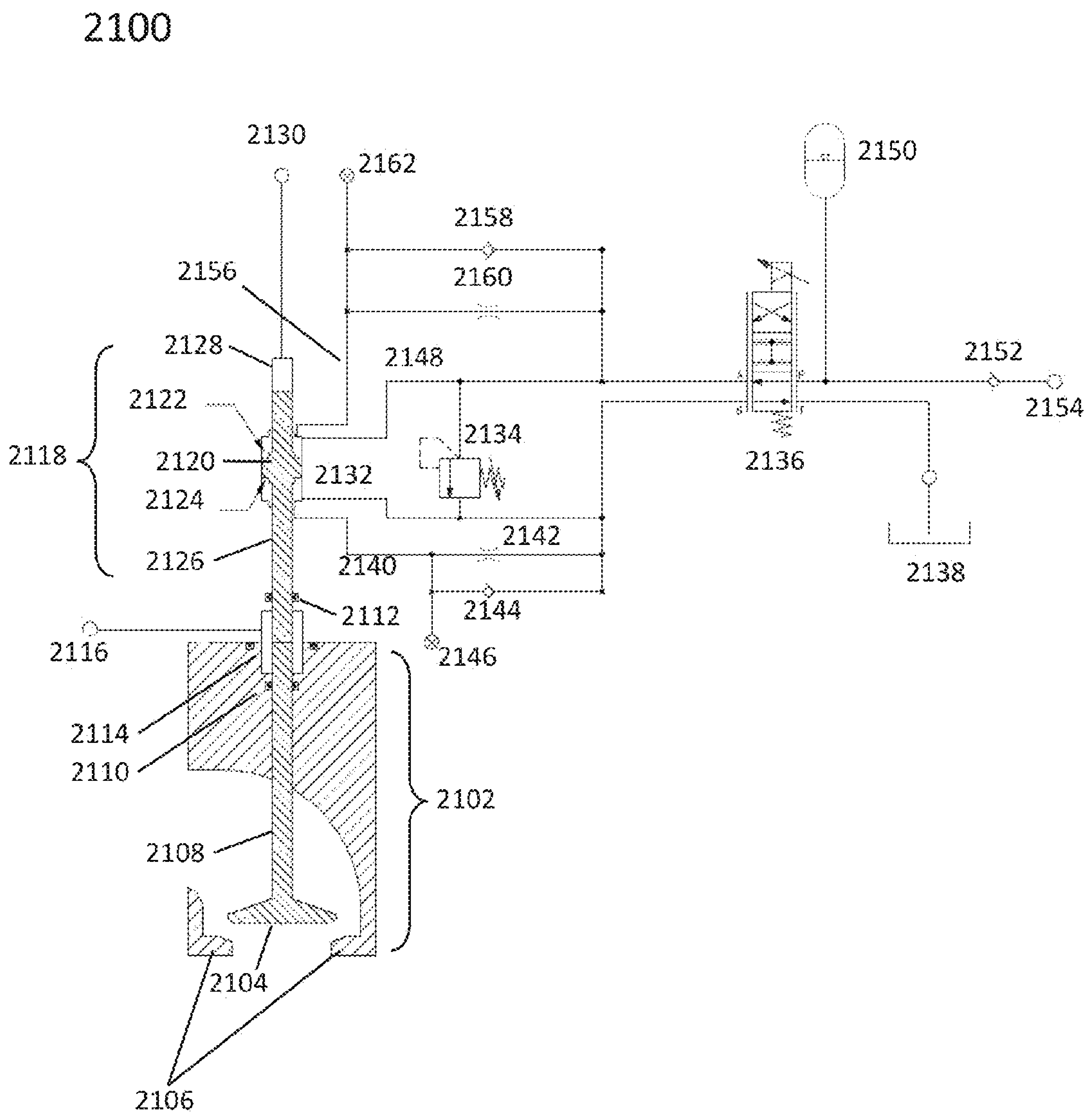

2100
2130
2162
2150
2158
2160
2156
2128
2148
2152
2154
2122
2134
2118
2120
2132
2136
2124
2142
2126
2140
2138
2112
2144
2116
2146
2114
2110
2102
2108
2104
2106

FIG. 22

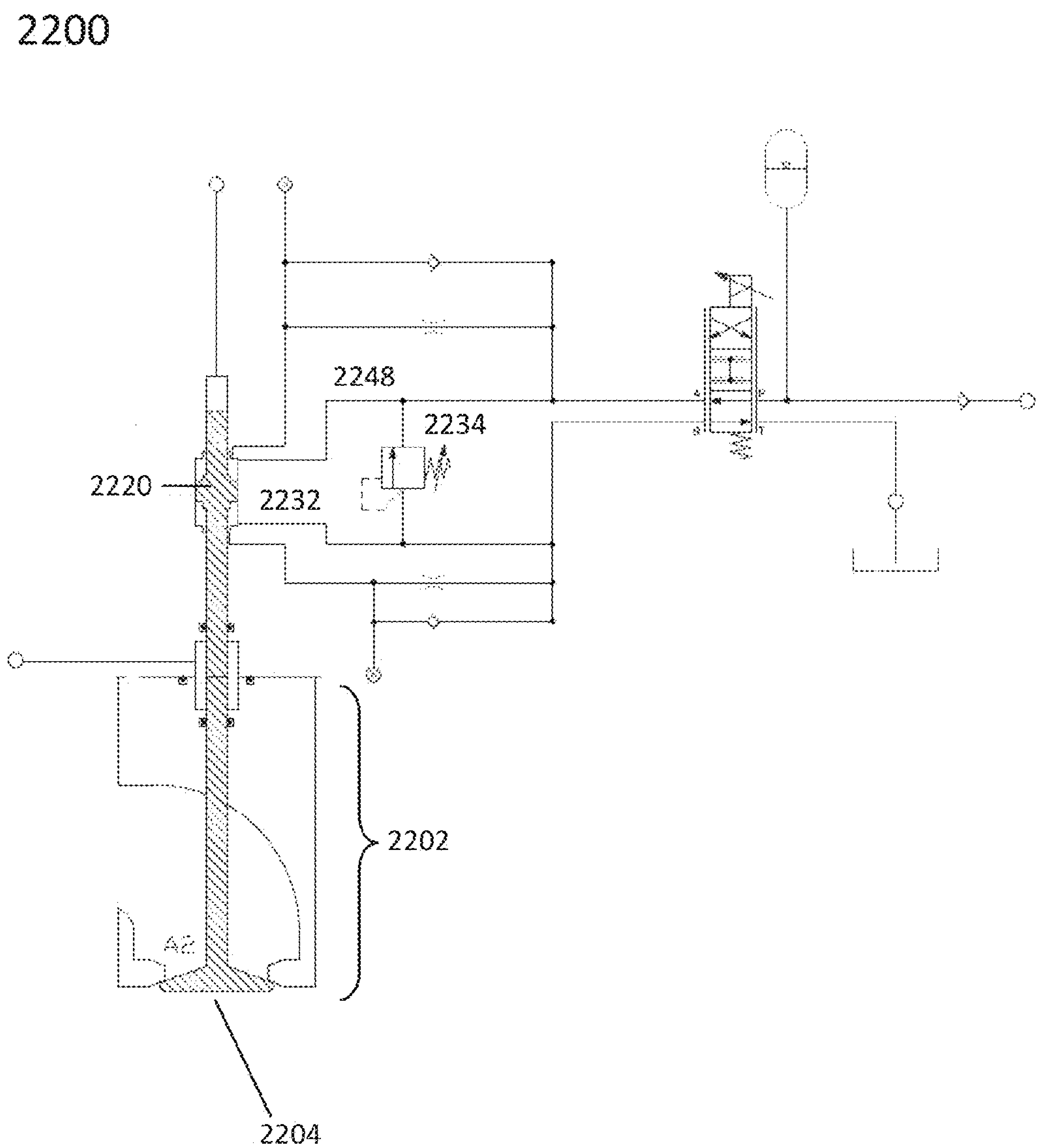


FIG. 23A

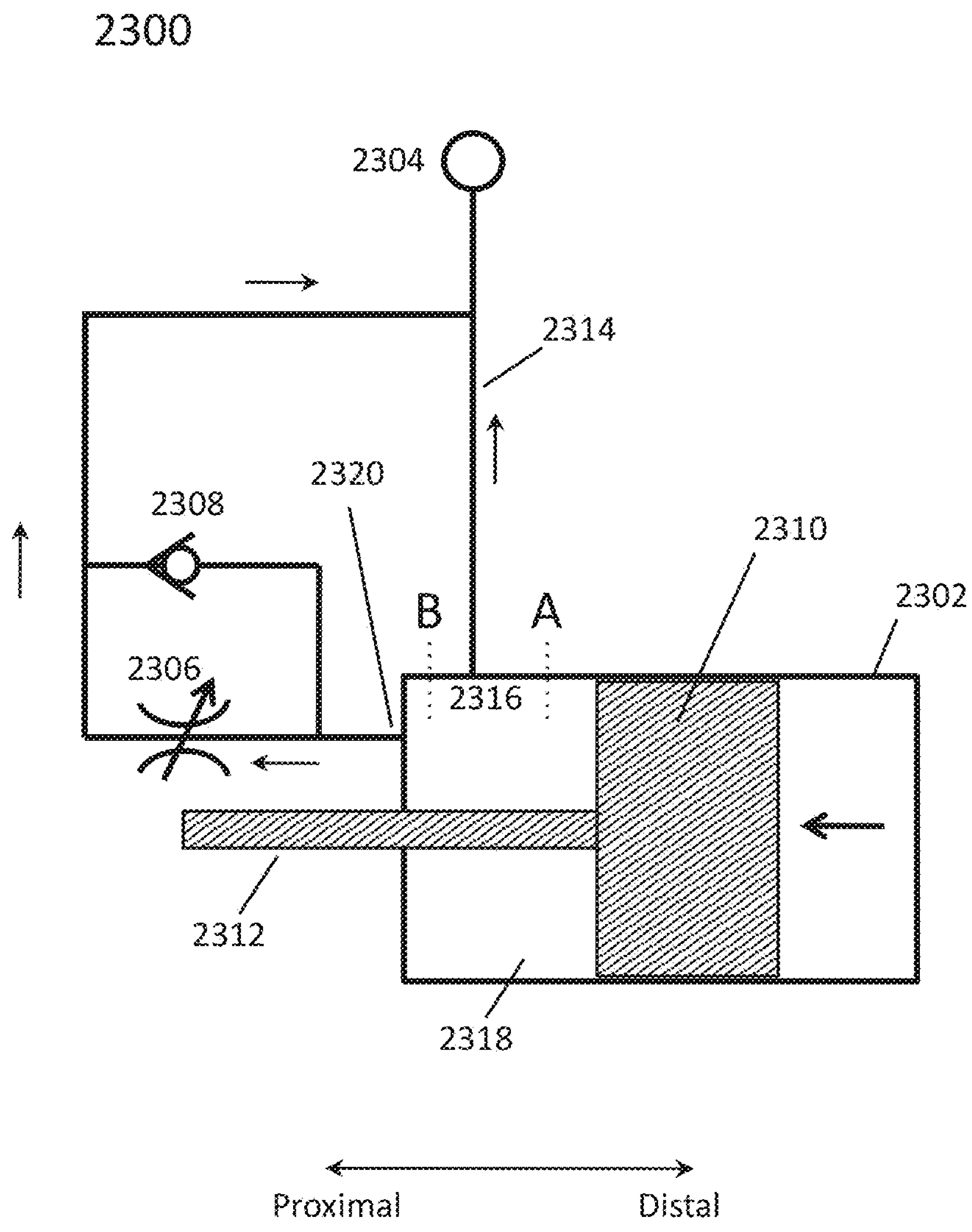

2300
2304
2314
2320
2308
2310
2302
B
A
2306
2316
2312
2318
Proximal
Distal

FIG. 23B

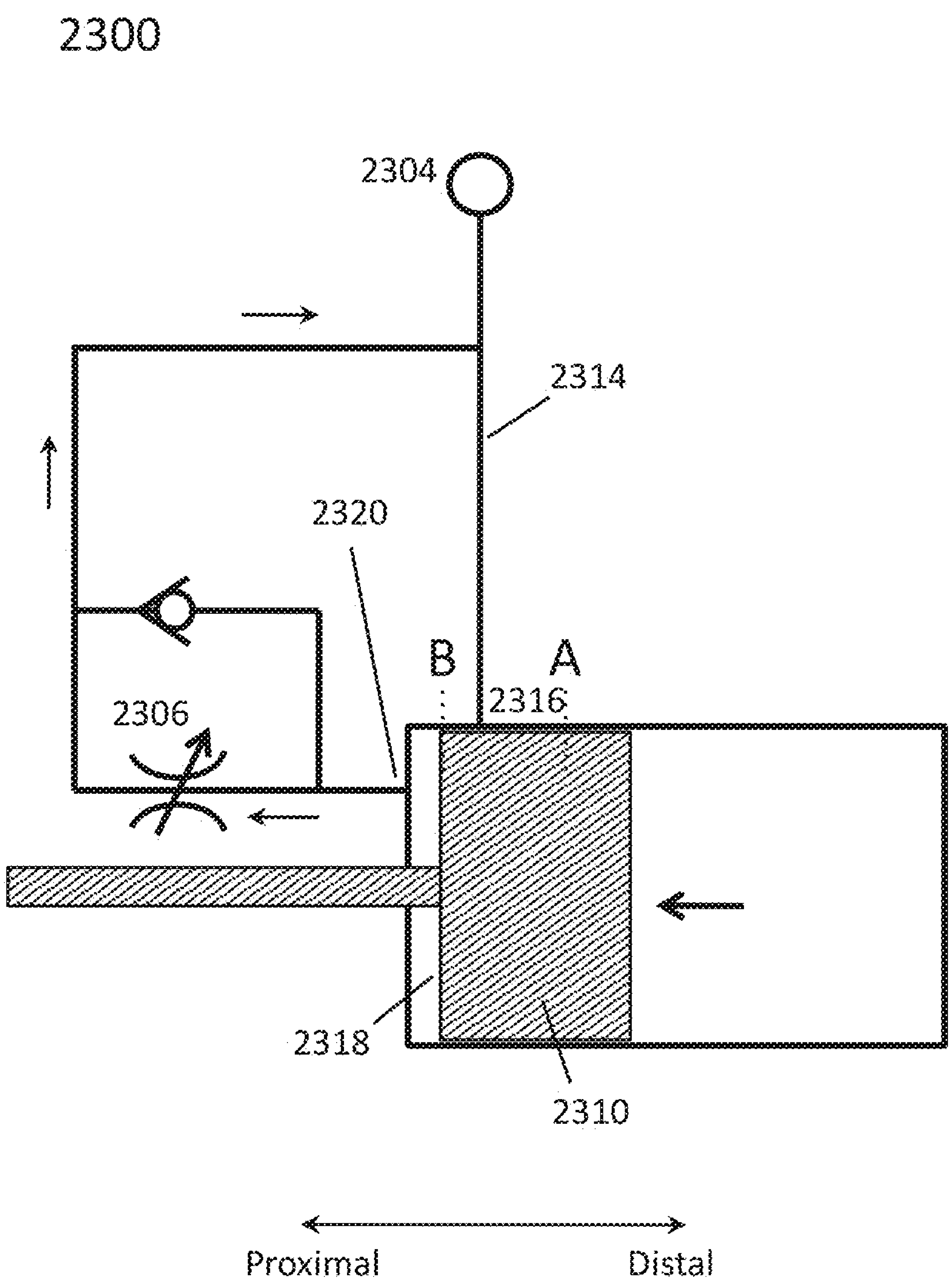


FIG. 23C

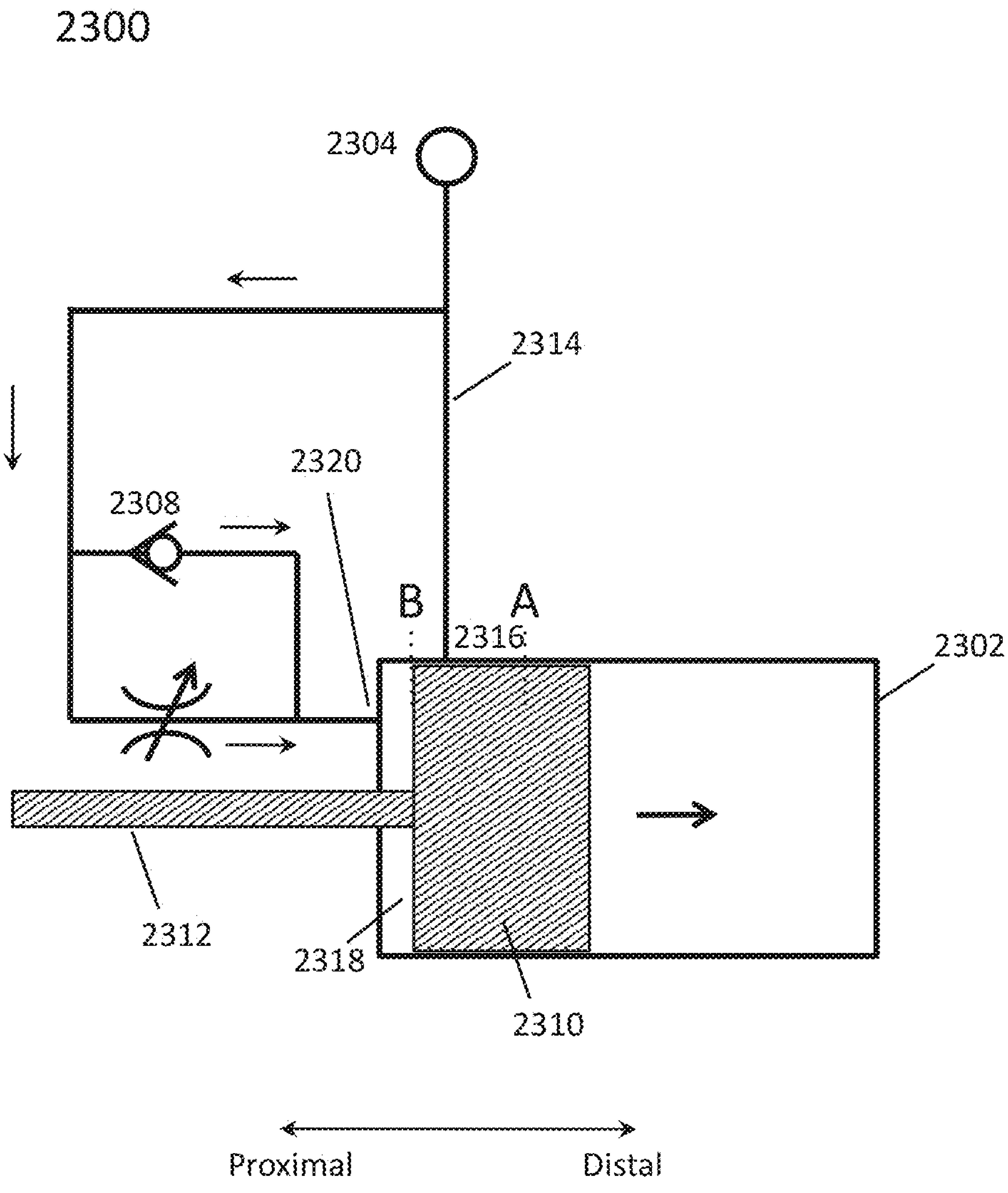

2300
2304
2314
2320
2308
B
A
2316
2302
2312
2318
2310
Proximal
Distal

FIG. 23D

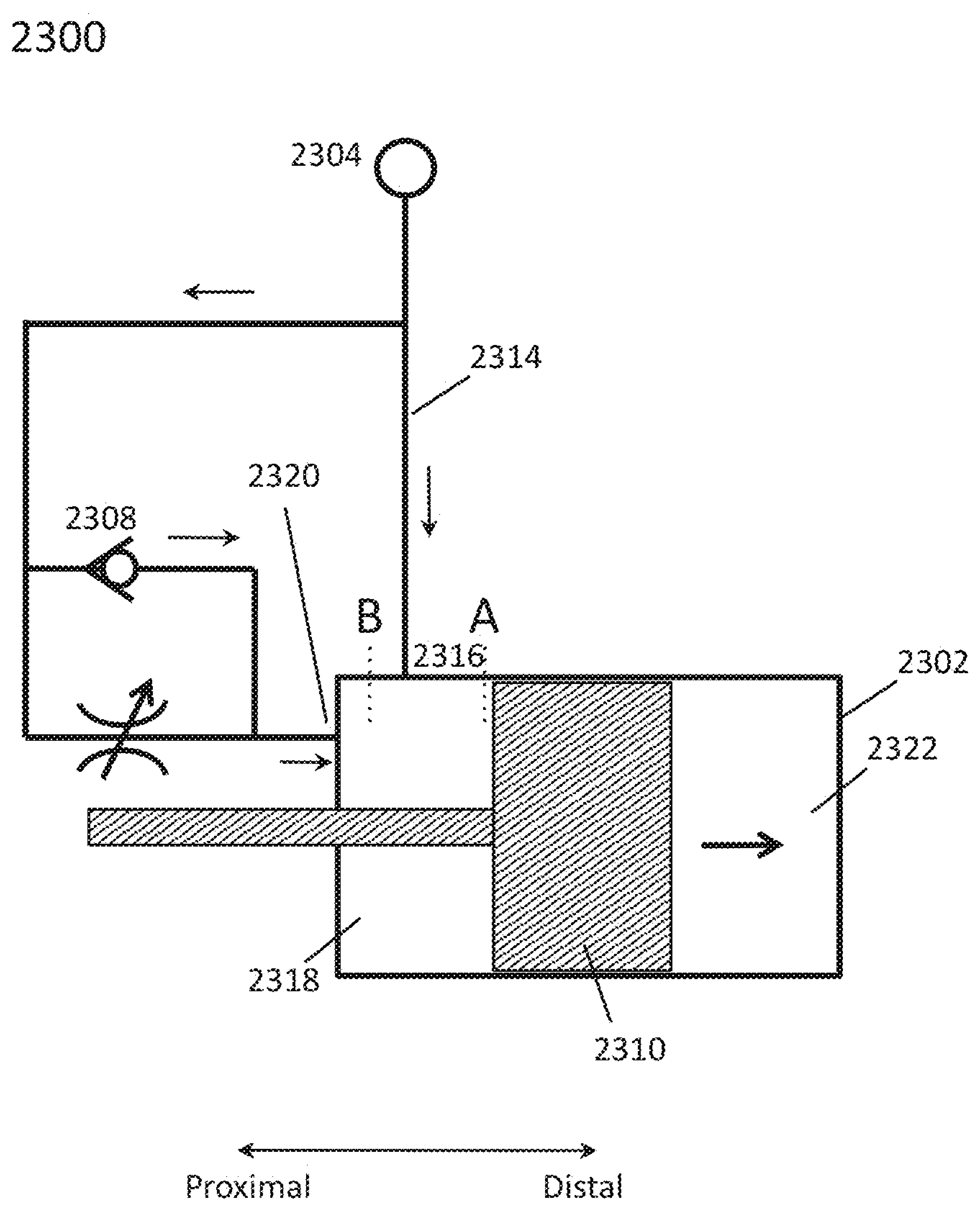

2300
2304
2314
2320
2308
B
A
2316
2302
2322
2318
2310
Proximal
Distal

FIG. 24A

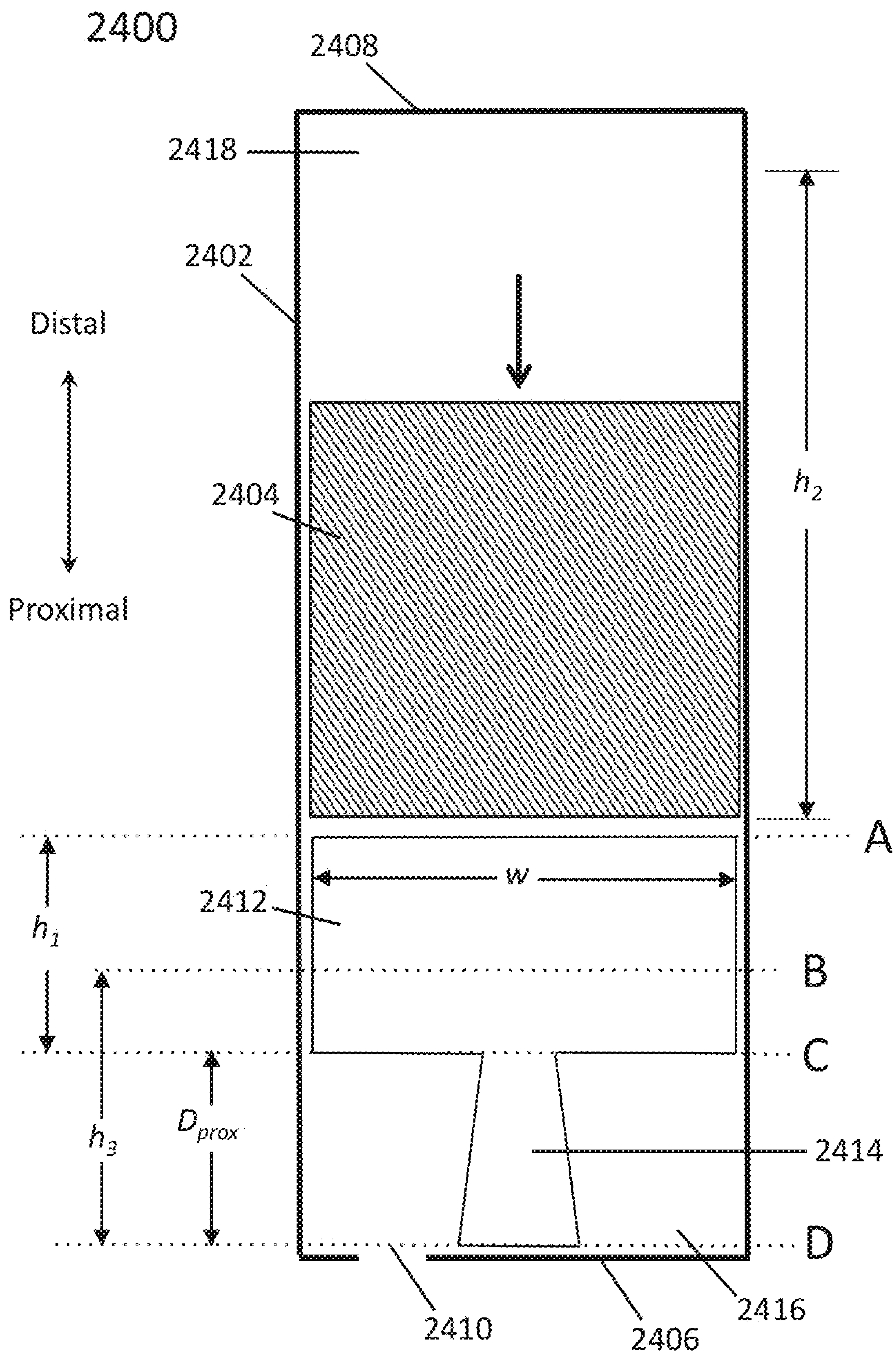
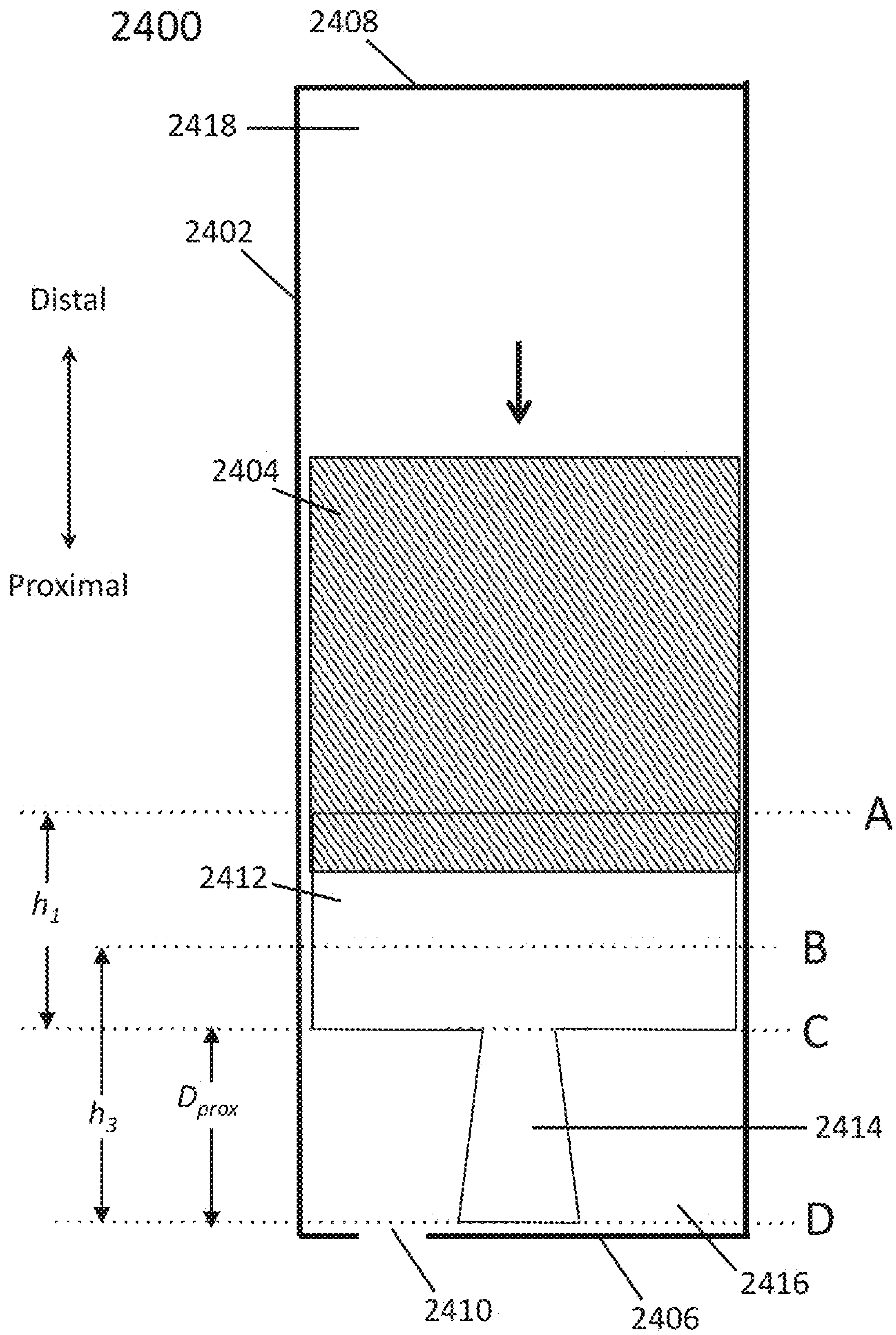

2400
2408
2418
2402
Distal
Proximal
2404
h_2
A
w
2412
h_1
B
C
h_3
D_{prox}
2414
D
2410
2406
2416

FIG. 24B



2400
2408
2418
2402
Distal
Proximal
2404
A
2412
h_1
B
C
h_3
D_{prox}
2414
D
2410
2406
2416

FIG. 24C

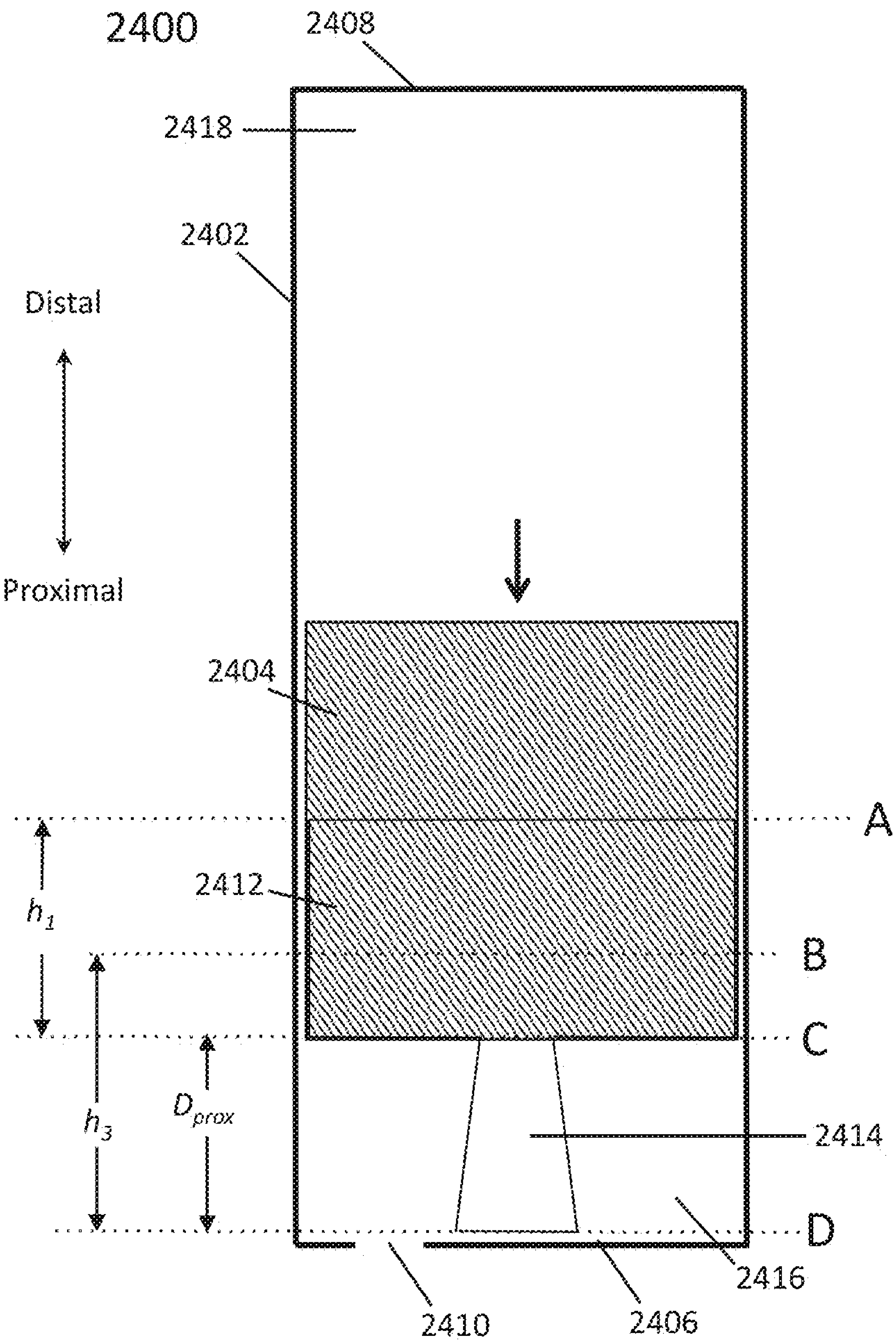
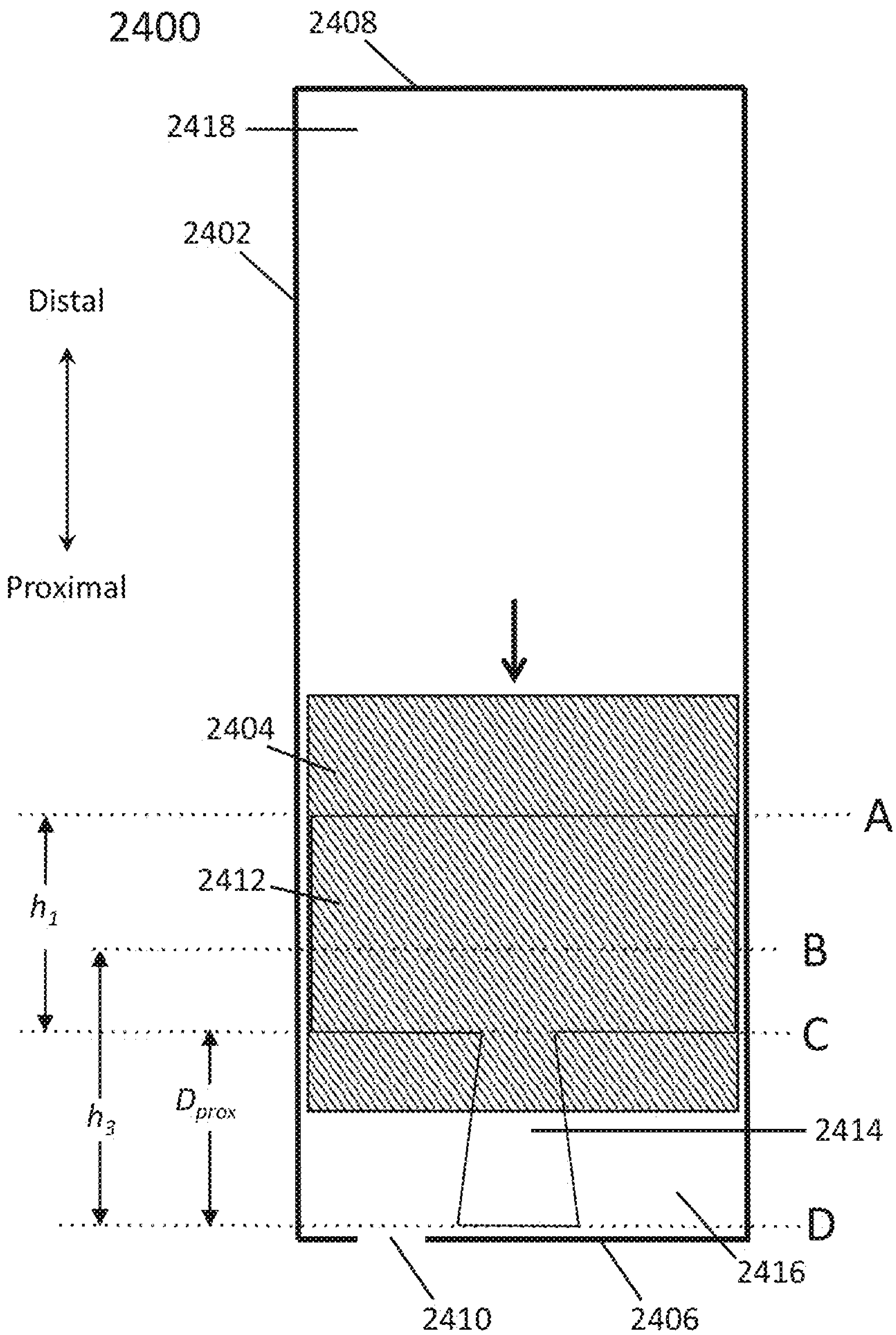
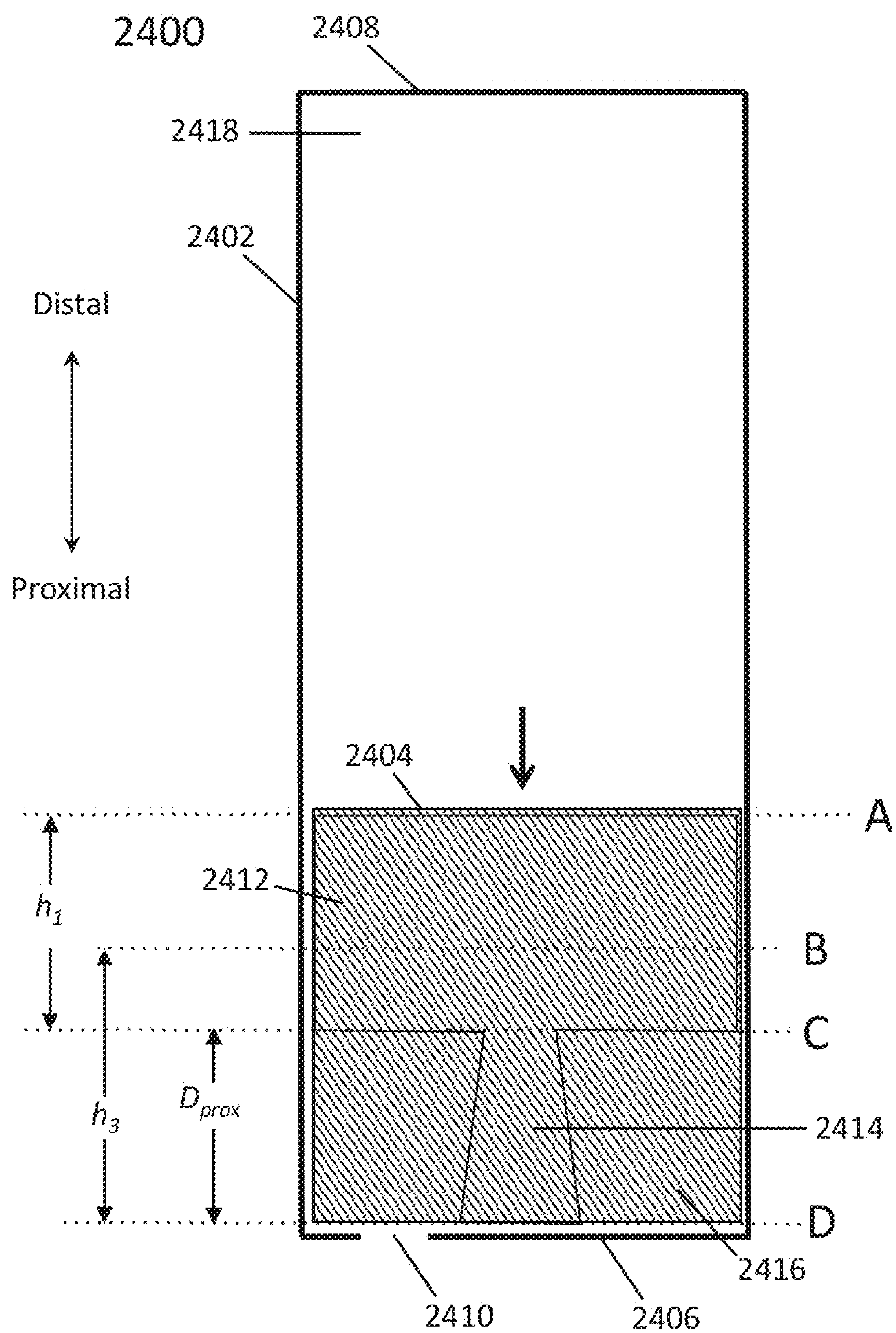

2400
2408
2418
2402
Distal
Proximal
2404
A
2412
h_1
B
C
h_3
D_{prox}
2414
D
2416
2410
2406

FIG. 24D



2400
2408
2418
2402
Distal
Proximal
2404
A
2412
h_1
B
C
h_3
D_{prox}
2414
D
2416
2410
2406

FIG. 24E



2400
2408
2418
2402
Distal
Proximal
2404
A
2412
h_1
B
C
h_3
D_{prox}
2414
D
2416
2410
2406

FIG. 24F

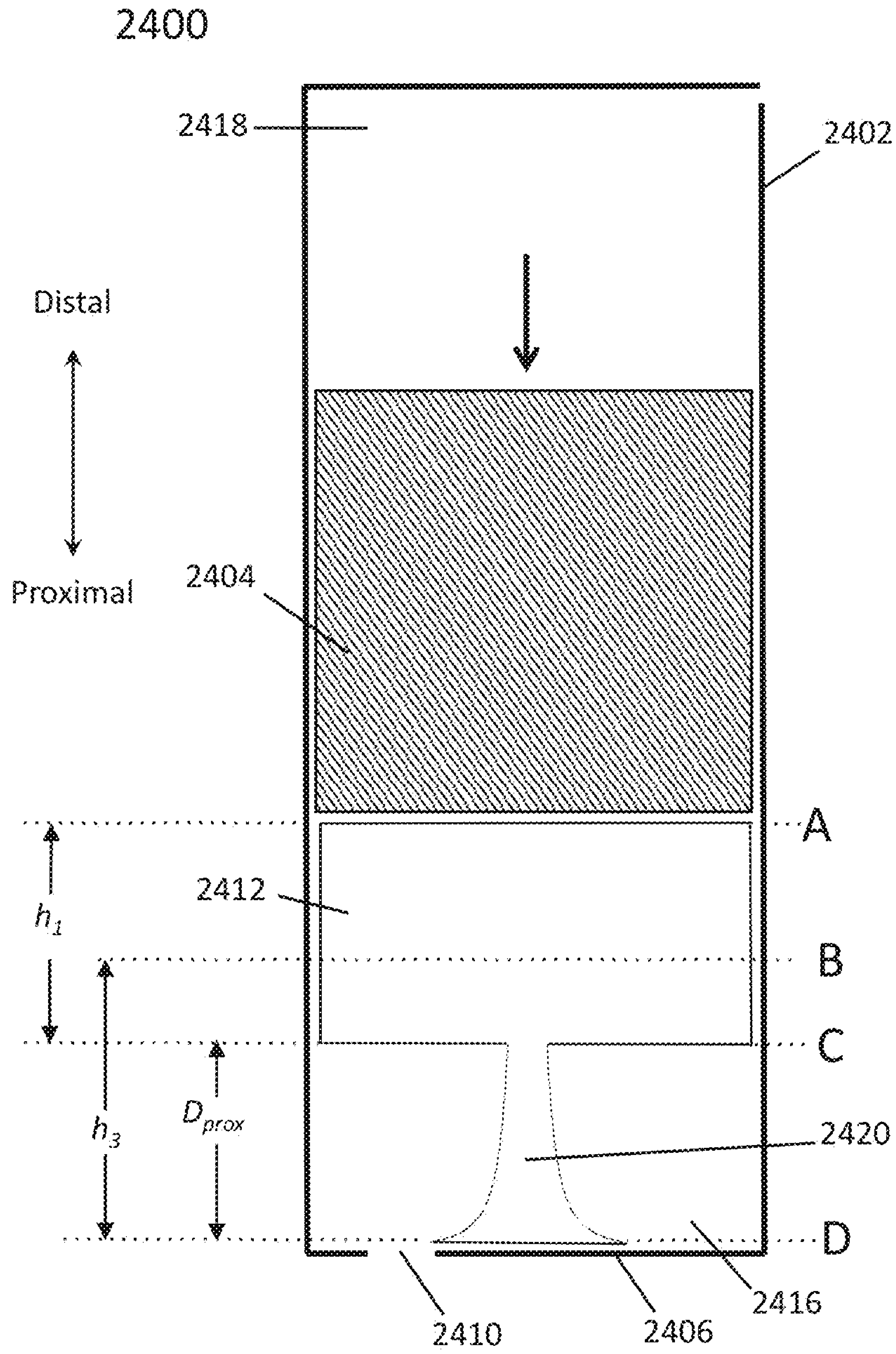

2400
2418
2402
Distal
Proximal
2404
A
2412
h_1
B
C
h_3
D_{prox}
2420
D
2416
2410
2406

FIG. 25

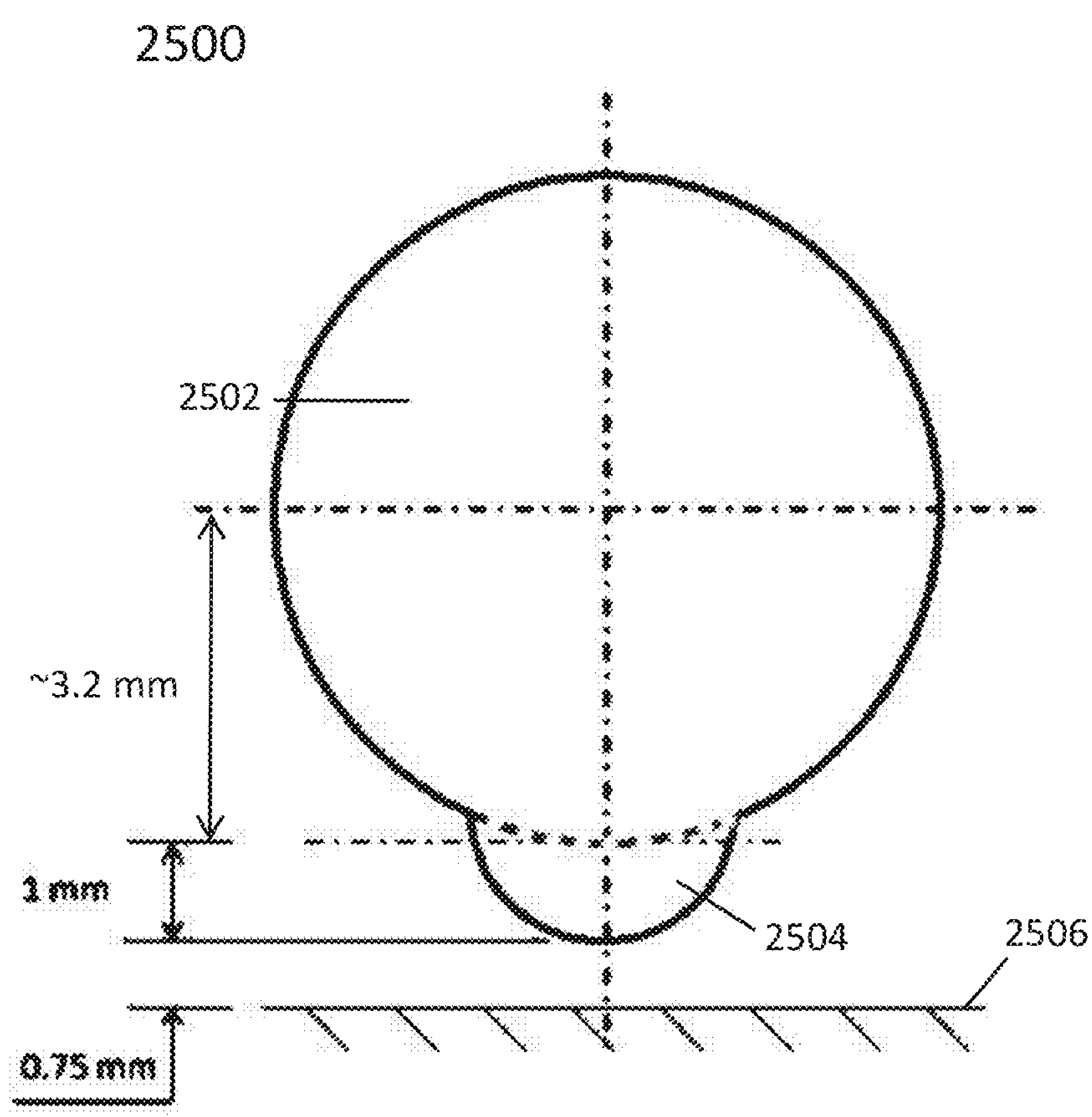


FIG. 26B

FIG. 26A

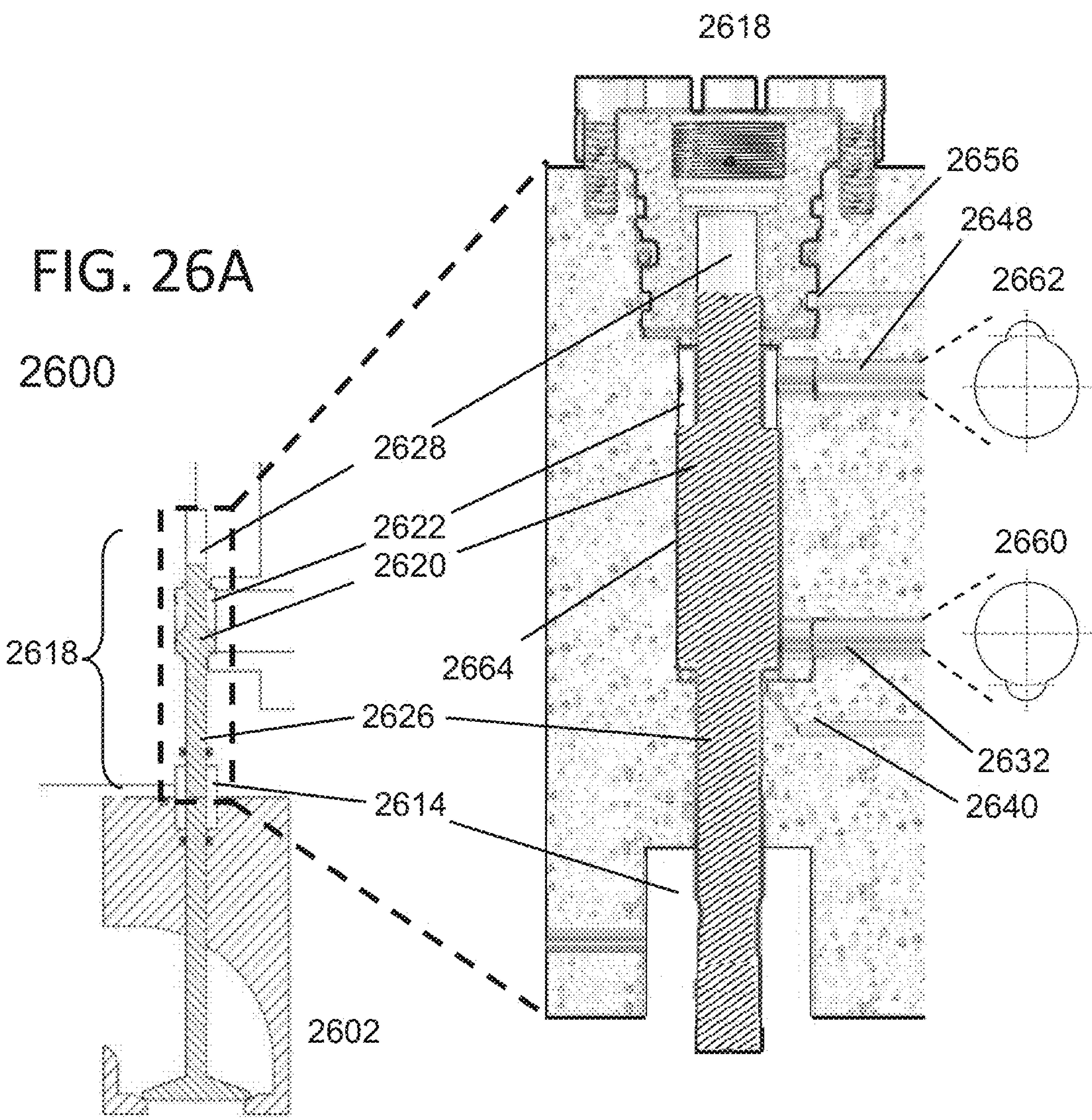


FIG. 27

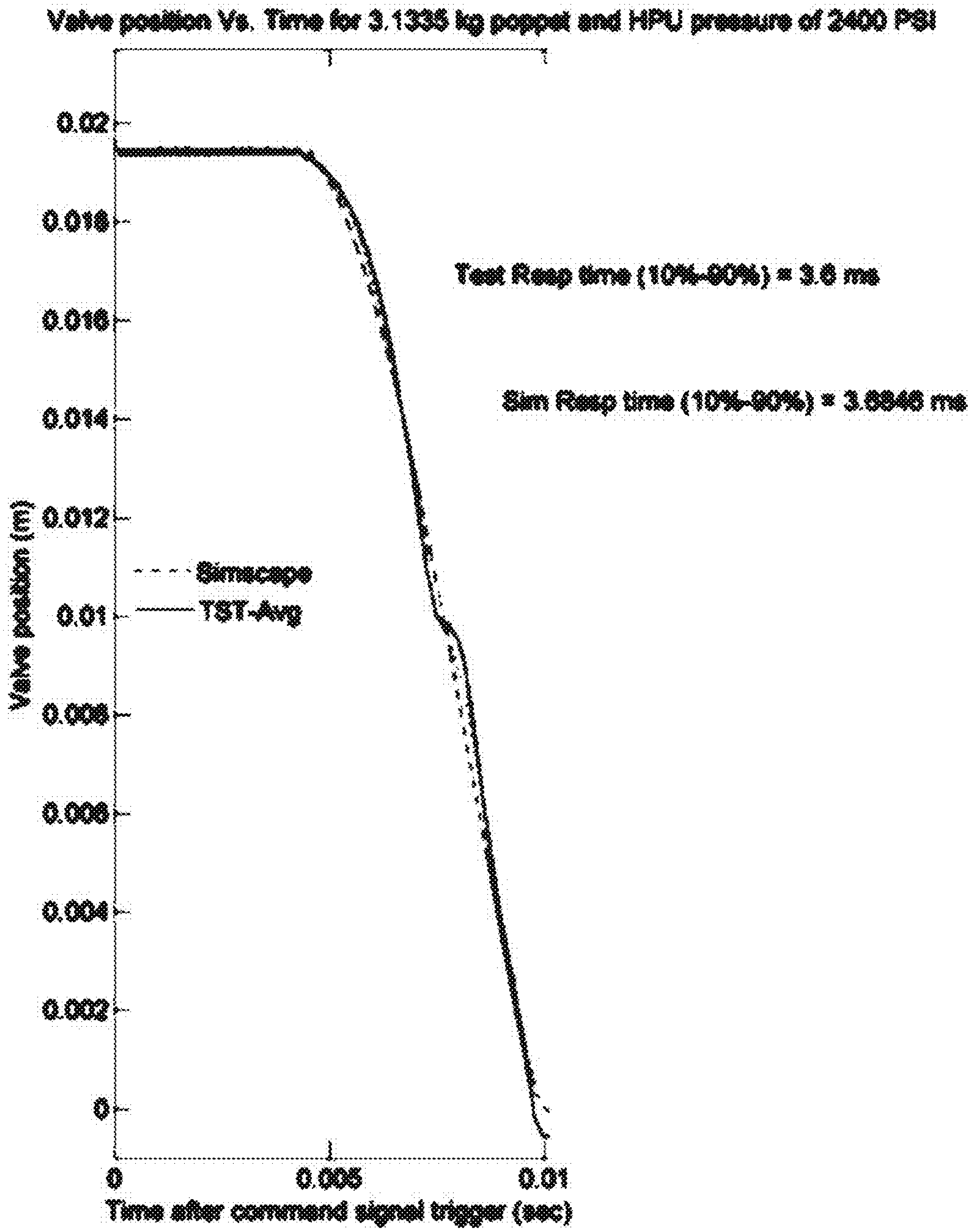

Valve position Vs. Time for 3.1335 kg poppet and HPU pressure of 2400 PSI
Test Resp time (10%-90%) = 3.6 ms
Sim Resp time (10%-90%) = 3.6846 ms
Simscape
TST-Avg
Valve position (m)
0.02
0.018
0.016
0.014
0.012
0.01
0.008
0.006
0.004
0.002
0
0.005
0.01
Time after command signal trigger (sec)

FIG. 28

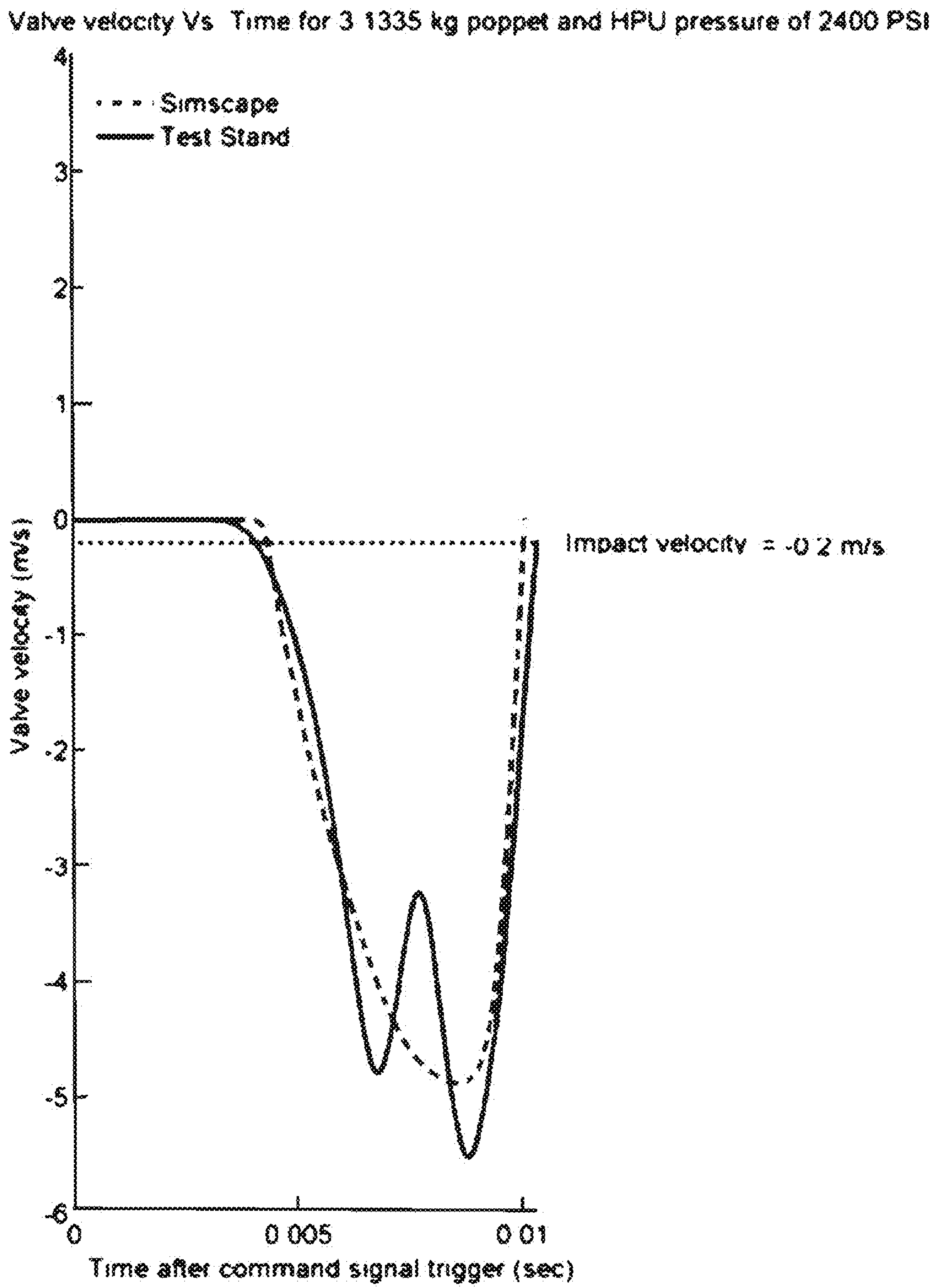

Valve velocity Vs Time for 3 1335 kg poppet and HPU pressure of 2400 PSI
Simscape
Test Stand
Impact velocity = -0 2 m/s
Valve velocity (m/s)
Time after command signal trigger (sec)
4
3
2
1
0
-1
-2
-3
-4
-5
-6
0
0 005
0 01

FIG. 29

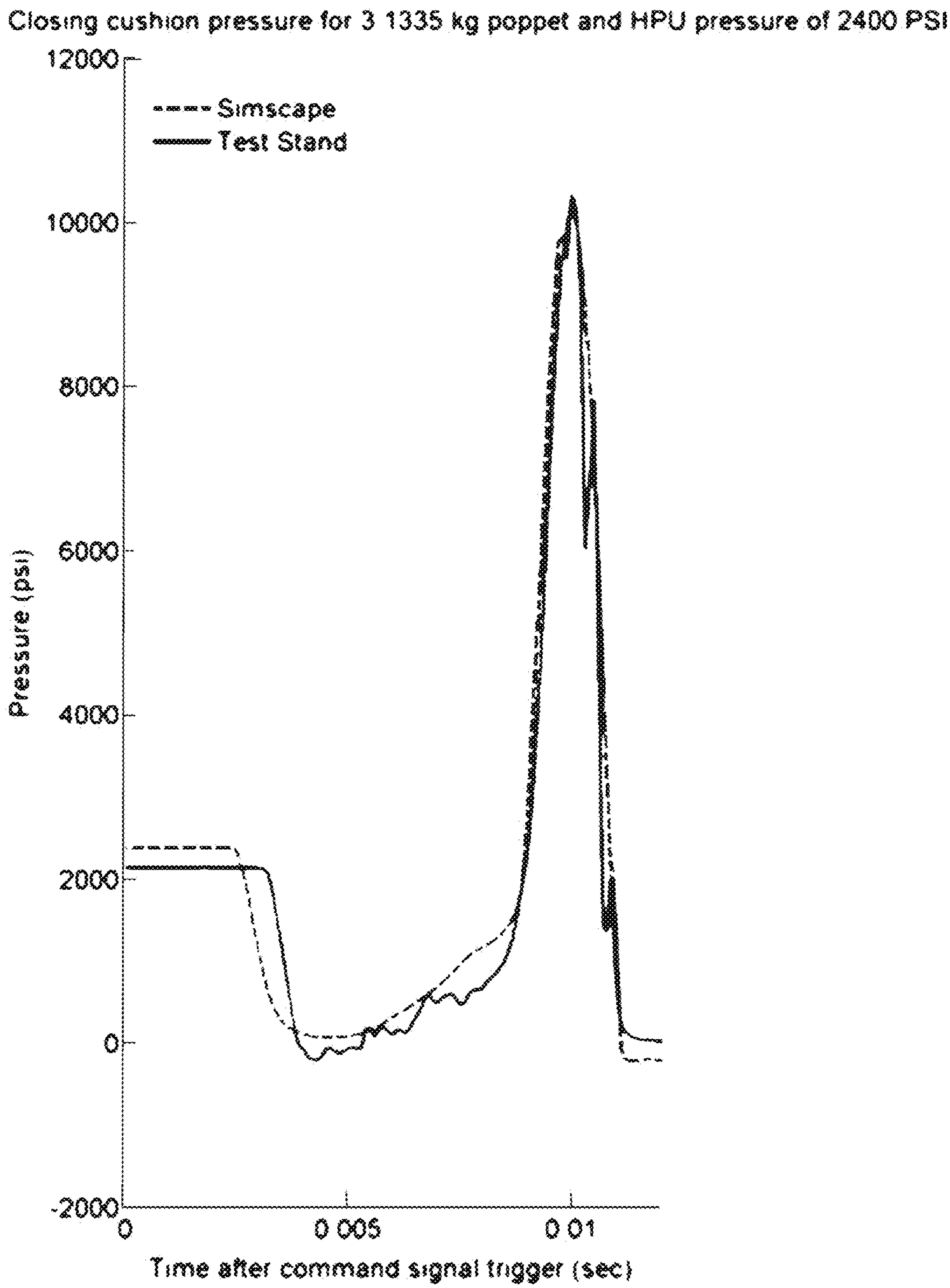

Closing cushion pressure for 3 1335 kg poppet and HPU pressure of 2400 PSI
Simscape
Test Stand
Pressure (psi)
12000
10000
8000
6000
4000
2000
0
-2000
0
0 005
0 01
Time after command signal trigger (sec)

FIG. 30A

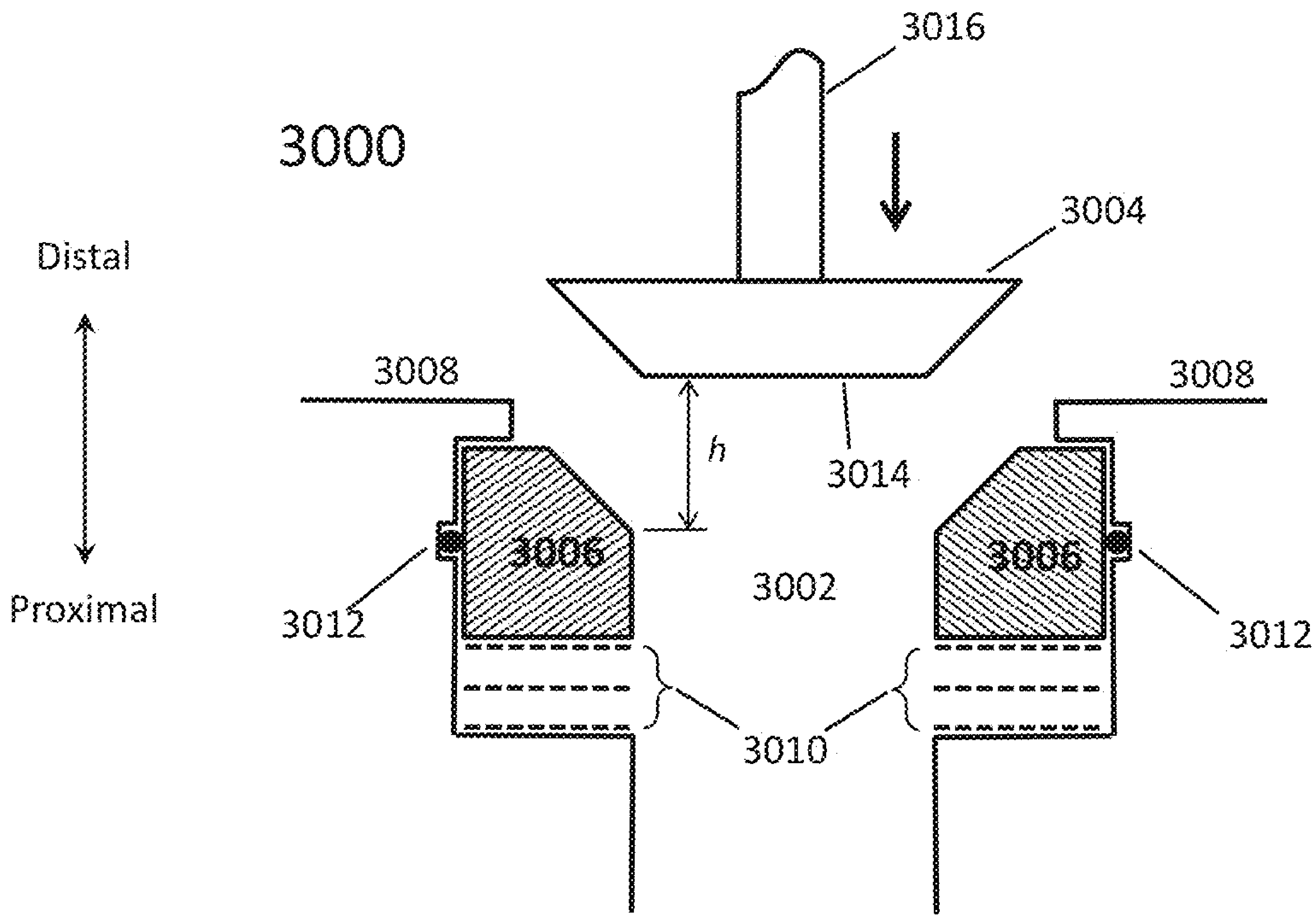


FIG. 30B

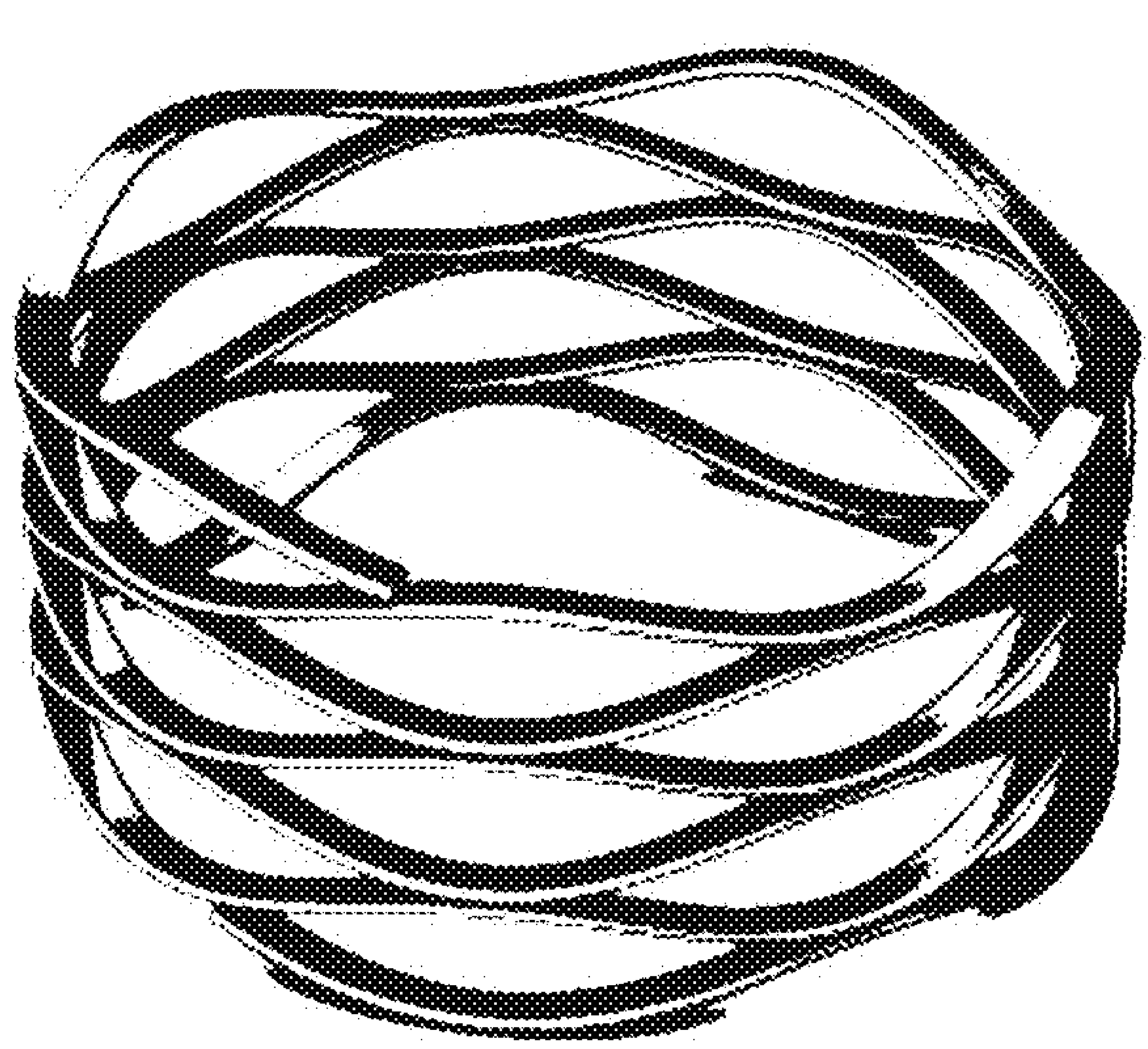


FIG. 30C

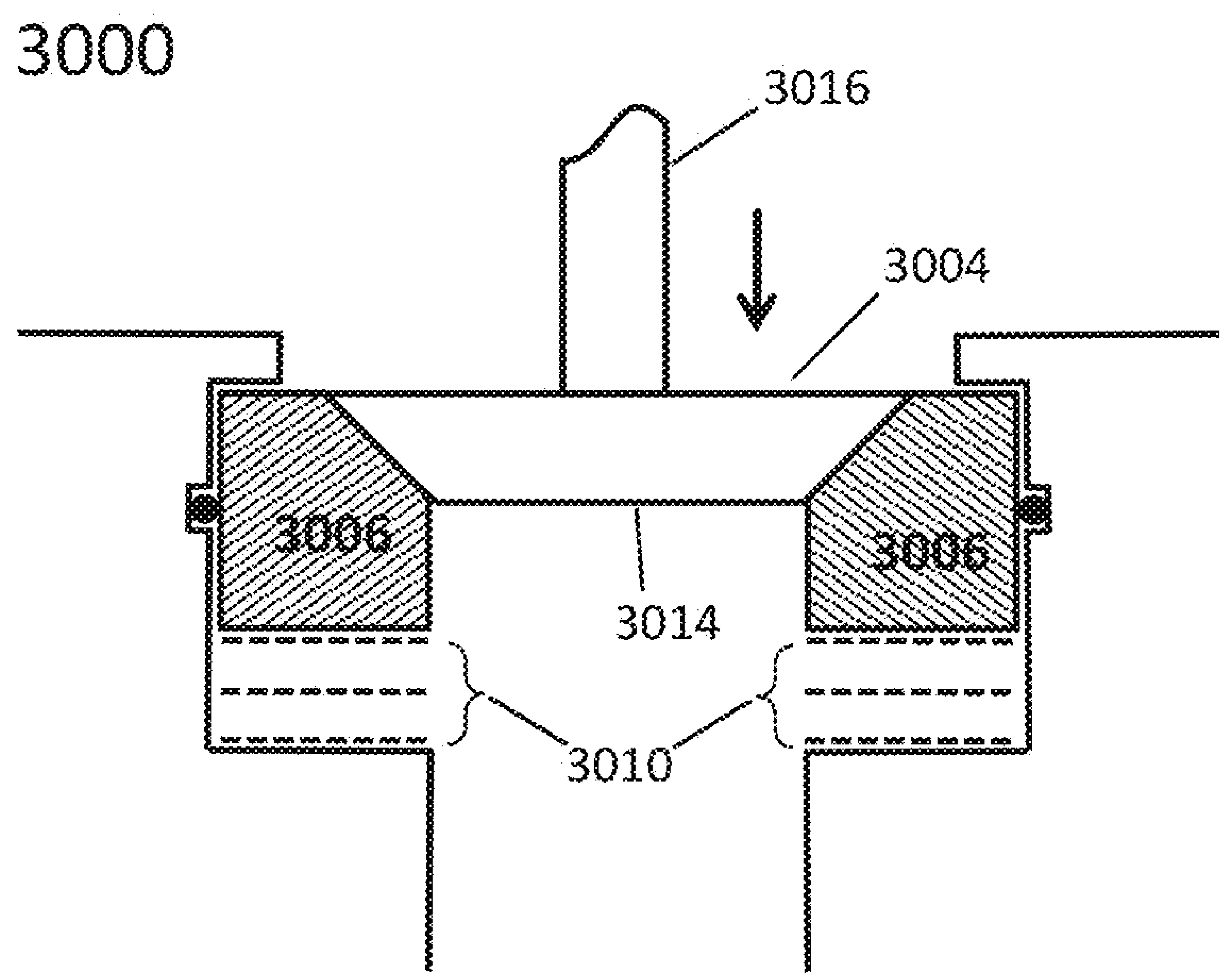
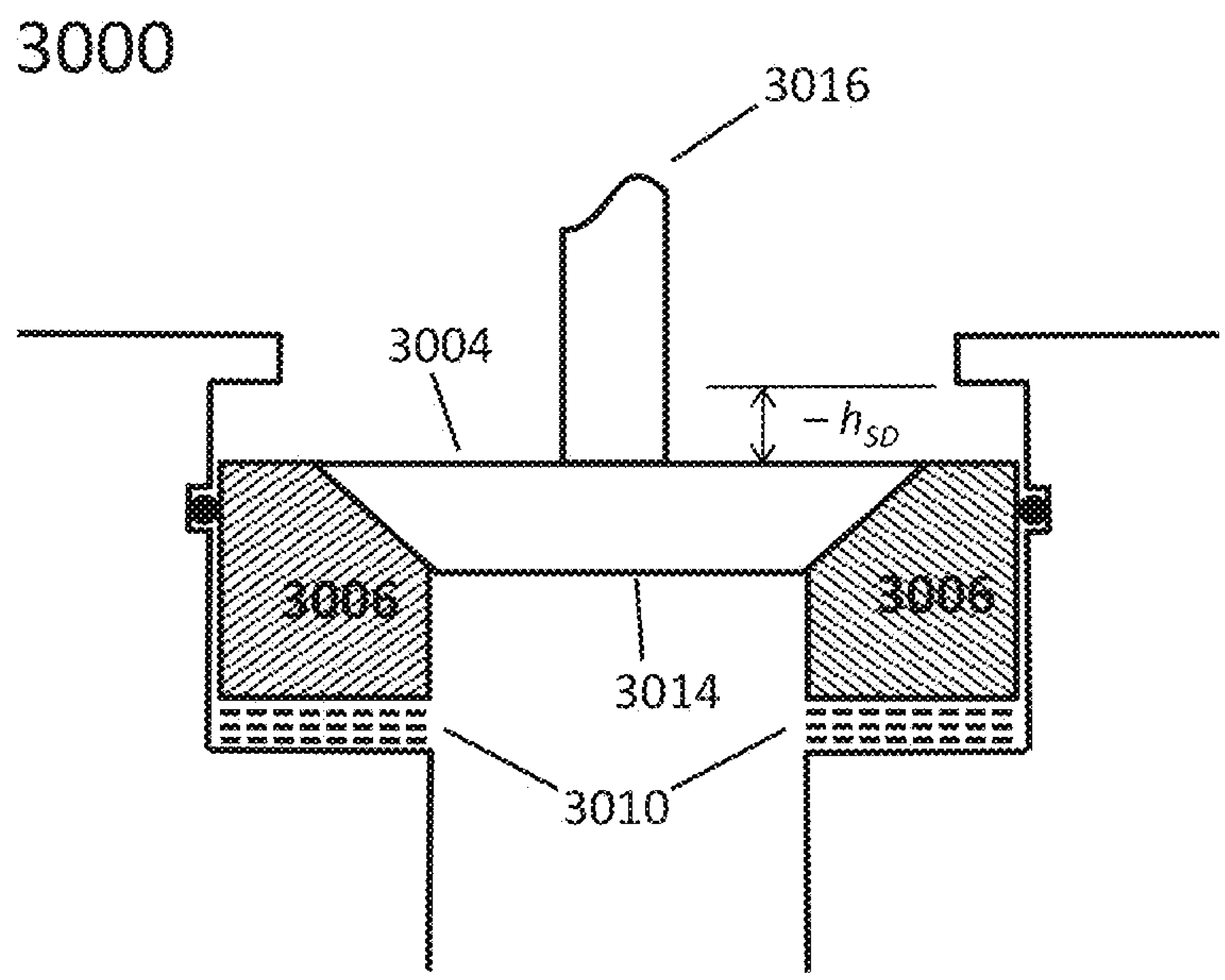


FIG. 30D



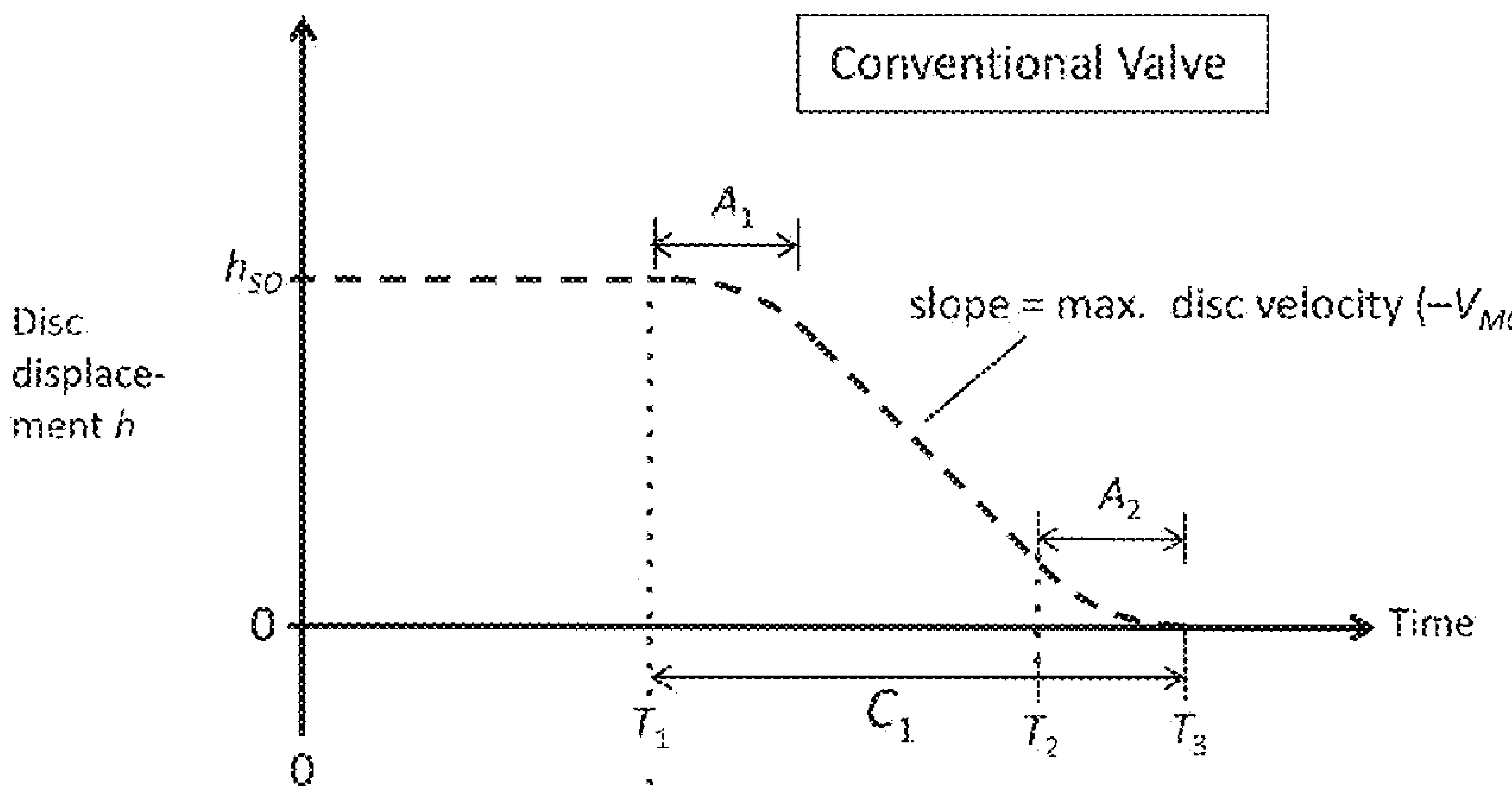


FIG. 31A

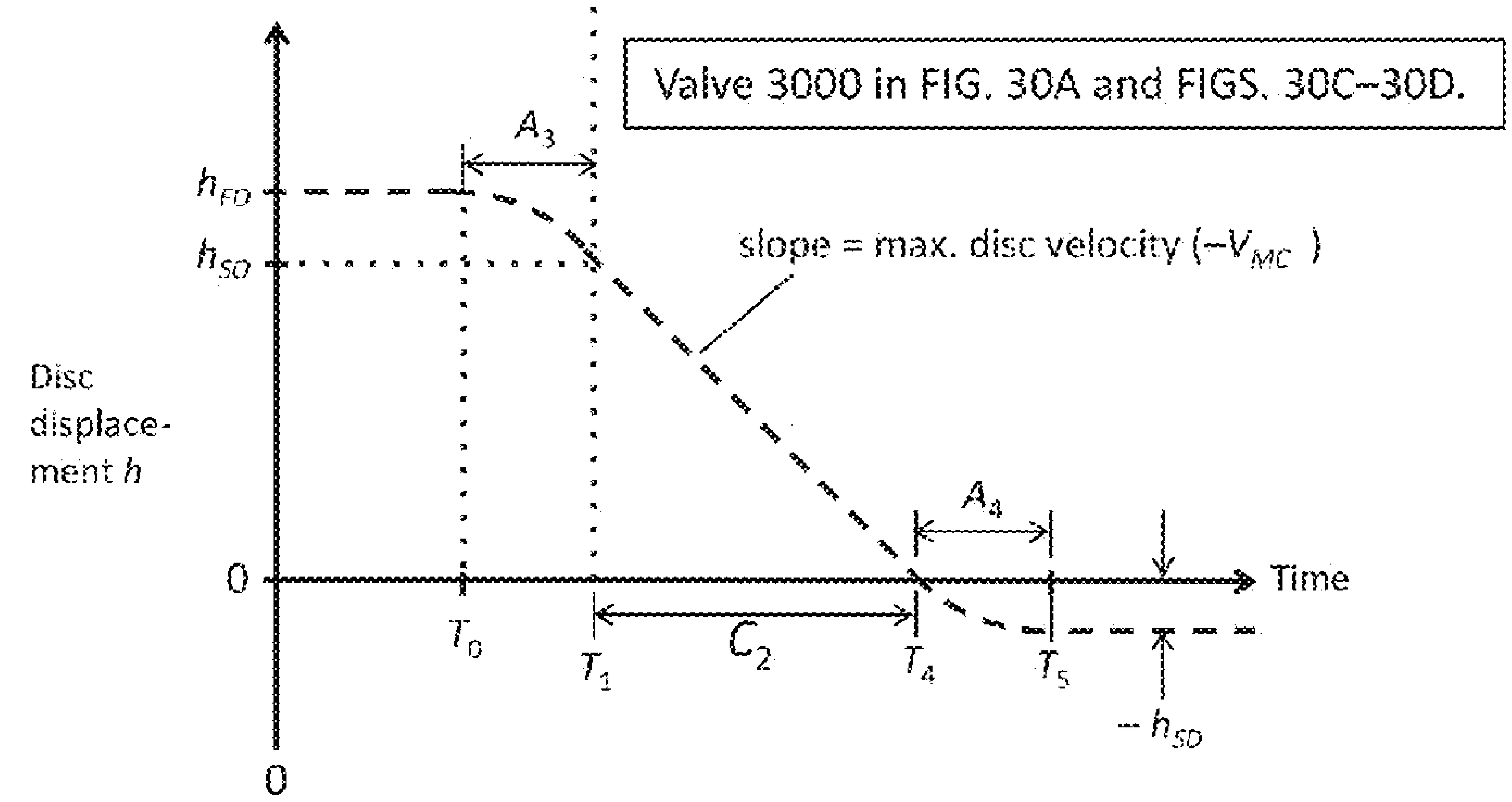


FIG. 31B

FIG. 32A

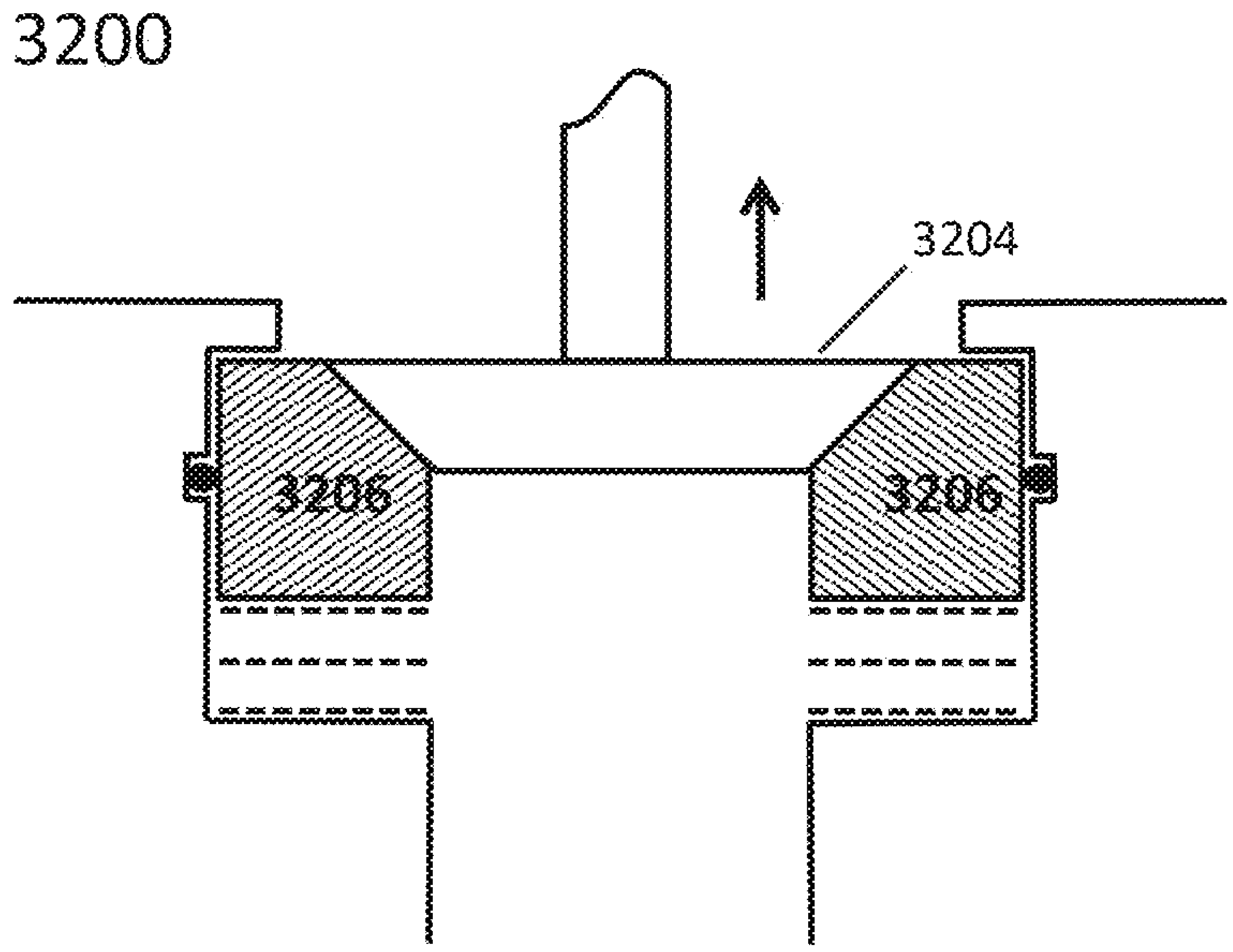


FIG. 32B

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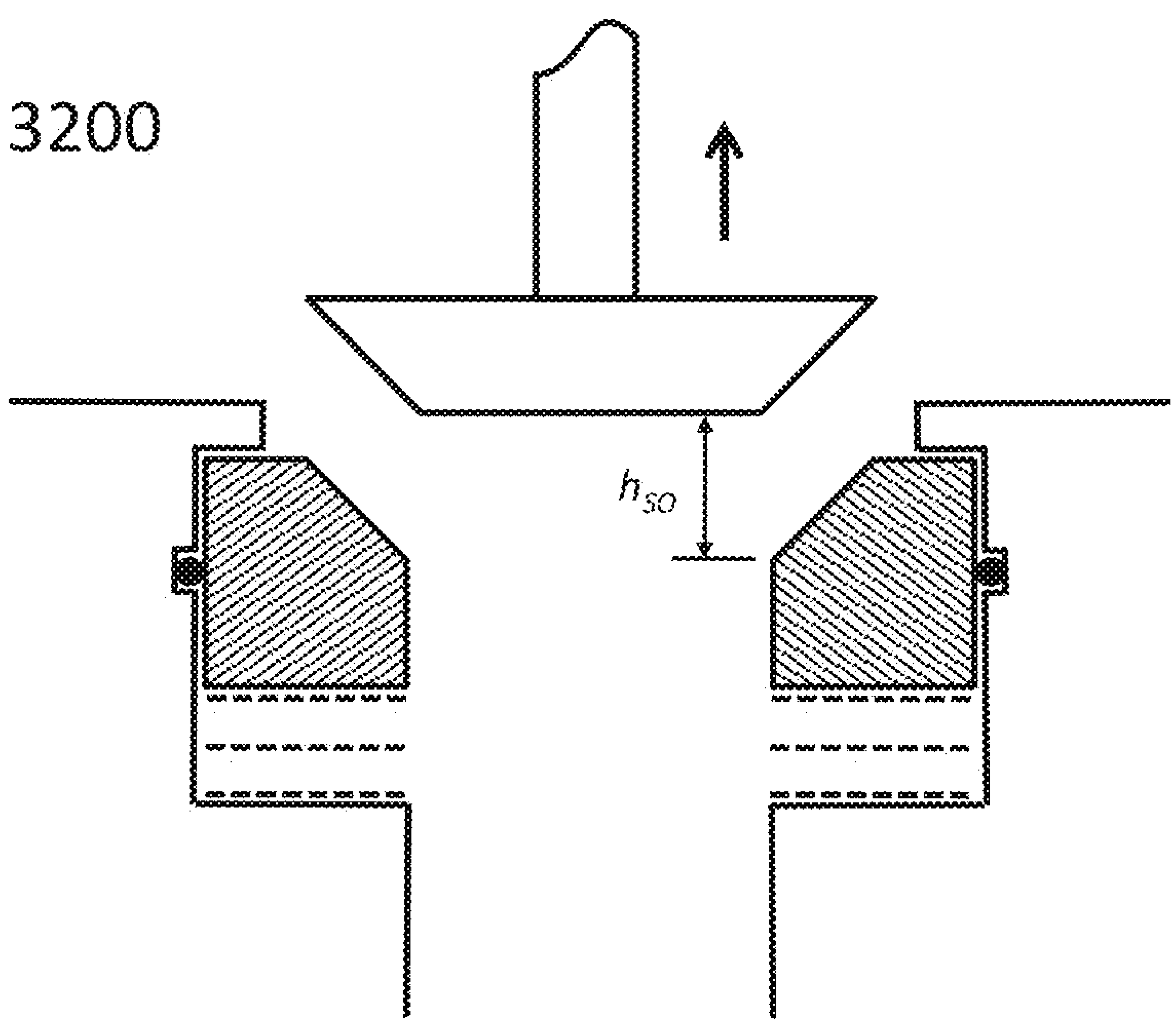

FIG. 32C

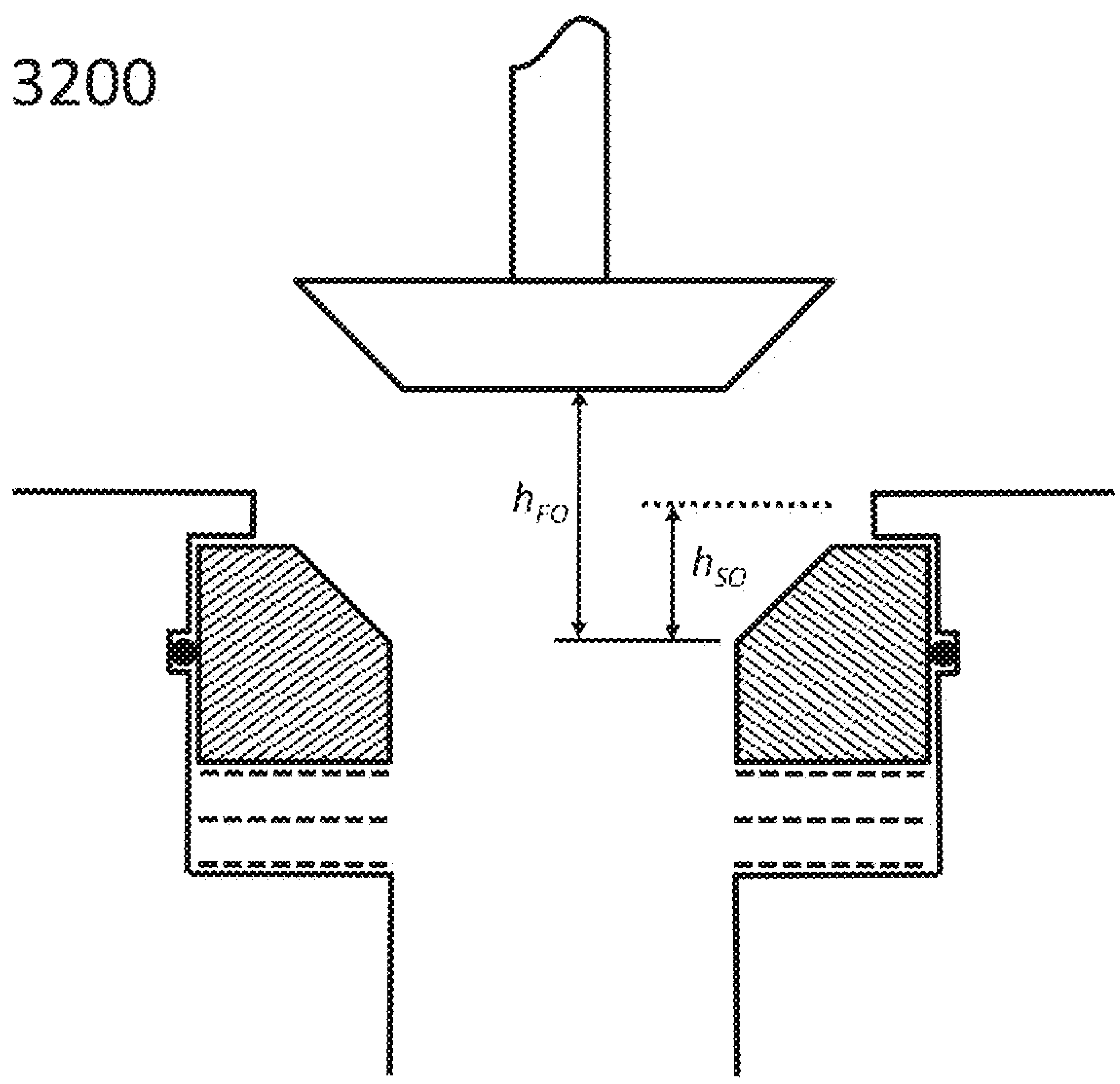

FIG. 33A

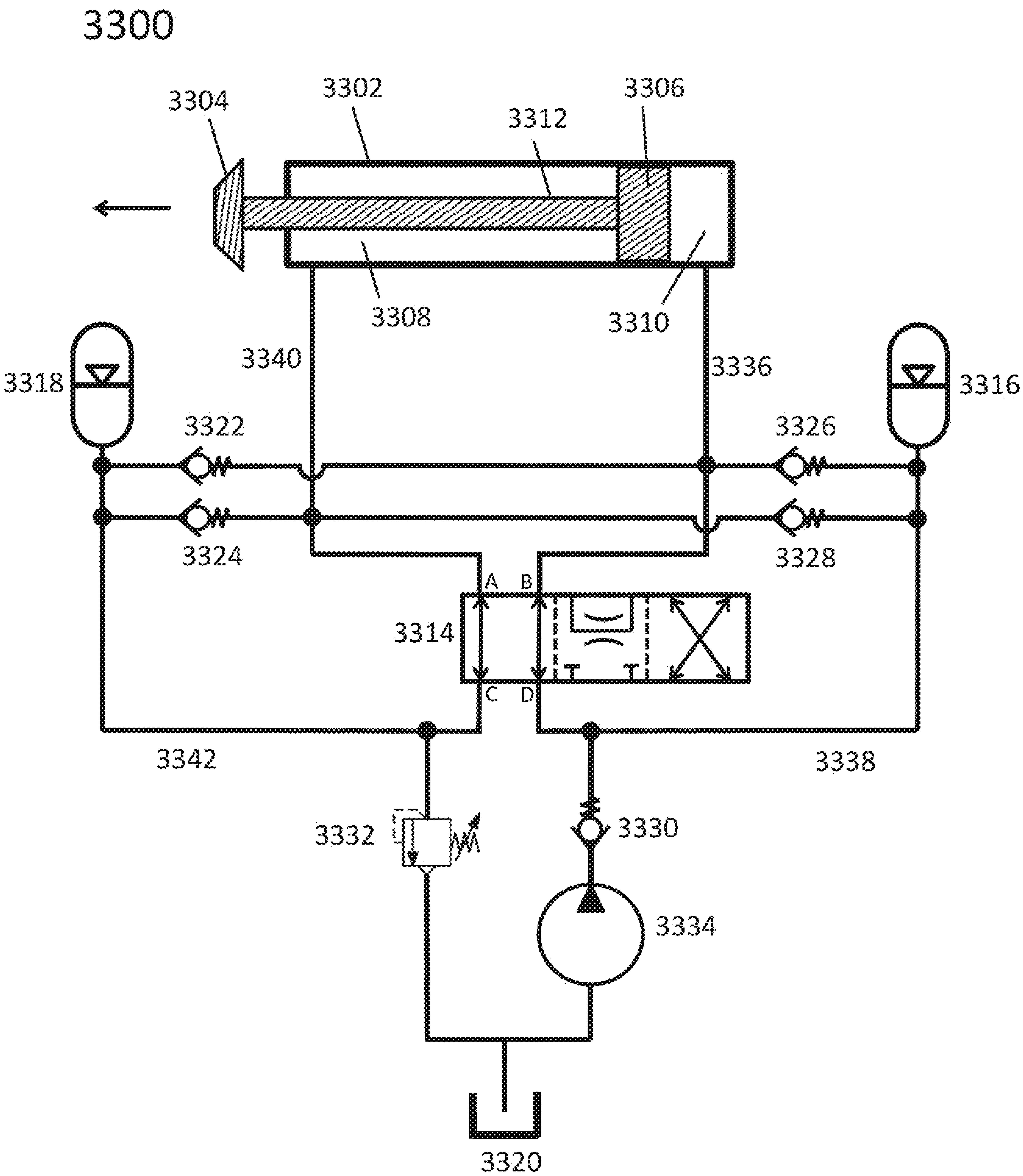
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FIG. 33B

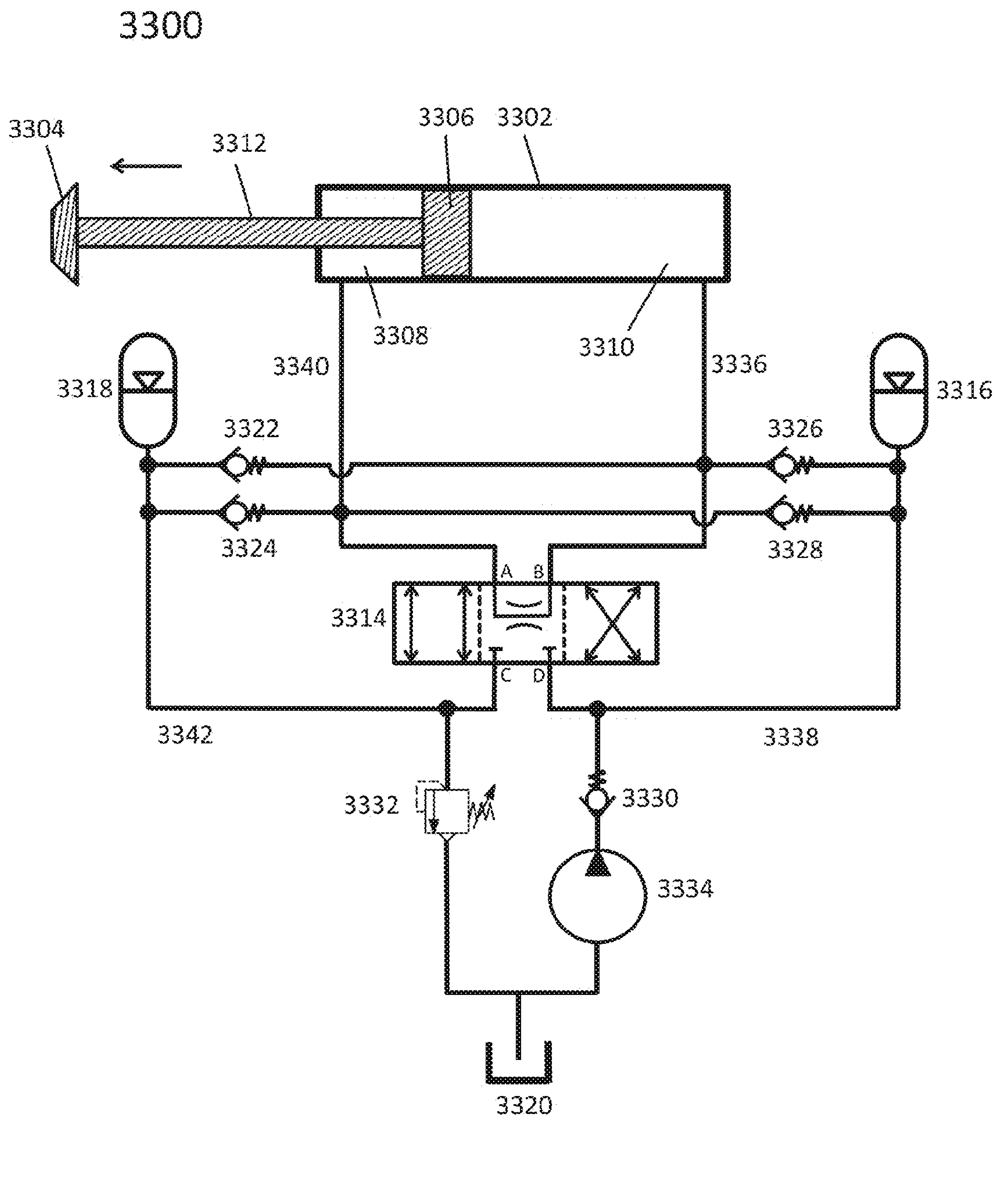
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FIG. 33C

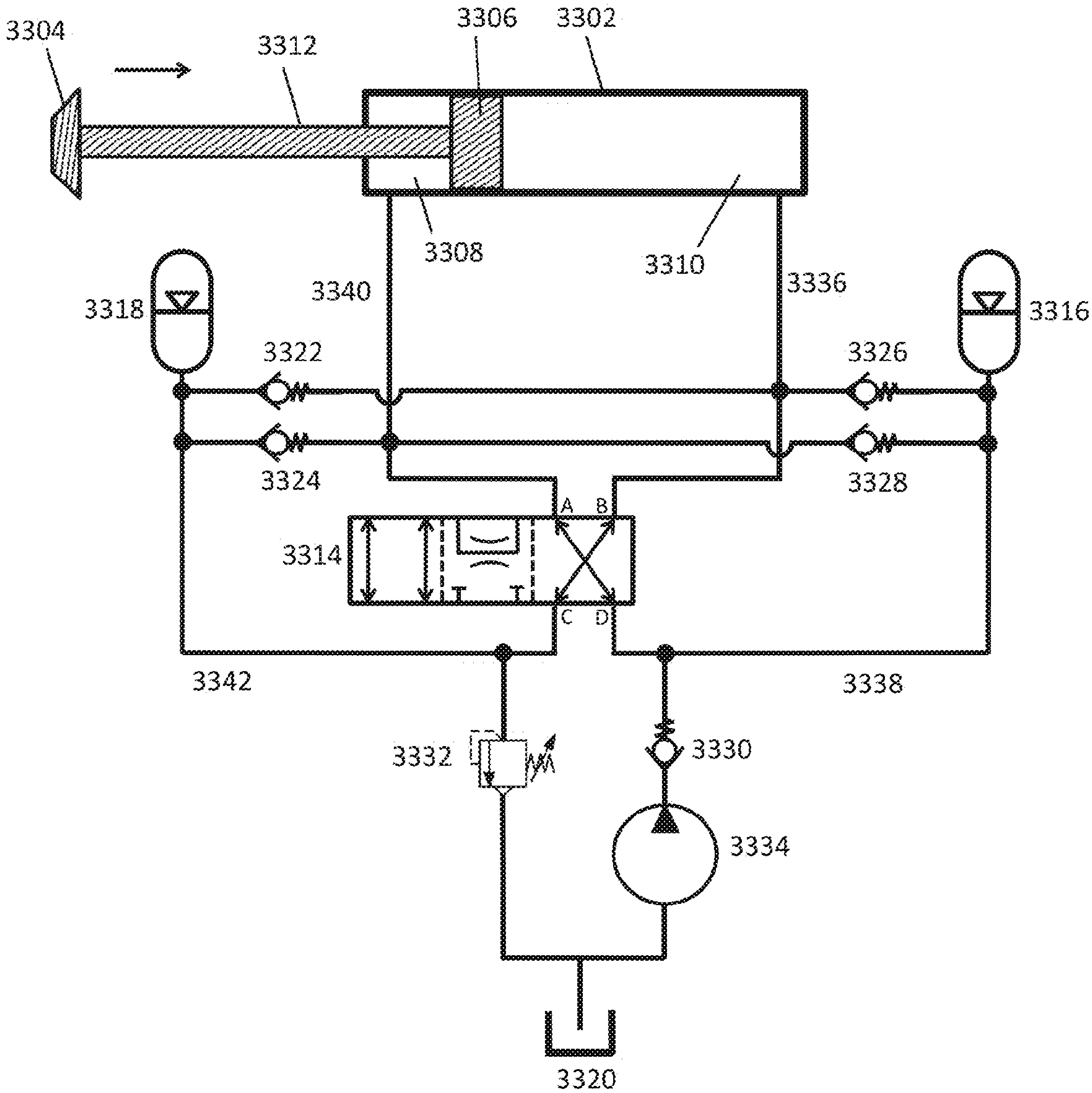
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FIG. 34A

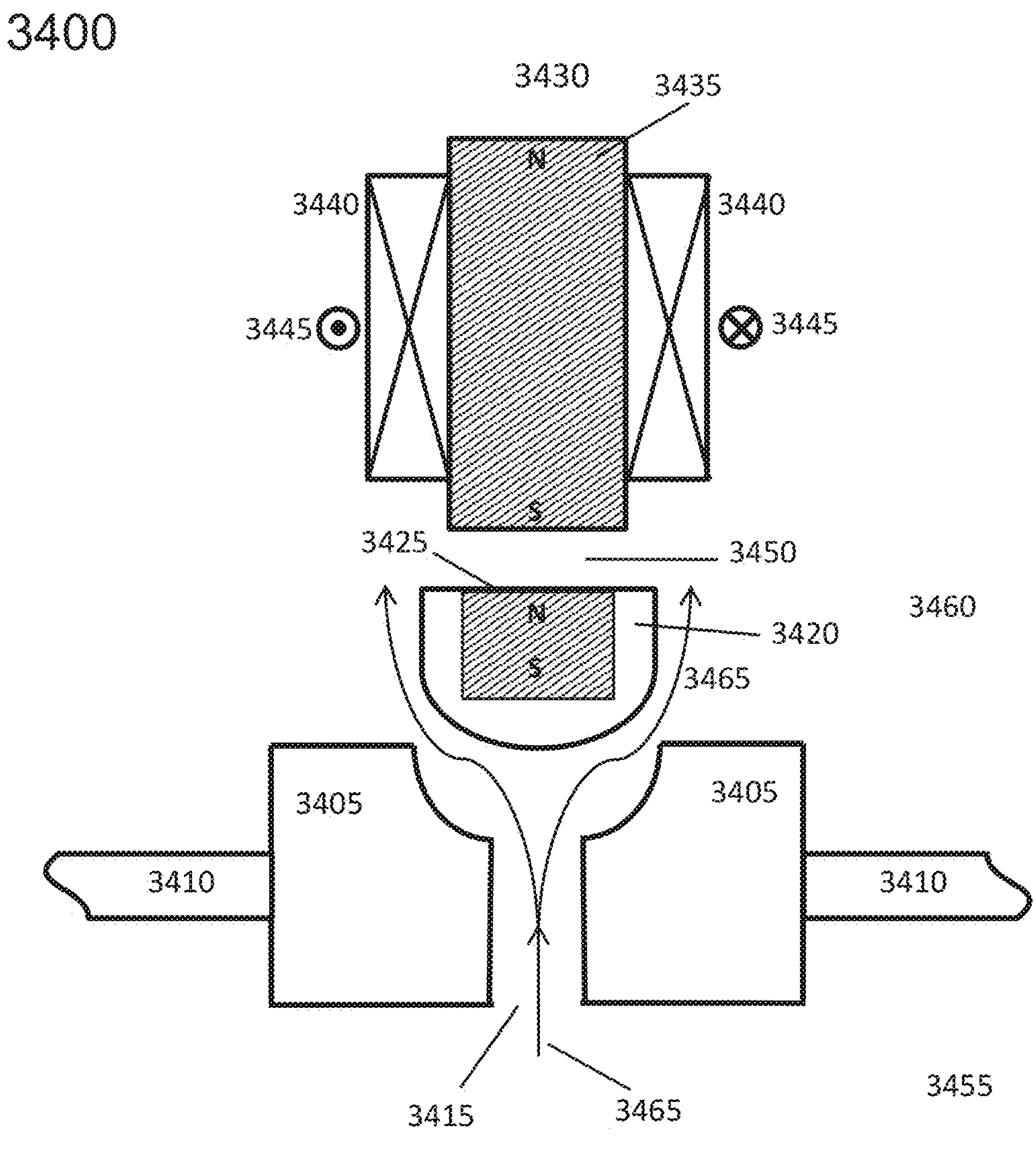
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FIG. 34B

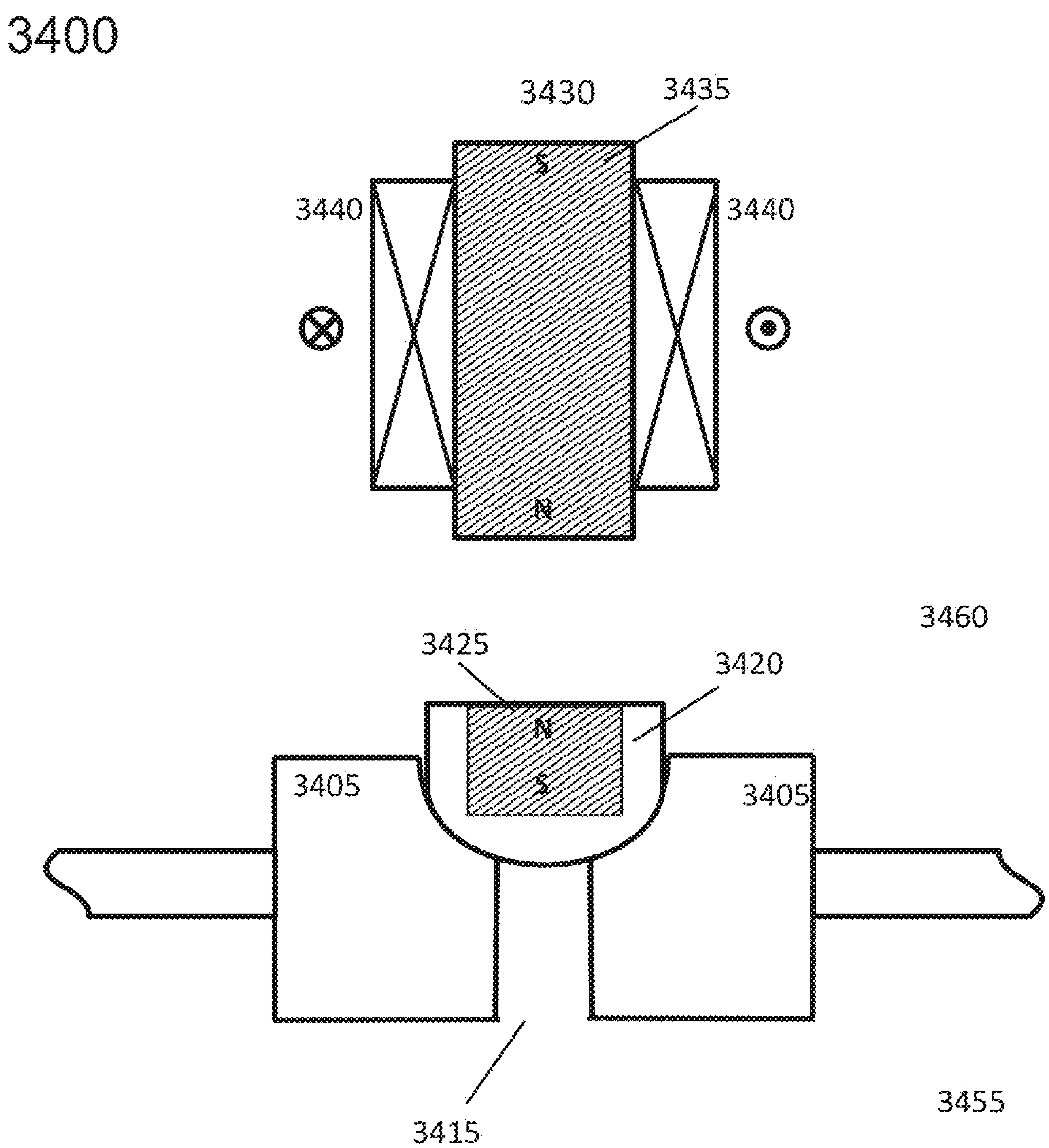
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FIG. 35A

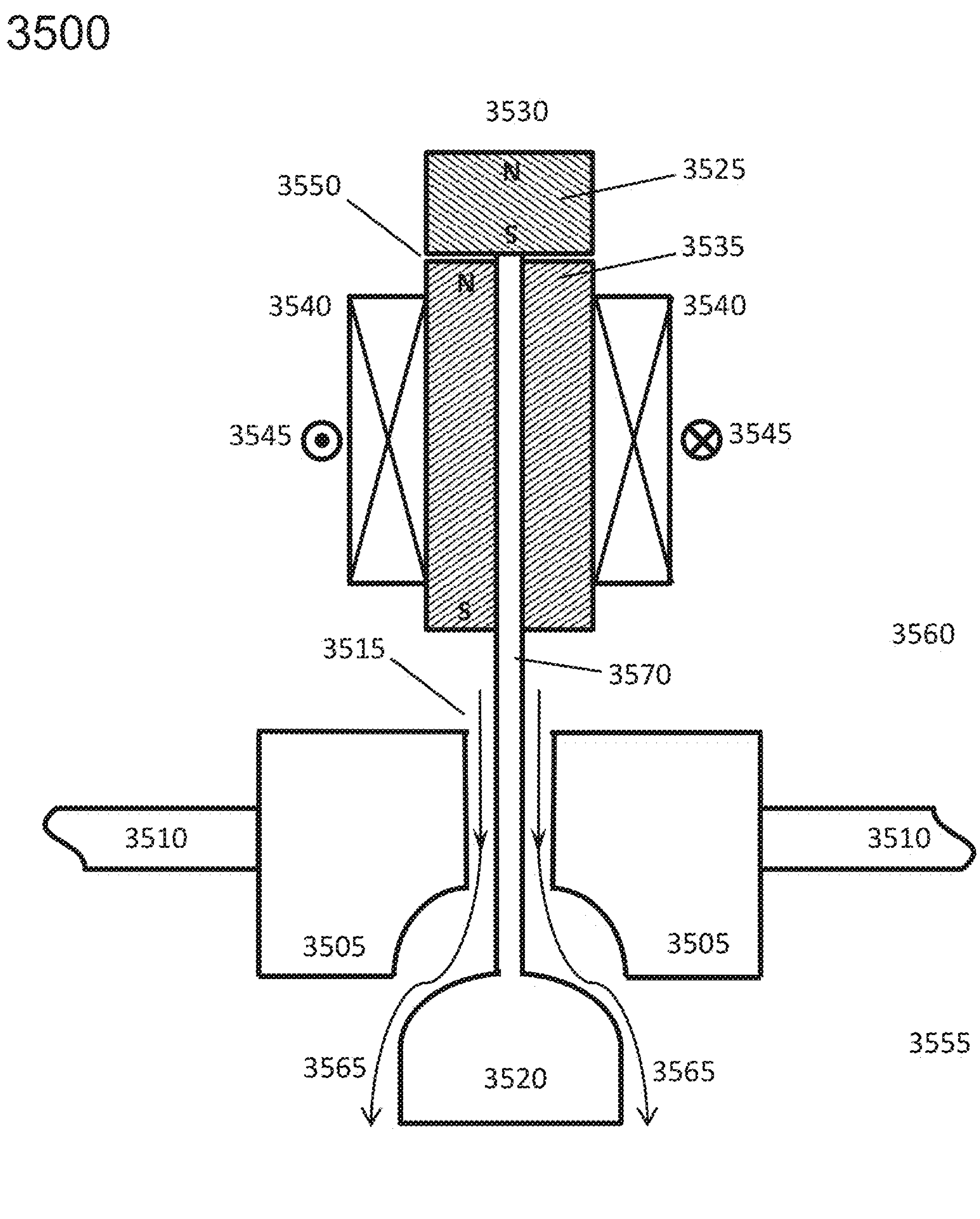
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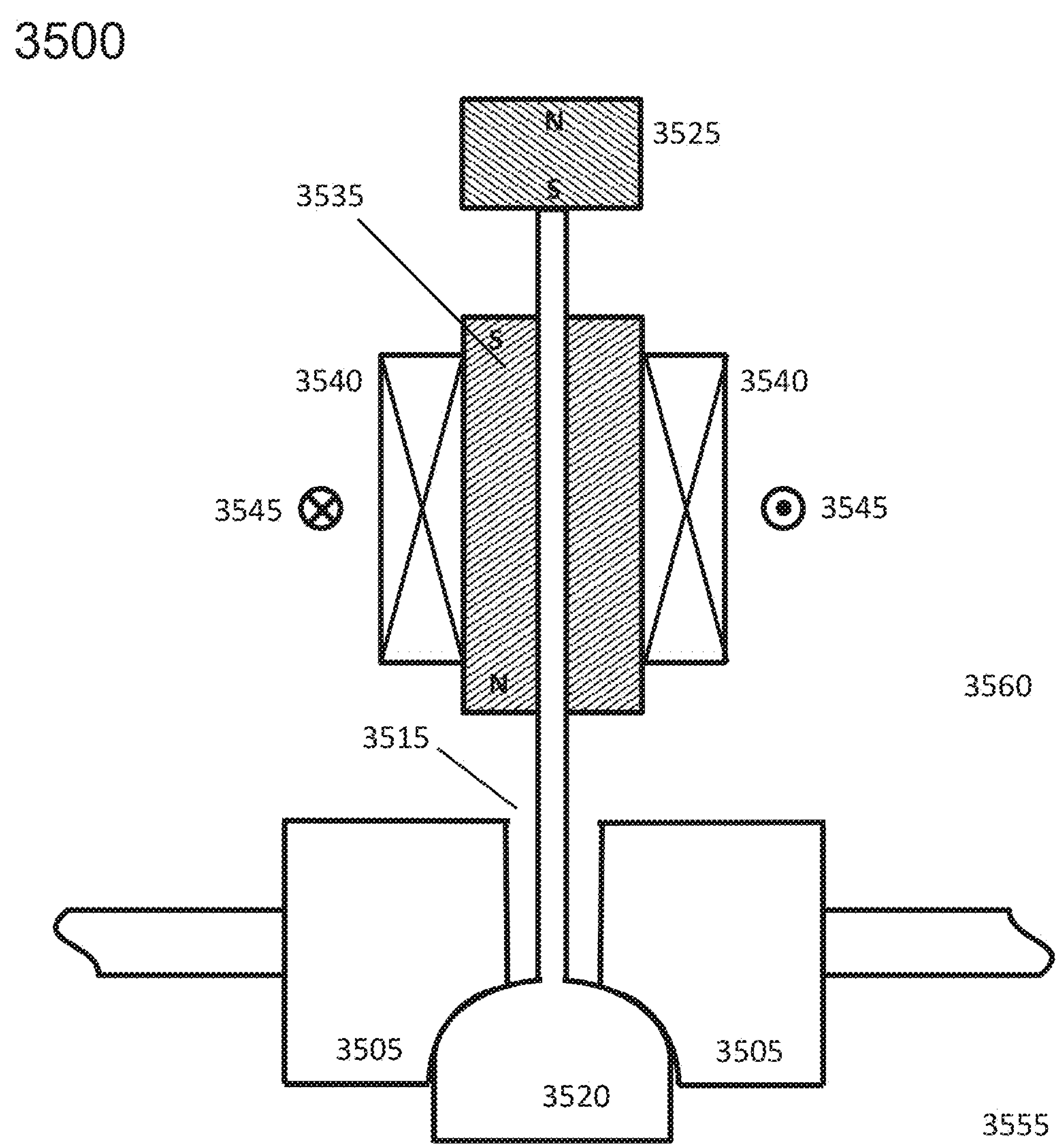
FIG. 35B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VALVE ACTIVATION IN COMPRESSED-GAS ENERGY STORAGE AND RECOVERY SYSTEMS

RELATED APPLICATIONS

[0001] This application claims the benefit of and priority to U.S. Provisional Patent Application No. 61/576,654, filed Dec. 16, 2011, U.S. Provisional Patent Application No. 61/614,045, filed Mar. 22, 2012, and U.S. Provisional Patent Application No. 61/620,018, filed Apr. 4, 2012. The entire disclosure of each of these applications is hereby incorporated herein by reference.

STATEMENT REGARDING FEDERALLY SPONSORED RESEARCH

[0002] This invention was made with government support under DE-OE0000231 awarded by the DOE. The government has certain rights in the invention.

FIELD OF THE INVENTION

[0003] In various embodiments, the present invention relates to pneumatics, hydraulics, power generation, and energy storage, and more particularly, to systems and methods using pneumatic, pneumatic/hydraulic, and/or hydraulic cylinders for energy storage and recovery.

BACKGROUND

[0004] Storing energy in the form of compressed gas has a long history and components tend to be well tested and reliable, and have long lifetimes. The general principle of compressed-gas or compressed-air energy storage (CAES) is that generated energy (e.g., electric energy) is used to compress gas (e.g., air), thus converting the original energy to pressure potential energy; this potential energy is later recovered in a useful form (e.g., converted back to electricity) via gas expansion coupled to an appropriate mechanism. Advantages of compressed-gas energy storage include low specific-energy costs, long lifetime, low maintenance, reasonable energy density, and good reliability.

[0005] If a body of gas is at the same temperature as its environment, and expansion occurs slowly relative to the rate of heat exchange between the gas and its environment, then the gas will remain at approximately constant temperature as it expands. This process is termed "isothermal" expansion. Isothermal expansion of a quantity of high-pressure gas stored at a given temperature recovers approximately three times more work than would "adiabatic expansion," that is, expansion where no heat is exchanged between the gas and its environment—e.g., because the expansion happens rapidly or in an insulated chamber. Gas may also be compressed isothermally or adiabatically.

[0006] An ideally isothermal energy-storage cycle of compression, storage, and expansion would have 100% thermodynamic efficiency. An ideally adiabatic energy-storage cycle would also have 100% thermodynamic efficiency, but there are many practical disadvantages to the adiabatic approach. These include the production of higher temperature and pressure extremes within the system, heat loss during the storage period, and inability to exploit environmental (e.g., cogenerative) heat sources and sinks during expansion and compression, respectively. In an isothermal system, the cost of adding a heat-exchange system is traded against resolving the difficulties of the adiabatic approach. In either case, mechanical energy from expanding gas is typically converted to electrical energy before use.

[0007] An efficient and novel design for storing energy in the form of compressed gas utilizing near isothermal gas compression and expansion has been shown and described in U.S. Pat. No. 7,832,207, filed Apr. 9, 2009 (the '207 patent) and U.S. Pat. No. 7,874,155, filed Feb. 25, 2010 (the '155 patent), the disclosures of which are hereby incorporated herein by reference in their entireties. The '207 and '155 patents disclose systems and techniques for expanding gas isothermally in staged cylinders and intensifiers over a large pressure range in order to generate electrical energy when required. Mechanical energy from the expanding gas may be used to drive a hydraulic pump/motor subsystem that produces electricity. Systems and techniques for hydraulic-pneumatic pressure intensification that may be employed in systems and methods such as those disclosed in the '207 and '155 patents are shown and described in U.S. Pat. No. 8,037,678, filed Sep. 10, 2010 (the '678 patent), the disclosure of which is hereby incorporated herein by reference in its entirety.

[0008] In the systems disclosed in the '207 and '155 patents, reciprocal mechanical motion is produced during recovery of energy from storage by expansion of gas in the cylinders. This reciprocal motion may be converted to electricity by a variety of techniques, for example as disclosed in the '678 patent as well as in U.S. Pat. No. 8,117,842, filed Feb. 14, 2011 (the '842 patent), the disclosure of which is hereby incorporated herein by reference in its entirety. The ability of such systems to either store energy (i.e., use energy to compress gas into a storage reservoir) or produce energy (i.e., expand gas from a storage reservoir to release energy) will be apparent to any person reasonably familiar with the principles of electrical and pneumatic machines.

[0009] In order to reduce overall pressure ranges of operation, various CAES systems may utilize designs involving multiple interconnected cylinders. In such designs, trapped regions of "dead volume" may occur such that pockets of gas remain in cylinders before and after valve transitions. Such volumes may occur within the cylinders themselves and/or within conduits, valves, or other components within and interconnecting the cylinders. Bringing relatively high-pressure gas into communication (e.g., by the opening of a valve) with relatively low-pressure gas within a dead volume tends to lead to a diminishment of pressure of the higher-pressure gas without the performance of useful work, thereby disadvantageously reducing the amount of work recoverable from or stored within the higher-pressure gas. Air dead volume tends to reduce the amount of work available from a quantity of high-pressure gas brought into communication with the dead volume. This loss of potential energy may be termed a "coupling loss." For example, during a compression stage a volume of gas that was compressed to a relatively high pressure may remain inside the compression cylinder, or conduits attached thereto, after the movable member of the cylinder (e.g., piston, hydraulic fluid, or bladder) reaches the end of its stroke. The volume of compressed air that is not pushed onto the next stage at the end of stroke constitutes "dead volume" (also termed, in compressors, "clearance volume"). If the volume of high-pressure gas within the cylinder is then brought into fluid communication (e.g., by the opening of a valve) with a section of intake piping, a portion of the high-pressure gas will tend to enter the piping and mix with the contents thereof, equalizing the pressure within the two volumes. This equalization of pressure entails a loss of exergy (i.e., energy available as work). In a preferred scenario, the gas in the dead volume is allowed to expand in a manner that performs useful work (e.g., by pushing on a piston), equalizing in pressure with the gas in the piping prior to valve opening. In another scenario, the gas in the dead volume is allowed to expand below the pressure of the gas within the intake piping, and pressure equalization takes place during valve transition (opening).

[0010] In another example of formation of dead space in a CAES system during an expansion procedure, if gas is to be introduced into a cylinder through a valve for the purpose of performing work by pushing against a piston within the cylinder, and a chamber or volume exists adjacent to the piston that is filled with low-pressure gas at the time the valve is opened, the high-pressure gas entering the chamber is immediately reduced in pressure during free expansion and mixing with the low-pressure gas and, therefore, performs less mechanical work upon the piston than would have been possible without the pressure reduction. The low-pressure volume in such an example constitutes air dead volume. Dead volume may also appear within the portion of a valve mechanism that communicates with the cylinder interior, or within a tube or line connecting a valve to the cylinder interior, or within other components that contain gas in various states of operation of the system. It will be clear to persons familiar with hydraulics and pneumatics that dead volume may also appear during compression procedures, and that energy losses due to pneumatically communicating dead volumes tend to be additive.

[0011] Moreover, in an expander-compressor system operated to expand or compress gas near-isothermally (i.e., at approximately constant temperature) within a cylinder, gas that escapes the cylinder to become dead volume (e.g., by displacing an incompressible fluid) in a hydraulic subsystem may, as pressures change within the system, expand and compress adiabatically (i.e., at non-constant temperature), with associated energy losses due to heat transfer between the gas and materials surrounding the dead volume. These thermal energy losses will tend to be additive with losses that are due to non-work-performing expansion of gas to lower pressures in dead volumes.

[0012] Therefore, in various compressor-expander systems, including isothermal compressor-expander systems, preventing the formation of dead volume will generally enable higher system efficiency. Attempts to minimize dead volume frequently involve reducing the sizes and lengths of conduits interconnecting cylinders and other components. However, such efforts may not eliminate all dead volume and tend to constrain the overall geometry and placement of individual system components. Therefore, there is a need for alternative or additional approaches to reducing dead volume and/or the deleterious effects of dead volume in pneumatic components in order to reduce coupling losses and improve efficiency during compression and/or expansion of gas.

[0013] Furthermore, the efficiency of compressed-gas energy-conversion systems may be limited by the valving systems that control flow of the gas (and/or other fluid) into and out of the cylinder and/or into, out of, or through other components. For example, conventional designs may entail valve arrangements that do not prevent contamination between actuation fluid and working fluid, that do not prevent damage from hydrolocking, that require excessive actuation energy, that require an excessive time to actuate (i.e., open and close), that have excessive pressure drops, that do not fail shut (e.g., that prevent the unwanted venting of high-pressure gas when power to valve actuation mechanisms fails), that contain dead space in piping, and that have other disadvantages. Designs that mitigate or eliminate such features will be tend to be advantageous. It will also be advantageous, in general, for valves to open and close rapidly, in order that as little fluid as possible may be passed through valves that are in partially open or closed states, as such passage entails throttling losses that decrease overall system efficiency.

[0014] Furthermore, the use of cam- or piston-actuated valves to control flow of the gas (and/or other fluid) into and out of the cylinder may also entail disadvantages that can be surmounted or mitigated by novel design. Conventional or prior-art reciprocating piston-type expanders typically use cam- or piston-actuated valves to control admission of fluid from a high-pressure source into the cylinder when the piston is near top dead center (i.e., the piston's closest point of approach to the end of the cylinder designated the "top") and discharge of fluid from the cylinder when the piston is near bottom dead center (i.e., the piston's closest point of approach to the end of the cylinder opposite the top end and designated the "bottom"). Conventional reciprocating piston-type compressors typically use passive check-style valves to control admission of fluid into the cylinder during the intake (or suction) stroke and expulsion of fluid from the cylinder during the discharge stroke. These conventional valving techniques for reciprocating piston-type expanders and compressors may not be optimal for use in compressed-air energy storage systems; for example, cam- or piston-actuated valves, with their fixed timing, may not be optimal for variable-speed, constant-power expansion of gas from a source (e.g., gas-storage reservoir) the pressure of which is declining. Typically, more desirable are intake and outlet valves that may operate passively in compression mode, i.e., based solely on cylinder pressure, and that may also be actively or semi-actively controlled during expansion mode (e.g., the timing of the valve's operation may be set by an operator or control system). It may also be desirable that the intake and outlet valves check closed in certain operating conditions and check open in other operating conditions. For example, the intake valve in its passive or unpowered state should prevent high-pressure air from flowing from the high-pressure store into the cylinder when the pressure within the cylinder is lower than the pressure within the high-pressure store, but should allow air from the cylinder to flow into the high-pressure store when the pressure within the cylinder is higher than the pressure within the high-pressure store. Similarly, the outlet valve in its passive or unpowered state should prevent high-pressure air from venting from the cylinder to an environmental vent or other secondary volume when the pressure within the cylinder is higher than the pressure in the secondary volume, but should allow air from the secondary volume to flow into the cylinder when the pressure within the cylinder is lower than the pressure within the secondary volume. Therefore, opportunities exist to improve the overall efficiency of energy storage-and-recovery systems via further enhancements to valve design and valve-actuation efficiency.

SUMMARY

[0015] Embodiments of the invention reduce the impact of dead volume in pneumatic cylinders and/or pneumatic chambers of pneumatic/hydraulic cylinders during compression and/or expansion in CAES systems; increase the efficiency with which fluid (i.e., gas, liquid, or a mixture of gas and liquid) may be admitted to or exhausted from a pneumatic or pneumatic-hydraulic cylinder that is part of an energy-conversion system; and improve the performance of an energy-conversion system by employing one or more valves that may be integrated into the head of a cylinder and that typically (i) use differential pressure to open, (ii) use electromagnetic force to hold open, and (iii) check closed.

[0016] In various embodiments, the impact of dead volume is reduced by time-coordinated matching of gas pressures within system components that would, absent such matching, suffer coupling losses and potential equipment damage. Herein, a space within any component of a CAES system is termed a "dead volume" or "dead space" if its volume cannot, in some or all states of operation, be reduced to zero due to mechanical constraints (e.g., imperfect fit of a piston to the interior face of a cylinder head when the piston is at top dead center, forming an ineradicable, residual chamber volume) and if in some states of system operation the space contains gas at a pressure that can be brought into fluid communication with gas at a significantly different pressure (e.g., through a valve transition).

[0017] In cylinders, the time-coordinated matching of pressures may be accomplished using actuated valves that are selectively closed and opened in a manner that yields approximately matched pressures within system components about to be brought into fluid communication with each other. To reduce loss of exergy due to non-work-performing expansion of gas when components containing gas at relatively high pressure (e.g., 3,000 psi) are brought into fluid communication with components containing gas at relatively low pressure (e.g., 300 psi or atmospheric pressure), the gas in one or more potential dead spaces is pre-compressed to a pressure approximately equal to that of the higher-pressure gas before the higher- and lower-pressure gas volumes are brought into fluid communication with each other. In other embodiments, the gas in one or more potential dead spaces is pre-expanded to a pressure approximately equal to that of the lower-pressure gas before the higher- and lower-pressure volumes are brought into fluid communication with each other. Such pre-compression and pre-expansion produce specific target pressures (e.g., 3,000 psi) at specific times or in specific states of system operation (e.g., when a cylinder piston reaches top-dead-center position and is poised to begin an expansion stroke). Both target pressure and timing of pressure matching may be altered adaptively during the course of system operation based on measurements of pressures within various parts of the CAES system and/or other aspects of system state. An actuated valve may be operated (i.e., opened or closed) at specific times in order to reduce the effect of dead space, e.g., a valve may be opened only when a pre-compression condition is met. The timing of actuated valve operation may, furthermore, be conditioned on feedback in order to provide increased system energy efficiency and/or other advantages. Herein, an "actuated" valve is a valve whose opening or closing occurs at a time that may be altered, either arbitrarily or within limits, by a system operator or control mechanism, as distinct from a passive "check"-type valve whose opening and closing are determined by differential pressures or a "cam-driven" valve whose times of opening or closing are dictated mechanically. Actuated valves may improve performance by opening at times different than would be entailed by operation of check valves by differential pressures. Variable-timing cam-driven valves may be considered actuated valves and are within the scope of this invention.

[0018] In certain embodiments of the invention, the CAES system may include or consist essentially of a cylinder assembly (or plurality of cylinder assemblies, e.g., multiple stages) that features a movable internal member (e.g., piston) or other boundary mechanism such as hydraulic fluid or a bladder. The internal boundary mechanism of the cylinder divides the interior of the cylinder into two chambers that may contain distinct bodies of fluid, and these may be at different pressures in various states of operation of the system. The system may further include a first control valve in communication with a high-pressure storage reservoir and the cylinder assembly, a second control valve in communication with the cylinder assembly and a vent to atmosphere, a heat-transfer subsystem in fluid communication with the cylinder assembly, an electric motor/generator in mechanical communication with a drive mechanism (e.g., crankshaft, hydraulic pump, linear generator mover) configured to drive the movable member disposed within the cylinder assembly, and a control system configured to operate the first and second control valves based on various information characterizing various aspects of the cylinder assembly and/or other components of the system (e.g., pressure, temperature, piston position, piston velocity).

[0019] One aspect of the invention relates to a method for reducing coupling losses and improving system performance during an expansion stage of a CAES system. In various embodiments, gas within a first chamber of a cylinder is pressurized to approximately some relatively high pressure (e.g., 3,000 psi) at or near the beginning of an expansion stroke of the cylinder. In this state of operation, the piston of the cylinder is at or near its top dead center position and the first chamber constitutes dead volume. A first control valve is then operated to place a volume of high-pressure gas (e.g., air at 3,000 psi) from an external source (e.g., a pressurized gas storage reservoir) in fluid communication with the first chamber. Because the gas within the first chamber is at approximately the same pressure as the high-pressure source placed in communication with the first chamber by the opening of the first control valve, gas from the high-pressure source does not tend to expand into the first chamber suddenly and without performing useful work. Coupling losses during the connection of the source to the cylinder are thus reduced or eliminated. In short, system performance may be improved by forestalling events of rapid pressure equalization of connecting spaces.

[0020] During a subsequent cylinder expansion stroke (also herein termed a "downward stroke"), useful work is recovered from high-pressure gas during both (1) admission of high-pressure gas to the first cylinder chamber while the boundary mechanism moves downward so as to allow the first cylinder chamber to enlarge, a phase of operation herein termed a "direct-drive phase" or "direct drive," and (2) a subsequent expansion phase (i.e., after the first control valve is closed) during which the boundary mechanism continues to move downward and a fixed mass of high-pressure gas expands in the enlarging first chamber). As shown and described in the '207 and '155 patents and U.S. patent application Ser. No. 13/473,128, filed May 16, 2012 (the '128 application), the entire disclosure of which is incorporated by reference herein, gas expansion may be maintained as substantially isothermal by introducing a certain volume of liquid (e.g., a quantity of foam or spray) at an appropriate temperature into the cylinder prior to and/or during the expansion.

[0021] At or near the end of the expansion stroke, when the gas reaches a lower pressure (e.g., 300 psi), a second control valve is operated to begin to exhaust the gas (e.g., to a vent and/or to a mid-pressure vessel and second cylinder assembly) as an upward stroke of the movable member within the piston occurs. During the first portion of the second half of the cylinder stroke (e.g., the upward stroke), the gas is exhausted through the second control valve (e.g., into a mid-pressure vessel and second cylinder assembly) by translating the movable member (e.g., piston) or other boundary mechanism to reduce the volume of the first chamber in the cylinder assembly. During a second portion of the second half of the cylinder stroke (e.g., the upward stroke), prior to the movable member reaching the end of stroke (e.g., top of stroke) inside the cylinder, the second control valve is closed and a "pre-compression stroke" is performed to compress the remaining volume of air (dead volume) and/or liquid inside the cylinder.

[0022] The time of closure of the second control valve, relative to the sequence of states of operation just described, is not arbitrary. Premature closure of the second valve will typically tend to trap an excessive quantity of gas in the first chamber, resulting in overpressurization of the gas in the first chamber when the volume of the first chamber attains a minimum (i.e., at top dead center of stroke). When this occurs, opening of the first valve at the start of the next expansion cycle will result in energy loss through non-work-performing expansion of the gas within the first chamber into the high-pressure storage reservoir and other components (e.g., piping) in fluid communication therewith.

[0023] On the other hand, tardy closure of the second valve will tend to trap an inadequate quantity of gas in the first chamber, resulting in underpressurization of the gas in the first chamber when the volume of the first chamber attains a minimum (i.e., at top dead center of stroke). When this occurs, opening of the first valve at the start of the next expansion cycle will generally result in energy loss through non-work-performing expansion of gas from the high-pressure storage reservoir and other components (e.g., piping) in fluid communication therewith into the first chamber.

[0024] Therefore, in certain embodiments of the invention the optimal time of actuation of the second control valve is based at least in part on sensed conditions in one or more portions of the CAES system, e.g., pressure in the first chamber, pressure in the high-pressure storage reservoir, piston position, piston velocity, etc. The principles on which the time of actuation of the second control valve is determined will be made clear hereinbelow.

[0025] When liquid is introduced into the first chamber to enable approximately isothermal expansion, a quantity of liquid will generally accumulate in the first chamber (e.g., on top of the piston or other movable member) during an expansion stroke. In an ideal case, i.e., if all of the gas compressed in the first chamber is passed to the storage reservoir or to a higher-pressure cylinder stage by the time the volume of the first chamber is at a minimum, the remaining volume of the first chamber (i.e., the volume between the movable member and the interior face of the upper end-cap of the cylinder) will be occupied entirely by liquid; all gas will have been expelled and there will be no gas-filled dead volume in the first chamber at the commencement of a new expansion stroke. However, in practice, the first chamber volume at the commencement of a new expansion stroke will tend to contain both liquid and gas remaining from the previous expansion stroke. The gas fraction of this volume may constitute dead volume at the start of the new stroke. During the pre-compression stroke, therefore, as already described hereinabove, the effective coupling loss due to this dead volume is minimized by compressing the remaining air to a pressure substantially equal to the pressure of the air in the storage reservoir (or the pressure of the gas to be introduced into the cylinder for expansion, if such gas is not arriving directly from the storage reservoir). Thus, when additional high-pressure gas is admitted to the cylinder for expansion, no or minimal pressure difference exists between the two sides of the first control valve. This allows the first control valve to be operated with a lower actuation energy, further improving system efficiency.

[0026] Most or even substantially all of the work done upon the air in the first chamber during a pre-compression stroke is typically recovered as the air re-expands during the subsequent expansion stroke. Furthermore, if the pressure of the air in the dead volume is approximately equal to the pressure of the air in the storage vessel or the next higher-pressure stage, then there will be substantially no coupling loss when the first valve to the storage reservoir or next-higher-pressure stage is opened. The higher pressure within the dead volume entails less gas flow from the storage reservoir or next-higher-pressure stage to the cylinder when the valve is opened during an expansion stage, thereby reducing coupling loss and improving efficiency. Moreover, the longevity of some system components may be increased because transient mechanical stresses caused by high-pressure air rushing suddenly into dead volume are minimized or eliminated.

[0027] Employing measurements of pressures within various components (e.g., lines and chambers) allows the timing of actuated-valve closings and openings (i.e., valve transition events) to be optimized for specific system conditions. For example, CAES systems constructed according to similar designs may differ in pipe lengths and other details affecting potential dead space. In such a case, with actuated valves, valve transition events may be tuned to optimize efficiency of an individual system by minimizing dead-space coupling losses. For example, if an overpressure is detected in a pre-pressurized cylinder chamber, closure of the valve that permits gas to exit the chamber during a return stroke may be retarded to reduce the amount of pre-pressurized gas. A CAES system in which valve transition event tuning is performed by a computerized system controller may be considered, in this respect, a self-tuning system. For another example, as the pressure within a gas storage reservoir declines as the gas within the reservoir cools or is exhausted (or increases as the gas within the reservoir warms or is augmented), valve transition event timing may be tuned in a manner that adaptively, continuously maximizes the energy efficiency of the CAES system. Thus, a CAES system in accordance with various embodiments of the invention may adaptively self-tune its valve transition events so as to minimize dead-space coupling losses in response both to idiosyncrasies of system construction and to changing conditions of operation.

[0028] Furthermore, to minimize the impacts of dead space, the timing of valve actuations may be chosen in light of the non-ideal features of actual valves. Non-instantaneous valve transitions tend to entail tradeoffs between system capacity (amount of air compressed or expanded per stroke) and system efficiency (partly determined by energy losses due to dead space). The impact of non-ideal valve actuation on CAES system operation is considered for certain embodiments of the invention hereinbelow.

[0029] Every compression or expansion of a quantity of gas, where such a compression or expansion is herein termed "a gas process," is generally one of three types: (1) adiabatic, during which the gas exchanges no heat with its environment and, consequently, rises or falls in temperature, (2) isothermal, during which the gas exchanges heat with its environment in such a way as to remain at constant temperature, and (3) polytropic, during which the gas exchanges heat with its environment but its temperature does not remain constant. Perfectly adiabatic gas processes are not practical because some heat is always exchanged between any body of gas and its environment (ideal insulators and reflectors do not exist); perfectly isothermal gas processes are not practical because for heat to flow between a quantity of gas and a portion of its environment (e.g., a body of liquid), a nonzero temperature difference must exist between the gas and its environment—e.g., the gas must be allowed to heat during compression in order that heat may be conducted to the liquid. Hence real-world gas processes are typically polytropic, though they may approximate adiabatic or isothermal processes.

[0030] The Ideal Gas Law states that for a given quantity of gas having mass m, pressure p, volume V, and temperature T, $pV=mRT$, where R is the gas constant ($R=287$ J/K·kg for air). For an isothermal process, T is a constant throughout the process, so $pV=C$, where C is some constant.

[0031] For a polytropic process, as will be clear to persons familiar with the science of thermodynamics, $pV^n=C$ throughout the process, where n, termed the polytropic index, is some constant generally between 1.0 and 1.6. For $n=1$, $pV^n=pV^1=pV=C$, i.e., the process is isothermal. In general, a process for which n is close to 1 (e.g., 1.05) may be deemed approximately isothermal.

[0032] For an adiabatic process, $pV^\gamma=C$, where γ, termed the adiabatic coefficient, is equal to the ratio of the gas's heat capacity at constant pressure C_P to its heat capacity at constant volume, C_V, i.e., $\gamma=C_P/C_V$. In practice, γ is dependent on pressure. For air, the adiabatic coefficient γ is typically between 1.4 and 1.6.

[0033] Herein, we define a "substantially isothermal" gas process as one having $n \leqq 1.1$. The gas processes conducted within cylinders described herein are preferably substantially isothermal with $n \leqq 1.05$. Herein, wherever a gas process taking place within a cylinder assembly or storage reservoir is described as "isothermal," this word is synonymous with the term "substantially isothermal."

[0034] The amount of work done in compression or expansion of a given quantity of gas varies substantially with polytropic index n. For compressions, the lowest amount of work done is for an isothermal process and the highest for an adiabatic process, and vice versa for expansions. Hence, for gas processes such as typically occur in the compressed-gas energy storage systems described herein, the end temperatures attained by adiabatic, isothermal, and substantially isothermal gas processes are sufficiently different to have practical impact on the operability and efficiency of such systems. Similarly, the thermal efficiencies of adiabatic, isothermal, and substantially isothermal gas processes are sufficiently different to have practical impact on the overall efficiency of such energy storage systems. For example, for compression of a quantity of gas from initial temperature of 20° C. and initial pressure of 0 psig (atmospheric) to a final pressure of 180 psig, the final temperature T of the gas will be exactly 20° C. for an isothermal process, approximately 295° C. for an adiabatic process, approximately 95° C. for a polytropic compression having polytropic index $n=1.1$ (10% increase in n over isothermal case of $n=1$), and approximately 60° C. for a polytropic compression having polytropic index $n=1.05$ (5% increase in n over isothermal case of $n=1$). In another example, for compression of 1.6 kg of air from an initial temperature of 20° C. and initial pressure of 0 psig (atmospheric) to a final pressure of approximately 180 psig, including compressing the gas into a storage reservoir at 180 psig, isothermal compression requires approximately 355 kilojoules of work, adiabatic compression requires approximately 520 kilojoules of work, and a polytropic compression having polytropic index $n=1.045$ requires approximately 375 kilojoules of work; that is, the polytropic compression requires approximately 5% more work than the isothermal process, and the adiabatic process requires approximately 46% more work than the isothermal process.

[0035] It is possible to estimate the polytropic index n of gas processes occurring in cylinder assemblies such as are described herein by empirically fitting n to the equation $pV^n=C$, where pressure p and volume V of gas during a compression or expansion, e.g., within a cylinder, may both be measured as functions of time from piston position, known device dimensions, and pressure-transducer measurements. Moreover, by the Ideal Gas Law, temperature within the cylinder may be estimated from p and V, as an alternative to direct measurement by a transducer (e.g., thermocouple, resistance thermal detector, thermistor) located within the cylinder and in contact with its fluid contents. In many cases, an indirect measurement of temperature via volume and pressure may be more rapid and more representative of the entire volume than a slower point measurement from a temperature transducer. Thus, temperature measurements and monitoring described herein may be performed directly via one or more transducers, or indirectly as described above, and a "temperature sensor" may be one of such one or more transducers and/or one or more sensors for the indirect measurement of temperature, e.g., volume, pressure, and/or piston-position sensors.

[0036] In various embodiments of the present invention, one or more valves (e.g., poppet-type valves) are integrated into the head of the cylinder. By increasing the rapidity and efficiency of valve action, embodiments of the invention increase the overall power density and efficiency of the energy-conversion system. Other advantages accruing therefrom are not described but are contemplated and within the scope of the invention.

[0037] In various embodiments of energy-conversion systems described in the '207 patent, the '155 patent, and U.S. Pat. No. 7,802,426, filed Jun. 9, 2009 (the '426 patent, the entire disclosure of which is incorporated by reference herein), gas is admitted into a chamber of a cylinder at a range of pressures. After being expanded or compressed within the chamber, the gas is exhausted from the chamber. The source of the gas admitted into the chamber and the destination of the gas exhausted from the chamber may be different. For example, gas may be admitted to the chamber from a high-pressure reservoir (or "store") and exhausted from the chamber to a vent or to a chamber within another cylinder. A separate valve is typically required to regulate gas flow from each source or to each destination. It is desirable for valves regulating gas flow to and from a cylinder to operate (i.e., open and close) rapidly and with low expenditure of energy. Rapid valve operation enables the energy-conversion system to perform briefer operational cycles (e.g., (1) admission of fluid to a cylinder chamber, (2) expansion or compression of fluid within the cylinder chamber, and (3) egress of expanded or compressed fluid from the cylinder chamber), which tends to increase the power rating and power density of the energy-conversion system. Low-energy (efficient) valve operation increases the overall efficiency of the energy-conversion system. Moreover, rapid valve operation reduces throttling losses due to restricted flow through the valve opening during intervals when the valve is only partly open. It is also desirable for valves regulating gas flow to and from a cylinder to have high flow coefficient C_v, a dimensionless number used to characterize valve performance (high C_v is achieved when there is low pressure drop through the valve for high flow).

[0038] Embodiments of the present invention advantageously incorporate valve and valve-actuation arrangements that improves the efficiency of the energy storage and recovery when compared to poppet-valve arrangements constructed in accordance with the prior art. Arrangements to more-rapidly achieve sufficiently open and sufficiently closed valve states in accordance with various embodiments of the invention (where the terms "sufficiently open" and "sufficiently closed" shall be defined clearly below), such as are described herein, may allow for more rapid valve opening and closure, thus increasing overall system power rating and density. Arrangements to store energy recovered from valve-disc deceleration (i.e., at end of opening or closing valve strokes) and to use the stored energy in valve-disc acceleration (i.e., at the beginning of opening or closing valve strokes) in accordance with various embodiments of the invention, such as are described herein, may allow for reduction of average energies required to actuate valves, thus increasing overall system efficiency.

[0039] Gas undergoing expansion tends to cool, while gas undergoing compression tends to heat. To maximize efficiency (i.e., the fraction of elastic potential energy in the compressed gas that is converted to work, or vice versa), gas expansion and compression should be as near isothermal (i.e., constant-temperature) as possible. Several techniques of approximating isothermal expansion and compression may be employed in accordance with embodiments of the invention.

[0040] First, as described in the '426 patent, gas undergoing either compression or expansion may be directed, continuously or in installments, through a heat-exchange subsystem external to the cylinder. The heat-exchange subsystem either rejects heat to the environment (to cool gas undergoing compression) or absorbs heat from the environment (to warm gas undergoing expansion). An isothermal process may be approximated via judicious selection of this heat-exchange rate.

[0041] Additionally, as described in the '155 patent, droplets of a liquid (e.g., water) may be sprayed into a chamber of the cylinder in which gas is presently undergoing compression (or expansion) in order to transfer heat to or from the gas. As the liquid droplets exchange heat with the gas around them, the temperature of the gas is raised or lowered; the temperature of the droplets is also raised or lowered. The liquid is evacuated from the cylinder through a suitable mechanism. The heat-exchange spray droplets may be introduced through a spray head (in, e.g., a vertical cylinder), through a spray rod arranged coaxially with the cylinder piston (in, e.g., a horizontal cylinder), or by any other mechanism that permits formation of a liquid spray (or a foam, as described further below) within the cylinder. Droplets (and/or foam) may be used to either warm gas undergoing expansion or to cool gas undergoing compression. Again, an isothermal process may be approximated via judicious selection of this heat-exchange rate. When such liquid heat exchange is utilized, the contents of the chamber may include or consist essentially of a mixture of liquid and gas (e.g., a foam). Any valve used to admit gas to and/or exhaust gas from the chamber preferably accommodates flow of a liquid-gas mixture. Such two-phase flow may exceed a particular quality factor (e.g., >10% volume of liquid compared to the volume of gas, and in some cases >25% volume of liquid).

[0042] Various embodiments of the invention relate to a modified cylinder assembly. The piston within the cylinder divides the interior of the cylinder into two tubular chambers. Each tubular chamber is bounded at one end by the piston and at the other end by an end cap. In various embodiments of the invention, two or more hydraulically, electrically, or mechanically operated two-port poppet valves pass through one of the heads of the cylinder. Each valve comprises a body, actuating mechanism, stem, ring, disc (valve member), two ports, and seat. Each valve contains a chamber, herein termed the "flow chamber," through which fluid may flow.

[0043] In each valve, two ports (openings) allow communication between the interior of the valve chamber and the exterior of the valve. One port is typically open at all times and may be connected to a pipe; this port is herein termed the "outside port." The other port communicates with the interior of the cylinder and is gated by the disc; this port is herein termed the "gated port."

[0044] One end of the stem, herein termed the "distal end," is connected to an actuating mechanism that causes the stem to move along its axis; the other end of the stem, herein termed the "proximal end," is connected to the disc, which is a body of material wider than the stem. The distal end of the stem is farther from the gated port than the proximal end. When the valve is closed, the stem has reached its limit of motion in the proximal direction and the peripheral edge or surface of the disc is in contact with the seat, i.e., a tapered surface or flange surrounding the gated port.

[0045] The two or more valves are typically of at least two types. In one type of valve, the disc is outside the flow chamber. When the valve is open, the stem is at its limit of motion in the proximal direction (i.e., toward the gated port) and the disc is outside the flow chamber and out of contact with the seat, allowing fluid to flow between the flow chamber and the cylinder chamber through the gated port. When the valve is closed, the stem is at its limit of motion in the distal direction (i.e., away from the gated port) and the disc is in contact with the seat. A valve of this type is herein termed a "low-side valve."

[0046] In another type of valve, the disc is inside the flow chamber. When the valve is open, the stem is at its limit of motion in the distal direction (i.e., away from cylinder chamber) and the disc is positioned inside the flow chamber and out of contact with the seat, allowing fluid to flow between the flow chamber and the cylinder chamber through the gated port. When the valve is closed, the stem is at its limit of motion in the proximal direction (i.e., toward the gated port) and the disc is in contact with the seat. A valve of this type is herein termed a "high-side valve."

[0047] Although descriptions herein are typically phrased, for brevity and clarity, to refer to systems having a single intake (high-side) valve and singular outlet (low-side) valve, systems that have multiple intake and outlet valves, whether these multiple valves operate independently or synchronously, are also contemplated and within the scope of the invention.

[0048] When the cylinder is operated as an expander, gas stored high pressure (e.g., approximately 3,000 psi) in a reservoir is admitted to the cylinder assembly through piping and a high-side valve. In an initial state, the fluid gas or gas-liquid mixture within the cylinder chamber is at equal or lower pressure than the gas in the high-pressure reservoir. The high-side valve is open and the low-side valve is closed. High-pressure gas enters the flow chamber of the high-side valve through the high-side valve's outside port. The high-side valve is open, so the disc is not in contact with the seat and both the outside port and the gated port are open. Gas from the high-pressure store flows through the inlet valve into the cylinder.

[0049] In this initial state, the low-side valve is in a closed position. That is, the gated port is occluded by the disc, which is in contact with the seat. Herein, the side of a valve disc connected to the stem is termed the "inner side" of the disc and the opposing side of the disc is termed the "outer side" of the disc. When a high-side or low-side valve is closed, fluid inside the flow chamber of the valve exerts hydraulic force on the inner side of the disc and the fluid contents of the cylinder chamber exert hydraulic force on the outer side of the disc. Force is thus exerted on the disc by fluid on both sides of the disc. Force may also be exerted on the disc by the stem through an actuation mechanism. If the stem and the fluid within the flow chamber of a closed low-side valve exert a greater total force on the disc than the fluid within the cylinder chamber, the disc remains in contact with the seat and the gated port remains closed. If the stem and the fluid within the flow chamber of a low-side valve exert a smaller combined force on the disc than the fluid within the cylinder chamber, the disc moves in the distal direction (i.e., away from the seat) and the gated port opens.

[0050] In the initial state described above, the cylinder chamber fills with high-pressure gas. The outside port of the low-side valve communicates through piping with a body of gas at lower pressure, e.g., the atmosphere or the contents of another cylinder. The force exerted by the fluid within the flow chamber is smaller than the total force on the disc from the fluid within the cylinder chamber and any stem forces exerted by the actuation mechanism. The gated port therefore remains occluded by the disc, i.e., the low-side valve remains closed. No force need be supplied by the activation mechanism of the low-side valve for the valve to remain closed in this state or any other state in which the contents of the cylinder chamber exerts more force on the disc than do the contents of the flow chamber. The low-side valve may thus fail shut.

[0051] In a subsequent operating state, the gaseous component of the fluid within the cylinder chamber has expanded to a pressure (e.g., approximately 300 psi) below that of the high-pressure store. It will be evident to any person reasonably familiar with the art of pneumatic and hydraulic machinery that the high-side valve will fail shut in this operating state, i.e., no force need be supplied by the activation mechanism of the high-side valve in order for the high-side valve to remain closed. In this operating state, sufficient force applied to the stem of the low-side valve by the activation mechanism of the low-side valve will open the low-side valve, allowing fluid within the cylinder chamber to be exhausted through the low-side valve.

[0052] In other modes of operation of the embodiment, not explicitly described, gas may be admitted through the low-side valve, compressed within the cylinder chamber, and forced through the high-side valve to the high-pressure store. In compression mode, the valves may be operated in a check-valve mode, wherein no external actuation force is required.

[0053] Reference is now made to an idealized valve in which, when the valve is closed, the perimeter of a disc (not necessarily circular) makes contact at every point with the perimeter of an opening (i.e., gated port) of similar shape and size. In this case, no flow through the valve is possible. When the valve is open, the area of the opening through which fluid may flow is A_{GP}, the area of the gated port, which is typically slightly smaller but approximately equal to A_D, the area of the disc. The rate of fluid flow that would occur through the open gated port of area A_{GP} for a given pressure difference across the gated port in a case where the flow encounters no obstacle on either side of the gated ported is $F_{max,p}$ (maximum flow for a given differential pressure). The presence of the valve disc in the vicinity of the gated port tends to reduce the effective area available for fluid flow. Assuming disc displacement perpendicular to the plane of the gated port (i.e., typical poppet-style disc motion), the area available for fluid flow through the valve is the area of an imaginary surface connecting the perimeter of the disc to the perimeter of the gated port. Herein, this available area is termed the "curtain area," $A_{curtain}$, after the analogical resemblance of the imaginary surface to a curtain dropped from the perimeter of the disc to the perimeter of the gated port. For example, for a circular disc of radius R displaced perpendicularly by distance h from a gated port, $A_{GP}=\pi R^2$ and $A_{curtain}=2\pi Rh$ by elementary geometry. For $h<R/2$, we have $A_{curtain}<A_{GP}$. That is, when the disc is closer to the gated port than half its own radius, the area $A_{curtain}$ available for flow through the valve is less than the area of the gated port, A_{GP}. At $h=R/2$, we have $A_{curtain}=A_{GP}$, and for $h>R/2$, we have $A_{curtain}>A_{GP}$.

[0054] It will be apparent to persons reasonably familiar with the science of fluid dynamics that the rate of flow through the valve will tend to be less for a given pressure drop through the valve, given the presence of the disc at any finite distance, than it would be in the absence of the disc. Equivalently, a given rate of flow through the valve, given the presence of the disc at some finite distance, will tend to entail a higher pressure drop (and thus energy loss) than the same rate of flow would entail in the absence of the disc. In short, the disc tends to impede flow through the gated port to some extent no matter how far it is from the seat.

[0055] For a given pressure drop across the valve, flow through the valve is a function F(h) of disc displacement h. In particular, $F(h)=0$ when $h=0$ (i.e., valve is closed) and $F(h)\rightarrow F_{max,p}$ as $h\rightarrow\infty$. That is, F(h) approaches its maximum possible value $F_{max,p}$ asymptotically as the disc is moved to an ever-greater distance from the seat. Thus, even where $h\geqq R/2$, and the available flow area $A_{curtain}$ is equal to or greater than the gated port area A_{GP}, flow F(h) for a given pressure drop is less than the theoretical maximum flow $F_{max,p}$.

[0056] However, most (e.g., approximately 90%) of the potential gain in flow with increasing h is realized by displacing the disc to $h=R/2$, that is, making the curtain area $A_{curtain}$ equal to the gated port area A_{GP}. Poppet-type valves are therefore typically deemed fully open when they are "sufficiently open," i.e., when an opening valve stroke has moved the disc to a distance h=R/2 away from the gated port, or to a distance of similar magnitude. Poppet valves are typically designed to move the disc to approximately h=R/2, or to a displacement of similar magnitude, as a final, fully open position. Herein, we term the flow through the valve for a given differential pressure when h equals R/2, or a value of similar magnitude, the sufficiently open flow, $F_{SO,p}$.

[0057] A valve is closed when the disc is in contact with the seat and the valve opening is thus 100% occluded. Sufficient closure may be defined as a state wherein flow for a given differential pressure is, e.g., less than 1% of the sufficiently open flow, $F_{SO,p}$.

[0058] Values of sufficiently open displacement h_{SO} approximately equal to R/2 are discussed herein as typical, but other values of h_{SO} are also envisaged. Realistic values of h_{SO} will typically be of similar magnitude to R/2.

[0059] When a disc and stem are moved so as to open or close a valve, the stem and disc are first accelerated from rest, and then, at the end of the stroke, decelerated until they are again at rest. The deceleration may occur suddenly: for example, during valve closure, sudden deceleration occurs if the disc is allowed to impact the seat at its maximum closure velocity, without prior deceleration. However, undecelerated impact entails the action of short-lived, high-magnitude forces on the seat, disc, stem, and possibly other valve and system components. These sudden, strong forces may cause component wear, noise, disc bounce, and other undesirable effects. Sudden deceleration of the disc during valve opening does not entail collision of the disc with the seat (because the disc is moving away from the seat during opening), but typically does entail similar impact forces elsewhere in the valve mechanism.

[0060] Therefore, it is typical to provide arrangements for decelerating the stem and disc before end of valve stroke, during both opening and closing. Such pre-deceleration, entails the action of longer-lived, lower-magnitude forces on valve components than does the acceptance of deceleration by impact. However, such pre-deceleration, by reducing the average velocity of the valve during the opening or closing stroke, slows valve actuation and increases throttling losses during prolonged, partially-opened valve states.

[0061] In embodiments of the invention, provision is made for rapid opening and closing of both low-side and high-side valves in a manner that preserves the primary advantage of pre-deceleration (i.e., avoidance or mitigation of impact forces) while shortening valve actuation time in comparison to a valve arranged in accordance with the prior art. Embodiments of the invention also include provisions for storing a portion of the energy transferred from the disc and stem, and possibly other components, during deceleration, whether during opening or closing, and restoring a portion of the stored energy to the disc and other moving components during acceleration.

[0062] Embodiments of the present invention are typically utilized in energy storage and generation systems utilizing compressed gas. In a compressed-gas energy storage system, gas is stored at high pressure (e.g., approximately 3,000 psi). This gas may be expanded into a cylinder having a first compartment (or "chamber") and a second compartment separated by a piston slidably disposed within the cylinder (or by another boundary mechanism). A shaft may be coupled to the piston and extend through the first compartment and/or the second compartment of the cylinder and beyond an end cap of the cylinder, and a transmission mechanism may be coupled to the shaft for converting a reciprocal motion of the shaft into a rotary motion, as described in the '678 and '842 patents. Moreover, a motor/generator may be coupled to the transmission mechanism. Alternatively or additionally, the shaft of the cylinders may be coupled to one or more linear generators, as described in the '842 patent.

[0063] As also described in the '842 patent, the range of forces produced by expanding a given quantity of gas in a given time may be reduced through the addition of multiple, series-connected cylinder stages. That is, as gas from a high-pressure reservoir is expanded in one chamber of a first, high-pressure cylinder, gas from the other chamber of the first cylinder is directed to the expansion chamber of a second, lower-pressure cylinder. Gas from the lower-pressure chamber of this second cylinder may either be vented to the environment or directed to the expansion chamber of a third cylinder operating at still lower pressure; the third cylinder may be similarly connected to a fourth cylinder; and so on.

[0064] The principle may be extended to more than two cylinders to suit particular applications. For example, a narrower output force range for a given range of reservoir pressures is achieved by having a first, high-pressure cylinder operating between, for example, approximately 3,000 psig and approximately 300 psig and a second, larger-volume, lower-pressure cylinder operating between, for example, approximately 300 psig and approximately 30 psig. When two expansion cylinders are used, the range of pressure within either cylinder (and thus the range of force produced by either cylinder) is reduced as the square root relative to the range of pressure (or force) experienced with a single expansion cylinder, e.g., from approximately 100:1 to approximately 10:1 (as set forth in the '853 application). Furthermore, as set forth in the '678 patent, N appropriately sized cylinders can reduce an original operating pressure range R to $R^{1/N}$. Any group of N cylinders staged in this manner, where $N \geqq 2$, is herein termed a cylinder group.

[0065] In one aspect, embodiments of the invention feature an energy storage and recovery system that includes a cylinder assembly for compression of gas to store energy and/or expansion of gas to recover energy therewithin, the cylinder assembly having an interior compartment and an end cap disposed at one end. Integrated within the end cap are (i) a first valve for admitting fluid into the interior compartment of the cylinder assembly prior to expansion and exhausting fluid from the interior compartment of the cylinder assembly after compression and (ii) a second valve for exhausting fluid from the interior compartment of the cylinder assembly after expansion and admitting fluid into the interior compartment of the cylinder assembly prior to compression. Each of the first and second valves controls fluid communication with the interior compartment via a separate fluid path, and each comprises a gated port and an outside port. The system also includes a first actuation mechanism for actuating the first valve and a second actuation mechanism for actuating the second valve, as well as a control system for controlling the first and second actuation mechanisms based at least in part on the pressure inside the interior compartment of the cylinder assembly, the position of the gated port of the first valve, and/or the position of the gated port of the second valve.

[0066] Various embodiments of the invention incorporate one or more of the following in any of a variety of combinations:

[0067] (1) The actuation mechanisms of the first valve and/or second valve include arrangements for moving the valve disc to a full-open distance h_{FO} from the valve seat. (Herein, disc displacement is given as the distance from the proximal surface of the disc to the inner perimeter of the seat.) The full-open distance h_{FO} is substantially greater than the sufficiently-open distance h_{SO}. Herein, h_{FO} is "substantially greater" than h_{SO} if the difference of the two distances, $h_{FO}-h_{SO}$, suffices, during an opening valve stroke, for the actuation mechanism to decelerate the stem and disc from their maximum opening-stroke velocity V_{MO} to an acceptably low final opening velocity V_{OV} (e.g., zero).

[0068] During an opening stroke, the disc and stem are accelerated from rest by the actuation mechanism (and/or by pressurized fluids exerting hydraulic force on the disc) to the maximum opening velocity V_{MO}. The disc and stem may attain V_{MO} at or before reaching the sufficiently-open distance h_{SO}. When or after the disc reaches h_{SO}, the actuation mechanism begins to decelerate the disc and stem. By the time the disc reaches the final-open distance h_{FO}, the actuation mechanism has decelerated the disc and stem to the acceptable final-opening velocity V_{OV} (e.g., zero).

[0069] (2) The valve seat includes a contact ring of a suitable material (e.g., polyether ether ketone [PEEK]) connected to a shock-absorbing mechanism. The contact ring is the portion of the valve seat that touches the disc when the valve is closed. The shock-absorbing mechanism may comprise, for example, an annular wave spring mounted beneath the contact ring. Other types of shock-absorbing mechanism, such as air springs and polymer elastic materials, are contemplated and within the scope of the invention. The shock-absorbing mechanism allows motion of the ring from an initial position at h=0 to a substantially depressed position at $h=-h_{SD}$. Herein, negative distances denote proximal displacement from h=0. The distance $-h_{SD}$ is "substantial" if it suffices, during a closing valve stroke, for the shock-absorbing mechanism to decelerate the stem and disc from their maximum closing-stroke velocity V_{MC} to an acceptably low final closing velocity V_{CV} (e.g., zero).

[0070] During a closing stroke of the valve, the seat and stem are accelerated from rest by the actuation mechanism to the maximum closing velocity V_{MC}. The disc and stem may attain V_{MC} at or before the disc reaches the contact ring (h=0); when the disc reaches the contact ring, it is moving at V_{MC}. From the moment of first disc-and-ring contact forward, a deceleration mechanism (e.g., wave spring) connected to the ring presents resistance to the motion of the disc, decelerating the disc. By the time the disc reaches its substantially-depressed displacement h_{SD}, the deceleration mechanism has decelerated the disc to the acceptable final-closing velocity V_{CV}. Thereafter, in some embodiments, the deceleration mechanism restores the disc and contact ring to the neutral position h=0.

[0071] (3) Provision is made for a portion of the kinetic energy removed from the disc and stem during deceleration, whether during opening or closing, to be stored, and for a portion of this stored energy to be imparted, in the form of kinetic energy, to the disc and stem during acceleration. Herein, such an arrangement is termed "regenerative valving." For example, the energy may be stored as pressure potential energy of a fluid.

[0072] Various embodiments of the invention employ one or more valves (e.g., poppet-type valves) that may be integrated into the head of the cylinder. These valves provide quick valve action, high flow coefficient (i.e., low pressure drop through the valve for high flow), and other advantages, some of which are described below. By increasing the efficiency of valve action, embodiments of the invention increase the overall efficiency of the energy-conversion system.

[0073] In various embodiments that incorporate hydraulically activated two-port poppet valves that pass through one of the heads of the cylinder, the hydraulic activation mechanism of each poppet valve includes a hydraulic cylinder. The piston within the activation cylinder divides the interior of the activation cylinder into two tubular, fluid-filled chambers. Each tubular chamber is bounded at one end by the proximal surface of the piston and at the other end by an end cap. A stem is attached to the piston and passes through the proximal end-cap of the activation cylinder. The stem is aligned with and attached to a second stem passing through the distal end-cap of a poppet valve that gates fluid flow into and/or out of the modified cylinder assembly. The second stem is attached to the disc of the poppet valve. The piston and stem of the activation cylinder move in unison with the disc and stem of the poppet valve. The two chambers of the hydraulic activation cylinder are herein designated the proximal chamber (the chamber closer to the modified cylinder assembly) and the distal chamber (the chamber farther from the modified cylinder assembly).

[0074] When fluid pressure in the distal chamber of the activation cylinder exceeds the fluid pressure in the proximal chamber of the activation cylinder, the piston and stem of the activation cylinder, hence also the piston and stem of the poppet valve, which move in unison therewith, tend to accelerate toward the cylinder assembly. When the fluid pressure in the proximal chamber of the activation cylinder exceeds the fluid pressure in the distal chamber of the activation cylinder, the piston and stem of the activation cylinder, hence also the piston and stem of the poppet valve, tend to accelerate away from the cylinder assembly. Herein, the terms "accelerate" and "decelerate" are employed interchangeably: typically, "deceleration" denotes acceleration such that an object's velocity decreases in magnitude.

[0075] When the piston of the activation cylinder is at or near the proximal limit of its range of motion, the poppet disc is seated (i.e., the poppet valve is closed and no fluid enters or leaves the modified cylinder assembly through the poppet valve). When the piston of the activation cylinder is at or near the distal limit of its range of motion, the poppet disc is at a distance from the seat such that the poppet valve is fully open and fluid enters or leaves the modified cylinder assembly while undergoing a minimal pressure drop. When the piston of the activation cylinder is at any intermediate position, at (or near) neither its proximal nor distal limit of motion, the activation cylinder is generally in a transient opening or closing state. In particular, when the piston of the activation cylinder is near (but not at) its proximal limit of motion, the poppet valve is only partially open, i.e., the disc of the poppet valve is relatively near to the seat and, as regards this poppet valve, fluid enters or leaves the modified cylinder assembly only through the relatively constricted opening between the disc and the seat. Fluid flow through such an opening entails turbulence and throttling losses (i.e., losses of useful energy). Losses thus entailed are herein termed "disc-proximity losses." The period during valve opening or closure during which disc proximity losses are non-negligible is herein termed the "disc-proximity period," and the range of proximal-distal disc positions in which disc proximity losses are non-negligible is herein termed the "disc-proximity zone." The length of the disc-proximity zone is herein termed D_{prox}. In general, in order to maximize overall energy-storage system efficiency, the disc-proximity period is minimized during poppet valve opening and closure.

[0076] To minimize disc-proximity losses, the poppet valve disc is moved as quickly as possible from its fully open position into contact with the seat (when the poppet valve is being closed) and as quickly as possible away from the seat to its fully open position (when the poppet valve is being opened). One approach to minimizing disc-proximity losses during valve closure is to accelerate the piston of the activation cylinder, and thus the disc of the poppet valve, to some relatively high velocity in the proximal direction. The acceleration occurs mostly or entirely before the disc of the poppet reaches the disc-proximity zone (i.e., gets within D_{prox} of the seat). The disc of the poppet valve thus moves at high velocity through the disc-proximity zone until contact is made with the seat. Alternatively, the piston of the activation cylinder, and thus the disc of the poppet valve, may be accelerated in the proximal direction right up until the disc strikes the seat. In general, however, high-velocity impact between the disc and seat, whether the disc is still accelerating or not at the moment of impact, are disadvantageous, since they create shock waves and component wear and mandate heavier, more robust components (e.g., stems), which are subject to more force to undergo any given acceleration than do lighter, less robust components. In general, low disc-to-seat impact velocities are desirable.

[0077] In light of the foregoing considerations it is advantageous, during poppet-valve closure, to first (a) produce high-velocity motion of the piston of the activation cylinder, and thus of the disc of the poppet valve, and then (b) to rapidly decelerate these components as the disc approaches the seat, so that disc-to-seat impact velocity is acceptably low. In general, the shortest disc-proximity period, with minimal disc-proximity losses, will be achieved during poppet-valve closure if (a) the highest possible velocity V_{max} is imparted to the activation cylinder piston and poppet disc before the disc enters the disc-proximity zone, and (b) the activation cylinder piston and poppet disc are decelerated with the highest possible acceleration just prior to disc-to-seat impact, starting at V_{max} and ending with some acceptably low disc-to-seat impact velocity V_{end}. Herein, vectors of velocity and acceleration are referred to by their scalar magnitudes, with direction of action made clear by context or explicit statement.

[0078] As noted above, the activation cylinder piston will tend to accelerate in the distal direction—e.g., during poppet-valve closure, will slow down (decelerate)—when the hydraulic force acting on the proximal face of the piston is greater than the hydraulic force acting on the distal face of the piston. (Any other distal-directed force acting on the activation cylinder piston—e.g., an electromagnetic force, or a mechanical force applied to the stem of the activation cylinder piston by some mechanism—will also tend to accelerate the activation cylinder piston in the distal direction.) During deceleration of the activation cylinder piston, one therefore wishes in general to produce the maximum possible pressure within the proximal chamber of the activation cylinder. The maximum possible pressure in the proximal chamber of the activation cylinder during piston deceleration is in general determined by the hydraulic pressure rating P_{max} of the activation cylinder.

[0079] In various embodiments where no electromagnetic or other forces act on the piston in addition to the hydraulic forces within the activation cylinder, shortest-duration deceleration of the activation cylinder piston during poppet-valve closure occurs ideally as follows. (1) The activation cylinder piston and all system components moving in unison therewith, including the poppet disc, are accelerated to some proximally-directed maximum velocity V_{max}. V_{max} is achieved by the time the disc of the poppet valve reaches the beginning of the disc proximity zone: i.e., the disc is moving at V_{max} by the time the disc is a distance D_{prox} from the seat. (2) When the disc and other components reach distance D_{prox} from the seat, moving at V_{max}, the fluid pressure in the distal chamber of the activation cylinder drops suddenly to zero or some negative value and pressure in the proximal chamber jumps from zero or some negative value to P_{max}. Pressure in the proximal chamber remains constant at P_{max} during deceleration as fluid is expelled from the proximal chamber. A distally-directed net hydraulic force F_{decel} thus acts upon the activation cylinder piston and on the components moving in unison therewith. As the activation cylinder piston moves through distance D_{prox}, the constant force F_{decel} decelerates the activation cylinder piston to an acceptable disc-to-seat impact velocity, V_{end}. This deceleration takes place at constant acceleration A. By Newton's Second Law, $F_{decel}=M_T A$, where M_T is the total mass of the activation cylinder piston and all components moving in unison therewith. (3) During disc-to-seat impact, all hydraulic forces acting on the activation cylinder piston are zero or negligible, and impact forces dominate the deceleration to rest of the disc and components moving in unison therewith, including the activation cylinder piston. Some rebound motion of the disc and other components may occur subsequent to impact; such motions may be damped by friction and by hydraulic and/or other contrivances.

[0080] Maximum deceleration for a minimum time interval will, in general, minimize disc-proximity losses during closing of the poppet valve because the time interval during which such losses occur will be minimized. Maximum deceleration, in embodiments where deceleration of the poppet valve is primarily due to force exerted on the activation piston by fluid within the proximal chamber of an activation cylinder, will occur if and only if maximum-allowable pressure P_{max} is maintained in the proximal chamber of the activation cylinder throughout deceleration. In general, however, allowing the decelerating piston (whose effective mass will be equal to that of all components moving in unison therewith) to expel fluid from the proximal chamber via a path having fixed flow resistance (e.g., one or more fixed orifices connected to fixed exterior piping and other components) will not produce a constant pressure P_{max} in the proximal chamber of the activation cylinder. Rather, the pressure P(t) in the proximal chamber will decrease as the piston decelerates. Here the notation P(t) signifies that the pressure P is a function of time.

[0081] To produce a constant or approximately constant pressure P_{max} in the proximal chamber of the activation cylinder throughout deceleration of the piston from initial velocity V_{max} to final velocity V_{end}, fluid being expelled from the chamber by the decelerating piston is directed through a channel having a time-variable (modulated) flow resistance $R_{flow}(t)$. Such modulation is specifically timed and controlled in order to produce a constant or approximately constant pressure P_{max} (or any other specific time-dependent pressure profile) in the proximal chamber of the activation cylinder.

Variability in the flow resistance $R_{flow}(t)$ may be created by several means: for example, pressure drop through a valve in the flow path external to the activation cylinder may be actively modulated during deceleration by narrowing or widening the interior diameter of the valve.

[0082] In some embodiments of the present invention, the orifice through which fluid exits the proximal chamber is modulated during deceleration of the piston in order to produce an approximately constant pressure P_{max} (or some desired time-dependent pressure profile) in the proximal chamber. Orifice modulation during deceleration may be achieved by means of a shutter, dilating iris, or other contrivance within the valve assembly. Orifice modulation during valve closure may be achieved most simply by shaping and placing an orifice or orifices in the side-wall of the proximal chamber in such a manner that the activation cylinder piston itself progressively curtains off or occludes the orifice or orifices as it decelerates. Such progressive occlusion will modulate the flow resistance encountered by the fluid exiting the proximal chamber during deceleration. Herein, orifices shaped and placed to be progressively occluded by a moving piston are termed "occludable orifices," as distinct from "fixed orifices" (e.g., in the end-cap of the activation cylinder), whose opening area does not change during piston motion. Occludable orifices may be combined with fixed orifices in a given cylinder assembly. Appropriate sizing, shaping, and placing of occludable and fixed orifices may be used to tune the flow resistance encountered by fluid being forced from the proximal chamber throughout piston deceleration, and thus to tune the pressure P(t) within the proximal chamber throughout piston deceleration. Within the material limits of a given assembly, tuning of pressure P(t) within the proximal chamber is arbitrary (i.e., may take on any desired functional shape). One possible tuning, already described, is to arrange for a constant pressure $P(t)=P_{max}$ within the proximal chamber throughout deceleration.

[0083] The precise shaping and arrangement of multiple occludable orifices in an activation cylinder, and their possible combination with one or more fixed orifices, in order to produce any P(t) within the proximal chamber of the activation cylinder throughout deceleration, thus minimizing deceleration time and disc proximity losses, is in general non-unique (i.e., more than one arrangement of orifices may produce any specific P(t)). The relationship between orifice number, size, placement, and shape and P(t) is in practice dependent in a complex and non-obvious manner on cylinder geometry, properties of the hydraulic fluid, and other factors. All such shapings and arrangements of orifices, and all combinations of such orifice shapings and arrangements with other methods of modulating flow resistance during valve closure (e.g., modulation of external flow resistances) in order to tune or tailor the pressure within the proximal chamber, the distal chamber, or both, are contemplated and within the scope of the invention.

[0084] As will be clear to persons familiar with hydraulic pistons and valves, the methods and techniques described above for producing tuned stroke deceleration of the activation cylinder piston during closure of a poppet valve may also be applied to tuning deceleration during the opening stroke of an activation cylinder, or of any other hydraulic cylinder in which a piston is decelerated by end of stroke. The application to opening strokes of the methods and techniques described above for tuning stroke deceleration is not discussed further herein.

[0085] In various embodiments, it may be neither practical nor necessary to produce constant P(t) in the proximal chamber of the activation cylinder during deceleration of the piston. Rather, it may suffice to limit the peak value of P(t) during deceleration to some highest acceptable value P_{max} while decelerating the piston more rapidly than would be possible (without exceeding P_{max}) in the absence of an appropriately shaped occludable orifice or other contrivances for adjusting the relationship between fluid outflow from the proximal chamber and the velocity of the decelerating piston. That is, a flattening or smoothing of the P(t) curve by the methods and techniques described will in general be advantageous even if ideal deceleration of the piston (constant P_{max} during deceleration) is not achieved.

[0086] Various embodiments of the invention, as already noted hereinabove, employ one or more valves that may be integrated into the head of a cylinder and that typically (i) use differential pressure to open, (ii) use electromagnetic force to hold open, and (iii) check closed. Each valve may exert an electromagnetic force upon its sealing member (also herein termed its "valve member") at any time during opening, holding open, closing, or holding closed; the force thus exerted may act in either a closing direction or opening direction, and its magnitude may be controlled dynamically by a time-varying current or currents activating one or more electromagnets. Such valves, herein termed "electromagnetic valves" or simply "valves" (where context makes such usage unambiguous), provide quick valve action, low valve-actuation energy, high flow coefficient (i.e., low pressure drop through the valve for high flow), protection against loss of pressurized fluid with failure of valve-actuation power, fine control over valve actuation force as a function of time, and other advantages, some of which are described below. By increasing the efficiency of valve action, embodiments of the invention increase the overall efficiency of the energy-conversion system. Other advantages accruing from embodiments of the invention are not described but are contemplated and within the scope of the invention.

[0087] Electromagnetic valves having two ports, a single disc-shaped valve member, and a single seat are described herein, but electromagnetic valves that have multiple ports and multiple valve members and seats, and that also or alternatively have valve members and seats in the general form of rings, plates, and other geometric shapes, are all contemplated and within the scope of the invention. The number of ports and the shape and number of the valve members and rings in a given electromagnetic valve, as well as the number of valves employed in a given cylinder assembly, may all be varied without departing from the scope of the invention.

[0088] In some embodiments of the invention, in each electromagnetic valve, two ports (openings) allow communication between the interior of the flow chamber and the exterior of the valve. One port is typically open at all times and may be connected to a pipe; this port is herein termed the "outside port." The other port communicates with the interior of the cylinder and is gated by the valve member; this port is herein termed the "gated port."

[0089] In some embodiments of the invention, an electromagnetic actuation mechanism including an electromagnet may (for example, depending on the direct of the current in its windings), exert either an attractive or repulsive force on a permanent magnet in the valve member. The motion of the valve member may be constrained by mechanical guides (e.g., the walls of the flow chamber) to be either directly toward or away from the actuation mechanism. When the valve is closed, the peripheral edge or surface of the valve member is generally in contact with the seat, i.e., a tapered surface or flange surrounding the gated port. The valve member and seat are preferably shaped in a complementary manner such that when the valve member is in contact with the seat, the gated ported is entirely blocked. When the valve is open, the valve member is not in contact with the seat, and fluid may flow through the gap between the valve member and the seat and through the gated port.

[0090] In various embodiments of the invention, the valves are of at least two types. In one type of valve, the valve member is inside the flow chamber. A permanent magnet may be attached to, be within, or be a portion of the valve member. A magnetic field produced by the actuation mechanism exerts an attractive or repulsive force on the permanent magnet, tending to open or close the valve. When the valve is open, the valve member is out of contact with the seat, allowing fluid to flow between the flow chamber and the cylinder chamber through the gated port. When the valve is open, the distance between the facing poles of the magnets of the actuation mechanism and valve member is reduced or minimized. When the valve is closed, the flow chamber is in contact with the seat. The valve member and seat are preferably shaped in a complementary manner such that when the valve member is in contact with the seat, the gated port is entirely blocked. A valve of this type is herein termed a “high-side valve.”

[0091] In another type of valve according to various embodiments of the invention, the valve member is outside the flow chamber, and a stem may extend from the valve member through the flow chamber to the actuation mechanism. A permanent magnet may be attached to, be within, or be a portion of the stem. A magnetic field produced by the actuation mechanism exerts an attractive or repulsive force on the permanent magnet, tending to open or close the valve. The valve member and seat are preferably shaped in a complementary manner such that when the valve member is in contact with the seat, the gated port is entirely blocked. When the valve is open, the valve member is out of contact with the seat, allowing fluid to flow between the flow chamber and the cylinder chamber through the gated port. When the valve is open, the valve member is close to or in contact with the seat, and the distance between the facing poles of the magnets of the actuation mechanism and valve member is reduced or minimized. A valve of this type is herein termed a “low-side valve.”

[0092] In various embodiments of the invention, components of the actuation mechanism (e.g., conducting wires that may create a magnetic field) may be embedded within the body of the valve, surrounding the seat; springs may be attached to the valve member or positioned so that in some states of operation the valve member comes in contact with the springs; hydraulic or other types of mechanisms may exert force upon the valve member in various states of operation, additional to any electromagnetic forces that may be exerted on the valve member; the valve member may include an electromagnet rather than a permanent magnet; the actuation mechanism may include a permanent magnetic rather than an electromagnet; two or more electromagnets may be employed, rather than one electromagnet and one permanent magnet; the valve member of a low-side valve may not be connected to a stem; the valve member of a high-side valve may be connected to a stem; and other elements of valve structure may differ from those explicitly described herein, without departing from the scope of the invention.

[0093] When the cylinder is operated as an expander, gas stored high pressure (e.g., approximately 3,000 psi) in a reservoir is admitted to the cylinder assembly through piping and a high-side valve. In an initial state, the fluid gas or gas-liquid mixture within the cylinder chamber is at equal or lower pressure than the gas in the high-pressure reservoir. The high-side valve is open and the low-side valve is closed. High-pressure gas enters the flow chamber of the high-side valve through the high-side valve’s outside port. The high-side valve is open, so the valve member is not in contact with the seat and both the outside port and the gated port are open. Gas from the high-pressure store flows through the inlet valve into the cylinder.

[0094] In this initial state, the low-side valve is in a closed position. That is, the gated port is occluded by the valve member, which is in contact with the seat. Herein, the side of a valve member connected to the stem is termed the “inner side” of the valve member and the opposing side of the valve member is termed the “outer side” of the valve member. When a high-side or low-side valve is closed, fluid inside the flow chamber of the valve exerts hydraulic force on the inner side of the valve member and the fluid contents of the cylinder chamber exert hydraulic force on the outer side of the valve member. Force is thus exerted on the valve member by fluid on both sides of the valve member. Force may also be exerted on the valve member by the stem through the actuation mechanism. If the stem and the fluid within the flow chamber of a closed low-side valve exert a greater total force on the valve member than the fluid within the cylinder chamber, the valve member remains in contact with the seat and the gated port remains closed. If the stem and the fluid within the flow chamber of a low-side valve exert a smaller combined force on the valve member than the fluid within the cylinder chamber, the valve member moves in the distal direction (i.e., away from the seat) and the gated port opens.

[0095] In the initial state described above, the cylinder chamber fills with high-pressure (e.g., 3,000 psi) gas. The outside port of the low-side valve communicates through piping with a body of gas at lower pressure, e.g., the atmosphere or the contents of another cylinder. The force exerted by the fluid within the flow chamber is smaller than the total force on the valve member from the fluid within the cylinder chamber and any forces exerted on the valve member by the actuation mechanism. The gated port therefore remains occluded by the valve member, i.e., the low-side valve remains closed. No closing force need be supplied by the activation mechanism of the low-side valve for the valve to remain closed in this state or any other state in which the contents of the cylinder chamber exert more force on the valve member than do the contents of the flow chamber. The low-side valve may thus fail shut or “check closed.” The low-side valve may also be actuated closed or, when closed, provided with a closing bias (substantially constant closing force). Similarly, no closing force need be supplied by the activation mechanism of the high-side valve for the valve to remain closed in any state in which the contents of the flow chamber exert more force on the valve member than do the contents of the cylinder chamber. The high-side valve, like the low-side valve, may thus fail shut or “check closed.” The high-side valve may also be actuated closed or, when closed, provided with a closing bias.

[0096] In a subsequent operating state, the gaseous component of the fluid within the cylinder chamber has expanded to a pressure (e.g., 300 psi) below that of the high-pressure store. It will be evident to any person reasonably familiar with the art of pneumatic and hydraulic machinery that the high-side valve will fail shut in this operating state, i.e., no force need be supplied by the activation mechanism of the high-side valve in order for the high-side valve to remain closed. In this operating state, sufficient force applied to the valve member of the low-side valve by the activation mechanism of the low-side valve will open the low-side valve, allowing fluid within the cylinder chamber to be exhausted through the low-side valve.

[0097] In other modes of operation of the embodiment, not explicitly described, gas may be admitted through the low-side valve, compressed within the cylinder chamber, and forced through the high-side valve to the high-pressure store. In compression mode, the valves may be operated in a check-valve mode in which no external actuation force is required.

[0098] Methods of reducing coupling losses and improving system performance during an expansion stage of a compressed-gas energy storage system by pre-compressing any fluid remaining within the cylinder chamber before admitting additional high-pressure gas from a source outside the cylinder are disclosed in U.S. patent application Ser. No. 13/650,999, filed Oct. 12, 2012 (the '999 application), the disclosure of which is hereby incorporated by reference in its entirely. Embodiments of the present invention may be combined with the methods disclosed in the '999 application, realizing advantages of both the present invention and the methods disclosed in the '999 application.

[0099] Use of differential pressure (i.e., differences in pressure, in various states of operation, between fluids in the cylinder chamber and in inlet and outlet pipes communicating with the cylinder chamber through high-side and low-side valves) to open the high-side and low-side valves, with use of electromagnetic force for holding open, accelerating, or otherwise influencing the opening or closing of such valves, is generally advantageous compared to conventional systems. The electromagnetic force between two magnetic poles, whether attractive or repulsive, is proportional to $1/x^2$, where x is the distance (gap) between the poles. Therefore, in embodiments of the present invention where the distance between the magnets (e.g., the electromagnet of the actuation mechanism and the permanent magnet of the valve member) decreases during opening of either a high-side or low-side electromagnetic valve, an increasing attractive force between the magnets (i.e., increasing electromagnetic opening force on the valve member) may be obtained during opening by maintaining a constant or increasing current in the windings of the electromagnet; or, a constant opening force may be obtained by appropriately decreasing the current in the windings; or, a decreasing opening force may be obtained by more rapidly decreasing the current in the windings; or, zero or minimal opening force may be obtained by setting the current in the windings to zero (or to a constant DC value). The force between the magnets may be varied arbitrarily, or even reversed in direction, during closing by appropriate variation of the current in the windings.

[0100] In a state of operation where the opening motion of the valve member is favored by forces arising from differential pressure in the flow chamber and cylinder chamber, the energy that must be expended by the actuation mechanism to achieve a given opening speed is minimal, and the valve will tend to open even if the current in the windings is zero (i.e., if zero energy is expended by the actuation mechanism). However, non-zero opening force may accelerate the opening of the valve, advantageously speeding the commencement of flow between the flow chamber and the cylinder chamber. Moreover, by appropriate time-variation of the current in the windings of the actuation mechanism, the velocity of the valve member may be controlled precisely throughout opening of the valve, potentially realizing further advantages. For example, reversal of the current in the windings of the electromagnet at the end of the valve-opening process (i.e., as the gap between the two poles becomes relatively small) may decelerate the valve member as it approaches the actuation mechanism, reducing collision forces and so extending the life of the electromagnetic valve.

[0101] It is typically desirable to exert a holding force on the valve member of an open valve in order to keep the valve member in full-open position and thus assure that the flow chamber remains minimally obstructed while fluid is flowing through the valve. Because the force between the magnetic poles for a given winding current is proportional to $1/x^2$, and effective x is at a minimum when the magnets are at a minimal distance from each other (e.g., in contact), a holding force may be exerted on the valve member of an open valve using a minimal winding current and thus minimal energy. In some embodiments, the magnetic attraction between the ferromagnetic core of the actuation mechanism and the permanent magnet of the valve member may provide sufficient holding-open force even if zero current flows through the windings of the actuation mechanism. A valve may be opened entirely by differential pressure, then held open by electromagnetic force. A valve may also be opened by a combination of differential pressure and electromagnetic force acting on the valve member, or solely by electromagnetic force, and once open may be held open by electromagnetic force.

[0102] Advantages may be realized by embodiments of the invention not only during opening and holding-open of the electromagnetic valve, as described above, but during closure. For example, at the initiation of closure of a fully open valve, the distance x between the poles of the electromagnet and the permanent magnet is minimal, so a relatively large repulsive force between the two magnets may be produced by a relatively small current (small actuation energy). In this manner, the movement of the valve member to a closed position may be initially accelerated with relatively small expenditure of energy. The directional sense of the current flowing in the windings of the actuation mechanism during valve closure will generally be the reverse of the current flowing in the windings during valve opening. By appropriate time-variation of the current in the windings of the actuation mechanism, the velocity of the valve member may be controlled precisely throughout closing of the valve, potentially realizing further advantages. For example, reversal of the current in the windings of the electromagnet at the end of the valve-closing process (i.e., as the gap between the two poles becomes relatively large) may decelerate the valve member as it approaches the seat, reducing collision forces and so extending the life of the electromagnetic valve. A holding force may be exerted by the actuation mechanism on a closed valve member. However, in typical operation, forces placed on the valve member by differential pressure will render the generation of an electromagnetic closure-holding force unnecessary.

[0103] All of the approaches described above for converting potential energy in a compressed gas into mechanical and electrical energy may, if appropriately designed, be operated in reverse to store electrical energy as potential energy in a compressed gas. Since the accuracy of this statement will be apparent to any person reasonably familiar with the principles of electrical machines, power electronics, pneumatics, and the principles of thermodynamics, the operation of these mechanisms to both store energy and recover it from storage are not necessarily described for each embodiment. Such operation is, however, contemplated and within the scope of the invention and may be straightforwardly realized without undue experimentation.

[0104] The systems described herein, and/or other embodiments employing foam-based heat exchange, liquid-spray heat exchange, and/or external gas heat exchange, may draw or deliver thermal energy via their heat-exchange mechanisms to external systems (not shown) for purposes of cogeneration, as described in U.S. Pat. No. 7,958,731, filed Jan. 20, 2010 (the '731 patent), the entire disclosure of which is incorporated by reference herein.

[0105] The compressed-air energy storage and recovery systems described herein are preferably "open-air" systems, i.e., systems that take in air from the ambient atmosphere for compression and vent air back to the ambient atmosphere after expansion, rather than systems that compress and expand a captured volume of gas in a sealed container (i.e., "closed-air" systems). The systems described herein generally feature one or more cylinder assemblies for the storage and recovery of energy via compression and expansion of gas. The systems also include (i) a reservoir for storage of compressed gas after compression and supply of compressed gas for expansion thereof, and (ii) a vent for exhausting expanded gas to atmosphere after expansion and supply of gas for compression. The storage reservoir may include or consist essentially of, e.g., one or more one or more pressure vessels (i.e., containers for compressed gas that may have rigid exteriors or may be inflatable, that may be formed of various suitable materials such as metal or plastic, and that may or may not fall within ASME regulations for pressure vessels), pipes (i.e., rigid containers for compressed gas that may also function as and/or be rated as fluid conduits, have lengths well in excess (e.g., >100×) of their diameters, and do not fall within ASME regulations for pressure vessels), or caverns (i.e., naturally occurring or artificially created cavities that are typically located underground). Open-air systems typically provide superior energy density relative to closed-air systems.

[0106] Furthermore, the systems described herein may be advantageously utilized to harness and recover sources of renewable energy, e.g., wind and solar energy. For example, energy stored during compression of the gas may originate from an intermittent renewable energy source of, e.g., wind or solar energy, and energy may be recovered via expansion of the gas when the intermittent renewable energy source is nonfunctional (i.e., either not producing harnessable energy or producing energy at lower-than-nominal levels). As such, the systems described herein may be connected to, e.g., solar panels or wind turbines, in order to store the renewable energy generated by such systems.

[0107] In an aspect, embodiments of the invention feature a method for storing energy in and/or recovering energy with an energy-storage system comprising (i) a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port, the valve comprising a valve member for occluding the gated port, and (ii) an actuation system for actuating the valve, the actuation system comprising (a) an actuation cylinder and (b) a piston disposed within and dividing the actuation cylinder into first and second chambers. Within the cylinder assembly, gas is compressed to store energy and/or gas is expanded to recover energy. Prior to, during, and/or after the compression and/or expansion, fluid is admitted into and/or fluid is exhausted from the cylinder assembly at least in part by actuating the valve from a closed state to an open state by admitting fluid into the first chamber of the actuation cylinder to increase fluid pressure therein, thereby moving the piston toward the second chamber. During the actuation, fluid exits the second chamber of the actuation cylinder at a first rate to maximize speed of the piston motion, and thereafter, fluid exits the second chamber at a second rate slower than the first rate to decelerate the piston before the piston reaches an end surface of the actuation cylinder.

[0108] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The second rate of fluid flow may decrease as the piston moves toward the end surface of the actuation cylinder. During the actuation, the piston may occlude at least a portion of an orifice in the second chamber as the piston moves toward an end surface of the actuation cylinder, thereby slowing the flow of fluid from the second chamber from the first rate to the second rate. When the piston is disposed proximate the end surface (e.g., at one limit of the piston's travel within the actuation cylinder), the orifice may be completely occluded by the piston. A lateral dimension of at least a portion of the orifice may vary as a function of distance from the end surface of the actuation cylinder. A lateral dimension of a first portion of the orifice may not vary as a function of distance from the end surface of the actuation cylinder, and a lateral dimension of a second portion of the orifice may vary as a function of distance from the end surface of the actuation cylinder. A lateral boundary of at least a portion of the orifice may have a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the orifice is partially occluded. Fluid may be admitted into the first chamber through both (i) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder. During at least a portion of the actuation, fluid may exit the second chamber through both (i) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

[0109] In another aspect, embodiments of the invention feature a method for at least one of storing energy in or recovering energy with an energy-storage system comprising (i) a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port, the valve comprising a valve member for occluding the gated port, and (ii) an actuation system for actuating the valve, the actuation system comprising (a) an actuation cylinder, (b) a piston disposed within and dividing the actuation cylinder into first and second chambers, and (c) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder. Within the cylinder assembly, gas is compressed to store energy and/or gas is expanded to recover energy. Prior to, during, and/or after the compression and/or expansion, fluid is admitted into and/or fluid is exhausted from the cylinder assembly at least in part by actuating the valve from a closed state to an open state by admitting fluid into the first chamber of the actuation cylinder to increase fluid pressure therein, thereby moving the piston toward the second chamber. During the actuation, (i) fluid flows out of the second chamber through the occludable orifice unoccluded by the piston, thereby maximizing speed of the piston motion, and (ii) thereafter, the piston occludes at least a portion of the occludable orifice, whereby fluid flow from the second chamber is decreased to decelerate the piston before the piston reaches an end surface of the actuation cylinder.

[0110] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The occludable orifice may be completely occluded by the piston by the end of the actuation (e.g., at one limit of the piston's travel within the actuation cylinder). A lateral dimension of at least a portion of the occludable orifice may vary as a function of distance from an end surface of the actuation cylinder. A lateral dimension of a first portion of the occludable orifice may not vary as a function of distance from an end surface of the actuation cylinder, and a lateral dimension of a second portion of the occludable orifice may vary as a function of distance from the end surface of the actuation cylinder. A lateral boundary of at least a portion of the occludable orifice may have a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the occludable orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the occludable orifice is partially occluded. Fluid may be admitted into the first chamber through both (i) a second occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder. During at least a portion of the actuation, fluid may exit the second chamber through both (i) the occludable orifice, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

[0111] In yet another aspect, embodiments of the invention feature an energy storage and recovery system that includes or consists essentially of a cylinder assembly (i) for, therewithin, compression of gas to store energy and/or expansion of gas to recover energy and (ii) having an interior compartment, a valve for admitting fluid into the interior compartment and/or exhausting fluid from the interior compartment through a gated port, and an actuation mechanism for actuating the valve. The valve includes a valve member for occluding the gated port. The actuation mechanism includes or consists essentially of (i) an actuation cylinder having a lateral surface and two opposing end surfaces, (ii) a piston disposed within and dividing the actuation cylinder into two chambers, the valve being configured for actuation by a difference in fluid pressure between the two chambers, and (iii) an occludable orifice defined by the lateral surface and configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder.

[0112] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The occludable orifice may be configured to be completely occluded by the piston when the piston is disposed proximate the end surface defining the fixed orifice (e.g., at one limit of the piston's travel within the actuation cylinder). A portion of the occludable orifice may be configured to not be occluded by the piston when the piston is disposed proximate an end surface of the actuation cylinder (e.g., at one limit of the piston's travel within the actuation cylinder). A lateral dimension of at least a portion of the occludable orifice may vary as a function of distance from one of the end surfaces of the actuation cylinder. A lateral dimension of a first portion of the occludable orifice may not vary as a function of distance from one of the end surfaces of the actuation cylinder, and a lateral dimension of a second portion of the occludable orifice may vary as a function of distance from one of the end surfaces of the actuation cylinder. A lateral boundary of at least a portion of the occludable orifice may have a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the occludable orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the occludable orifice is partially occluded. The actuation mechanism may include a fixed orifice defined by one of the end surfaces of the actuation cylinder. The system may include a high-pressure fluid source selectively connectable to both the occludable orifice and the fixed orifice. The system may include, disposed within a connection between the high-pressure fluid source and the fixed orifice, a check valve configured to enable substantially unrestricted flow of fluid to the fixed orifice when the occludable orifice is at least partially occluded by the piston. The system may include a low-pressure fluid reservoir selectively connectable to both the occludable orifice and the fixed orifice. A valve may connect the occludable orifice and the fixed orifice to (i) the high-pressure fluid source, (ii) the low-pressure fluid reservoir, or (iii) a chamber of the actuation cylinder opposite a chamber of the actuation cylinder in which the occludable orifice and fixed orifice are defined. A stem may be mechanically connected to the valve member and the piston. The occludable orifice and fixed orifice may be defined in one of the chambers of the actuation cylinder, and in the other chamber of the actuation cylinder, an end surface may define a second fixed orifice, and the lateral surface of the actuation cylinder may define a second occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder.

[0113] The cylinder assembly may be configured to compress gas from an initial pressure to a final pressure, and the system may include a control system. The control system may be configured to (i) pre-expand gas in the cylinder assembly to approximately the initial pressure, (ii) following the pre-expansion, admit gas at the initial pressure into the cylinder assembly, the pre-expansion reducing coupling loss during the admission of gas, (iii) compress the gas in the cylinder assembly to the final pressure, (iv) complete a compression cycle by exhausting only a portion of the compressed gas out of the cylinder assembly, and (v) repeat the foregoing steps at least once, thereby performing at least one additional compression cycle. The gas admission and/or the gas exhaustion may occur through the gated port of the valve.

[0114] The cylinder assembly may be configured to expand gas from an initial pressure to a final pressure, and the system may include a control system. The control system may be configured to (i) pre-compress gas in the cylinder assembly to approximately the initial pressure, (ii) following the pre-compression, admit compressed gas at the initial pressure into the cylinder assembly, the pre-compression reducing coupling loss during the admission of compressed gas, (iii) expand the gas in the cylinder assembly to the final pressure, (iv) complete an expansion cycle by exhausting only a portion of the expanded gas out of the cylinder assembly, and (v) repeat the foregoing steps at least once, thereby performing at least one additional expansion cycle. The gas admission and/or the gas exhaustion may occur through the gated port of the valve.

[0115] The system may include a high-side component, selectively fluidly connected to the cylinder assembly, for (i) supplying gas to the cylinder assembly for expansion therein and/or (ii) accepting gas from the cylinder assembly after compression therein, a low-side component, selectively fluidly connected to the cylinder assembly, for (i) supplying gas to the cylinder assembly for compression therein and/or (ii) accepting gas from the cylinder assembly after expansion therein, and a control system for operating the cylinder assembly to perform (i) a pre-compression of gas therewithin prior to admission therein of gas for expansion, thereby reducing coupling loss between the cylinder assembly and the high-side component, and/or (ii) a pre-expansion of gas therewithin prior to admission therein of gas for compression, thereby reducing coupling loss between the cylinder assembly and the low-side component. The system may include a sensor for sensing a temperature, a pressure, and/or a position of a boundary mechanism within the cylinder assembly to generate control information. The control system may be responsive to the control information. The control system may be configured to operate the cylinder assembly, during (i) pre-compression of gas therewithin and/or (ii) expansion of gas therewithin, based at least in part on control information generated during (i) a previous gas expansion within the cylinder assembly and/or (ii) a previous pre-compression of gas within the cylinder assembly. The control system may be configured to operate the cylinder assembly, during (i) pre-expansion of gas therewithin and/or (ii) compression of gas therewithin, based at least in part on control information generated during (i) a previous gas compression within the cylinder assembly and/or (ii) a previous pre-expansion of gas within the cylinder assembly. The high-side component may include or consist essentially of a compressed-gas storage reservoir or a second cylinder assembly for at least one of compressing gas or expanding gas within a pressure range higher than a pressure range of operation of the cylinder assembly. The system may include a second cylinder assembly for compressing gas and/or expanding gas within a pressure range higher than a pressure range of operation of the cylinder assembly, and the high-side component may include or consist essentially of a mid-pressure vessel for containing gas at a pressure within both of or between pressure ranges of operation of the cylinder assembly and the second cylinder assembly. The low-side component may include or consist essentially of a vent to atmosphere or a second cylinder assembly for compressing gas and/or expanding gas within a pressure range lower than a pressure range of operation of the cylinder assembly. The system may include a second cylinder assembly for compressing gas and/or expanding gas within a pressure range lower than a pressure range of operation of the cylinder assembly, and the low-side component may include or consist essentially of a mid-pressure vessel for containing gas at a pressure within both of or between pressure ranges of operation of the cylinder assembly and the second cylinder assembly.

[0116] In a further aspect, embodiments of the invention feature an energy storage and recovery system including or consisting essentially of a cylinder assembly (i) for, therewithin, at least one of compression of gas to store energy or expansion of gas to recover energy and (ii) having an interior compartment, a valve for admitting fluid into the interior compartment and/or exhausting fluid from the interior compartment through a gated port, and an actuation mechanism for actuating the valve. The valve includes a valve member for occluding the gated port. The actuation mechanism includes or consists essentially of (i) an actuation cylinder having a lateral surface and first and second opposing end surfaces, (ii) a piston disposed within and dividing the actuation cylinder into first and second chambers, a difference in fluid pressure between the two chambers actuating the valve, (iii) within the first chamber, a first occludable orifice defined by the lateral surface and configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, (iv) within the first chamber, a first fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder, (v) within the second chamber, a second occludable orifice defined by the lateral surface and configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (vi) within the second chamber, a second fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

[0117] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The first fixed orifice may be defined by the first end surface of the actuation cylinder and/or the second fixed orifice may be defined by the second end surface of the actuation cylinder. The first occludable orifice may be configured to be completely occluded by the piston when the piston is disposed proximate the first end surface. The second occludable orifice may be configured to be completely occluded by the piston when the piston is disposed proximate the second end surface. A lateral dimension of at least a portion of the first occludable orifice may vary as a function of distance from the first end surface. A lateral dimension of a first portion of the first occludable orifice may not vary as a function of distance from the first end surface, and a lateral dimension of a second portion of the first occludable orifice may vary as a function of distance from the first end surface. A lateral boundary of at least a portion of the first occludable orifice may have a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the first occludable orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the first occludable orifice is partially occluded. The system may include a high-pressure fluid source selectively connectable to (i) both the first occludable orifice and the first fixed orifice or (ii) both the second occludable orifice and the second fixed orifice. The system may include, disposed within a connection between the high-pressure fluid source and the first fixed orifice, a first check valve configured to enable substantially unrestricted flow of fluid to the first fixed orifice when the first occludable orifice is at least partially occluded by the piston. The system may include, disposed within a connection between the high-pressure fluid source and the second fixed orifice, a second check valve configured to enable substantially unrestricted flow of fluid to the second fixed orifice when the second occludable orifice is at least partially occluded by the piston. The system may include a low-pressure fluid reservoir selectively connectable to (i) both the first occludable orifice and the first fixed orifice or (ii) both the second occludable orifice and the second fixed orifice. The system may include a valve having different settings for connecting (i) the first occludable orifice and the first fixed orifice to the high-pressure fluid source, and the second occludable orifice and the second fixed orifice to the low-pressure fluid reservoir, (ii) the first occludable orifice and the first fixed orifice to the low-pressure fluid reservoir, and the second occludable orifice and the second fixed orifice to the high-pressure fluid source, or (iii) the first occludable orifice and the first fixed orifice to the second occludable orifice and the second fixed orifice. A stem may be mechanically connected to the valve member and the piston.

[0118] The cylinder assembly may be configured to compress gas from an initial pressure to a final pressure, and the system may include a control system. The control system may be configured to (i) pre-expand gas in the cylinder assembly to approximately the initial pressure, (ii) following the pre-expansion, admit gas at the initial pressure into the cylinder assembly, the pre-expansion reducing coupling loss during the admission of gas, (iii) compress the gas in the cylinder assembly to the final pressure, (iv) complete a compression cycle by exhausting only a portion of the compressed gas out of the cylinder assembly, and (v) repeat the foregoing steps at least once, thereby performing at least one additional compression cycle. The gas admission and/or the gas exhaustion may occur through the gated port of the valve.

[0119] The cylinder assembly may be configured to expand gas from an initial pressure to a final pressure, and the system may include a control system. The control system may be configured to (i) pre-compress gas in the cylinder assembly to approximately the initial pressure, (ii) following the pre-compression, admit compressed gas at the initial pressure into the cylinder assembly, the pre-compression reducing coupling loss during the admission of compressed gas, (iii) expand the gas in the cylinder assembly to the final pressure, (iv) complete an expansion cycle by exhausting only a portion of the expanded gas out of the cylinder assembly, and (v) repeat the foregoing steps at least once, thereby performing at least one additional expansion cycle. The gas admission and/or the gas exhaustion may occur through the gated port of the valve.

[0120] The system may include a high-side component, selectively fluidly connected to the cylinder assembly, for (i) supplying gas to the cylinder assembly for expansion therein and/or (ii) accepting gas from the cylinder assembly after compression therein, a low-side component, selectively fluidly connected to the cylinder assembly, for (i) supplying gas to the cylinder assembly for compression therein and/or (ii) accepting gas from the cylinder assembly after expansion therein, and a control system for operating the cylinder assembly to perform (i) a pre-compression of gas therewithin prior to admission therein of gas for expansion, thereby reducing coupling loss between the cylinder assembly and the high-side component, and/or (ii) a pre-expansion of gas therewithin prior to admission therein of gas for compression, thereby reducing coupling loss between the cylinder assembly and the low-side component. The system may include a sensor for sensing a temperature, a pressure, and/or a position of a boundary mechanism within the cylinder assembly to generate control information. The control system may be responsive to the control information. The control system may be configured to operate the cylinder assembly, during (i) pre-compression of gas therewithin and/or (ii) expansion of gas therewithin, based at least in part on control information generated during (i) a previous gas expansion within the cylinder assembly and/or (ii) a previous pre-compression of gas within the cylinder assembly. The control system may be configured to operate the cylinder assembly, during (i) pre-expansion of gas therewithin and/or (ii) compression of gas therewithin, based at least in part on control information generated during (i) a previous gas compression within the cylinder assembly and/or (ii) a previous pre-expansion of gas within the cylinder assembly. The high-side component may include or consist essentially of a compressed-gas storage reservoir or a second cylinder assembly for at least one of compressing gas or expanding gas within a pressure range higher than a pressure range of operation of the cylinder assembly. The system may include a second cylinder assembly for compressing gas and/or expanding gas within a pressure range higher than a pressure range of operation of the cylinder assembly, and the high-side component may include or consist essentially of a mid-pressure vessel for containing gas at a pressure within both of or between pressure ranges of operation of the cylinder assembly and the second cylinder assembly. The low-side component may include or consist essentially of a vent to atmosphere or a second cylinder assembly for compressing gas and/or expanding gas within a pressure range lower than a pressure range of operation of the cylinder assembly. The system may include a second cylinder assembly for compressing gas and/or expanding gas within a pressure range lower than a pressure range of operation of the cylinder assembly, and the low-side component may include or consist essentially of a mid-pressure vessel for containing gas at a pressure within both of or between pressure ranges of operation of the cylinder assembly and the second cylinder assembly.

[0121] In one aspect, embodiments of the invention feature a method for storing energy in and/or recovering energy with an energy-storage system comprising a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port. The valve includes a valve member for occluding the gated port and having a width W greater than or substantially equal to a width of the gated port. Within the cylinder assembly, gas is compressed to store energy and/or gas is expanded to recover energy. Prior to, during, and/or after the compression and/or expansion, fluid is admitted into and/or fluid is exhausted from the cylinder assembly at least in part by actuating the valve from a closed state to an open state. The actuation includes or consists essentially of (A) accelerating the valve member from a closed position such that the valve member attains a maximum velocity at a distance away from the gated port less than or substantially equal to a substantially-open position, wherein (i) a curtain area available for flow through the gated port at the substantially-open position is approximately equal to an area of the gated port, and (ii) the valve member continues to move past the substantially-open position to a full-open position farther away from the gated port than the substantially-open position, and/or (B) moving the valve member to the substantially-open position and, thereafter, (i) moving the valve member to the full-open position, and (ii) during at least a portion of the movement of the valve member from the substantially-open position to the full-open position, decelerating the valve member such that a velocity of the valve member is approximately zero when the valve member reaches the full-open position.

[0122] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The surface of the valve member facing the gated port may be circular with a diameter equal to W. In the substantially-open position, the valve member may be a distance W/4 away from the gated port. Accelerating the valve member from the closed position may include recovering at least a portion of energy stored during a prior closure of the valve. When the valve member is decelerated, at least a portion of the kinetic energy of the valve member may be stored during the deceleration. The energy may be stored as potential energy (e.g., spring potential energy and/or hydraulic-pressure potential energy). Accelerating the valve member from the closed position may include moving the valve member a finite distance from a fully-closed position to a sufficiently-closed position at which the valve member remains in contact with at least a portion of a seat disposed at the gated port. At least a portion of the seat may move in concert with the valve member as the valve member moves from the fully-closed position to the sufficiently-closed position. For a particular differential pressure, flow through the gated port when the valve member is in the sufficiently-closed position may be less than 1% of flow through the gated port when the valve member is in the sufficiently-open position.

[0123] In another aspect, embodiments of the invention feature a method for storing energy in and/or recovering energy with an energy-storage system comprising a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port. The valve includes a valve member for occluding the gated port and having a width W greater than or substantially equal to a width of the gated port. Within the cylinder assembly, gas is compressed to store energy and/or gas is expanded to recover energy. Prior to, during, and/or after the compression and/or expansion, the gated port is occluded by actuating the valve from an open state to a closed state. The actuation includes or consists essentially of (A) accelerating the valve member from a full-open position such that the valve member attains a maximum velocity at a distance away from the gated port less than or substantially equal to a substantially-open position, wherein (i) a curtain area available for flow through the gated port at the substantially-open position is approximately equal to an area of the gated port, and (ii) the valve member continues to move past the substantially-open position to a substantially-closed position at which the valve member contacts at least a portion of a seat disposed at the gated port, and/or (B) moving the valve member to the substantially-closed position and, thereafter, (i) moving the valve member to a fully-closed position beyond the substantially-closed position, and (ii) during at least a portion of the movement of the valve member from the substantially-closed position and the fully-closed position, decelerating the valve member to a velocity of approximately zero when the valve member reaches the fully-closed position, the valve member remaining in contact with the at least a portion of the seat in the substantially-closed position.

[0124] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The surface of the valve member facing the gated port may be circular with a diameter equal to W. In the substantially-open position, the valve member may be a distance W/4 away from the gated port. Accelerating the valve member from the full-open position may include recovering at least a portion of energy stored during a prior opening of the valve. When the valve member is decelerated, at least a portion of the kinetic energy of the valve member may be stored during the deceleration. The energy may be stored as potential energy (e.g., spring potential energy and/or hydraulic-pressure potential energy). The at least a portion of the seat may move in concert with the valve member as the valve member moves from the sufficiently-closed position to the fully-closed position. For a particular differential pressure, flow through the gated port when the valve member is in the sufficiently-closed position may be less than 1% of flow through the gated port when the valve member is in the sufficiently-open position. After the valve member reaches the fully-closed position, the valve member may be restored from the fully-closed position to the sufficiently-closed position while maintaining contact between the valve member and the at least a portion of the seat.

[0125] In yet another aspect, embodiments of the invention feature an energy storage and recovery system including or consisting essentially of a cylinder assembly (i) for, therewithin, at least one of compression of gas to store energy or expansion of gas to recover energy and (ii) having an interior compartment, a valve, and an actuation mechanism for actuating the valve. The valve admits fluid into the interior compartment and/or exhausts fluid from the interior compartment through a gated port, and the valve includes a valve member for occluding the gated port and having a width W greater than or substantially equal to a width of the gated port. The gated port includes a seat having a contact portion and, connected thereto, a shock-absorbing mechanism for (i) accelerating the valve member away from the seat, (ii) decelerating the valve member upon contact with the contact portion, and/or (iii) storing kinetic energy of the valve member as potential energy.

[0126] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The shock-absorbing mechanism may include or consist essentially of a wave spring, a coil spring, an air spring, and/or an elastic material (e.g., an elastomer). The profile of the contact portion may be complementary to the profile of the valve member such that, upon contact between the valve member and the contact ring, the gated port is substantially occluded. The contact portion may be beveled. The contact portion may include or consist essentially of polyether ether ketone. The gated port may be disposed within an end cap of the cylinder assembly. The actuation mechanism may be hydraulic, electrical, mechanical, and/or magnetic. The contact portion may be movable to induce or relieve pressure on the shock-absorbing material. A gasket may be disposed around the contact portion and may prevent fluid flow between the contact portion and the interior compartment of the cylinder assembly at least when the gated port is occluded by the valve member. The contact portion may include or consist essentially of an annular contact ring. The valve may be a high-side valve or a low-side valve. The valve member may be shaped as a truncated cone.

[0127] The actuation mechanism may include or consist essentially of a hydraulic cylinder containing a piston dividing an interior of the hydraulic cylinder into two chambers, a stem mechanically linking the valve member and the piston, a circulation mechanism for supplying fluid to at least one of the chambers, and a control mechanism for controlling fluid flow to, from, and between the two chambers. The difference in fluid pressure between the two chambers exerts pressure on the piston to actuate the valve. The actuation mechanism may include, selectively fluidly connected to each of the two chambers and to the circulation mechanism, a high-pressure accumulator for storing fluid at a pressure approximately equal to or greater than a pressure supplied by the circulation mechanism. The control mechanism may be configured to admit, during actuation of the valve, fluid from both the circulation mechanism and the high-pressure accumulator into one of the two chambers. The actuation mechanism may include, selectively fluidly connected to each of the two chambers and to the circulation mechanism, a low-pressure accumulator for storing fluid at a pressure approximately equal to or less than a pressure supplied by the circulation mechanism. A fluid reservoir, different from the low-pressure accumulator, may be fluidly connected to the circulation mechanism. The control mechanism may be configured to admit, during actuation of the valve, fluid from one of the two chambers to both the low-pressure accumulator and the fluid reservoir. The system may include a connection between the low-pressure accumulator and the fluid reservoir. The connection may include a pressure-relief valve configured to allow fluid flow from the low-pressure accumulator to the fluid reservoir when a pressure of the low-pressure accumulator exceeds a threshold pressure. The control mechanism may include or consist essentially of a three-way directional control valve having different settings that (i) fluidly connect the circulation mechanism with a first one of the two chambers, (ii) fluidly connect the circulation mechanism with a second one of the two chambers different from the first one, and (iii) fluidly connect the two chambers together. The system may include a control system configured to, in order to actuate the valve, (i) set the three-way directional control valve to one of the settings fluidly connecting the circulation mechanism with either the first or second chamber, thereby causing the piston in the hydraulic cylinder to move along a stroke length defined by a length of the hydraulic cylinder, and (ii) before the piston in the hydraulic cylinder moves along an entirety of the stroke length, set the three-way directional control valve to the setting fluidly connecting the first and second chambers together.

[0128] In a further aspect, embodiments of the invention feature an energy storage and recovery system including or consisting essentially of a cylinder assembly (i) for, therewithin, at least one of compression of gas to store energy or expansion of gas to recover energy and (ii) having an interior compartment, a valve, and a hydraulic actuation mechanism for actuating the valve. The valve admits fluid into the interior compartment and/or exhausts fluid from the interior compartment through a gated port, and includes a valve member for occluding the gated port and having a width W greater than or substantially equal to a width of the gated port. The hydraulic actuation mechanism includes a control mechanism for controlling fluid flow to, from, and between two chambers of a hydraulic cylinder, where the difference in fluid pressure between the two chambers actuates the valve.

[0129] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The hydraulic cylinder may contain a piston dividing the interior of the hydraulic cylinder into two chambers. The actuation mechanism may include (i) a stem mechanically linking the valve member and the piston, and (ii) a circulation mechanism for supplying fluid to at least one of the chambers of the hydraulic cylinder. The actuation mechanism may include, selectively fluidly connected to each of the two chambers and to the circulation mechanism, a high-pressure accumulator for storing fluid at a pressure approximately equal to or greater than a pressure supplied by the circulation mechanism. The control mechanism may be configured to admit, during actuation of the valve, fluid from both the circulation mechanism and the high-pressure accumulator into one of the two chambers. The actuation mechanism may include, selectively fluidly connected to each of the two chambers and to the circulation mechanism, a low-pressure accumulator for storing fluid at a pressure approximately equal to or less than a pressure supplied by the circulation mechanism. The system may include a fluid reservoir, different from the low-pressure accumulator, fluidly connected to the circulation mechanism. The control mechanism may be configured to admit, during actuation of the valve, fluid from one of the two chambers to both the low-pressure accumulator and the fluid reservoir. The system may include a connection between the low-pressure accumulator and the fluid reservoir. The connection may include a pressure-relief valve configured to allow fluid flow from the low-pressure accumulator to the fluid reservoir when a pressure of the low-pressure accumulator exceeds a threshold pressure. The control mechanism may include or consist essentially of a three-way directional control valve having settings (i) fluidly connecting the circulation mechanism with a first one of the two chambers, (ii) fluidly connecting the circulation mechanism with a second one of the two chambers different from the first one, or (iii) fluidly connecting the two chambers together. The system may include a control system configured to, in order to actuate the valve, (i) set the three-way directional control valve to one of the settings fluidly connecting the circulation mechanism with either the first or second chamber, thereby causing the piston in the hydraulic cylinder to move along a stroke length defined by a length of the hydraulic cylinder, and (ii) before the piston in the hydraulic cylinder moves along an entirety of the stroke length, set the three-way directional control valve to the setting fluidly connecting the first and second chambers together.

[0130] In an aspect, embodiments of the invention feature an energy storage and recovery system including or consisting essentially of a cylinder assembly (i) for, therewithin, at least one of compression of gas to store energy or expansion of gas to recover energy and (ii) having an interior compartment, a valve for admitting fluid into the interior compartment and/or exhausting fluid from the interior compartment, and an actuation mechanism for actuating the valve with a magnetic actuation force (i.e., magnetically).

[0131] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The valve may include or consist essentially of a gated port and a valve member for selectively controlling at least one of fluid flow into the interior compartment or fluid flow out of the interior compartment. The valve member may be disposed between the gated port and at least a portion of the interior compartment of the cylinder assembly. The gated port may be disposed between the valve member and the interior compartment of the cylinder assembly. The gated port may include a seat having a shape complimentary to a shape of the valve member, thereby enabling the closure of the gated port when the valve member is in contact with the seat. The valve member may include or consist essentially of a permanent magnet and/or an electromagnet. A stem may extend through the actuation mechanism and connect to the valve member. A permanent magnet and/or an electromagnet may be proximate the actuation mechanism and connected to the stem. The actuation mechanism may include or consist essentially of a permanent magnet and/or an electromagnet. The system may include a control system for controlling the magnetic actuation force in response to a position of a valve member of the valve and/or a difference between a pressure inside the interior compartment and a pressure inside the valve. The valve may be configured to check closed, thereby preventing fluid flow into or out of the interior compartment, in the absence of the magnetic actuation force. The system may include a mechanical or pneumatic spring for biasing the valve toward closing, cushioning opening forces, and/or providing at least a portion of a closing actuation force. The valve may be configured to control fluid flow between the interior compartment and (i) a compressed-gas storage reservoir or (ii) a second cylinder assembly for the expansion and/or compression of gas at a pressure range higher than that for which the cylinder assembly is configured. The valve may be configured to control fluid flow between the interior compartment and (i) a vent to atmosphere or (ii) a second cylinder assembly for the expansion and/or compression of gas at a pressure range lower than that for which the cylinder assembly is configured. The actuation mechanism and at least a portion of the valve may be integrated within an end cap of the cylinder assembly.

[0132] In another aspect, embodiments of the invention feature a method for energy storage and recovery. Within a cylinder assembly, gas is compressed to store energy and/or gas is expanded to recover energy. At least one of prior to, during, or after the compression and/or expansion, fluid is admitted into and/or fluid is exhausted from the cylinder assembly at least in part by actuating a valve with a magnetic actuation force (i.e., magnetically).

[0133] Embodiments of the invention may include one or more of the following in any of a variety of combinations. The admission and/or exhaustion of fluid may be initiated, maintained, and/or concluded by a hydraulic force resulting from a difference in pressure inside and outside of the cylinder assembly. Actuating the valve may include or consist essentially of applying the magnetic actuation force to act in concert with the hydraulic force. The hydraulic force may at least partially open the valve, and the magnetic actuation force may maintain the valve in an open position. The hydraulic force may at least partially close the valve, and the magnetic actuation force may maintain the valve in a closed position. Actuating the valve may include or consist essentially of applying the magnetic actuation force to act in opposition to the hydraulic force. The magnetic actuation force may reduce collision force between the actuation mechanism applying the magnetic actuation force and a valve member of the valve. Actuating the valve may include or consist essentially of applying a magnetic actuation force that varies over time during the actuation of the valve. The method may include, with a mechanical force, biasing the valve toward closing, cushioning opening forces, and/or providing at least a portion of a closing actuation force.

[0134] These and other objects, along with advantages and features of the invention, will become more apparent through reference to the following description, the accompanying drawings, and the claims. Furthermore, it is to be understood that the features of the various embodiments described herein are not mutually exclusive and can exist in various combinations and permutations. Note that as used herein, the terms "pipe," "piping" and the like shall refer to one or more conduits that are rated to carry gas or liquid between two points. Thus, the singular term should be taken to include a plurality of parallel conduits where appropriate. Herein, the terms "liquid" and "water" interchangeably connote any mostly or substantially incompressible liquid, the terms "gas" and "air" are used interchangeably, and the term "fluid" may refer to a liquid, a gas, or a mixture of liquid and gas (e.g., a foam) unless otherwise indicated. As used herein unless otherwise indicated, the terms "approximately" and "substantially" mean±10%, and, in some embodiments, ±5%. A "valve" is any mechanism or component for controlling fluid communication between fluid paths or reservoirs, or for selectively permitting control or venting. The term "cylinder" refers to a chamber, of uniform but not necessarily circular cross-section, which may contain a slidably disposed piston or other mechanism that separates the fluid on one side of the chamber from that on the other, preventing fluid movement from one side of the chamber to the other while allowing the transfer of force/pressure from one side of the chamber to the next or to a mechanism outside the chamber. At least one of the two ends of a chamber may be closed by end caps, also herein termed "heads." As utilized herein, an "end cap" is not necessarily a component distinct or separable from the remaining portion of the cylinder, but may refer to an end portion of the cylinder itself. Rods, valves, and other devices may pass through the end caps. A "cylinder assembly" may be a simple cylinder or include multiple cylinders, and may or may not have additional associated components (such as mechanical linkages among the cylinders). The shaft of a cylinder may be coupled hydraulically or mechanically to a mechanical load (e.g., a hydraulic motor/pump or a crankshaft) that is in turn coupled to an electrical load (e.g., rotary or linear electric motor/generator attached to power electronics and/or directly to the grid or other loads), as described in the '678 and '842 patents. As used herein, "thermal conditioning" of a heat-exchange fluid does not include any modification of the temperature of the heat-exchange fluid resulting from interaction with gas with which the heat-exchange fluid is exchanging thermal energy; rather, such thermal conditioning generally refers to the modification of the temperature of the heat-exchange fluid by other means (e.g., an external heat exchanger). The terms "heat-exchange" and "heat-transfer" are generally utilized interchangeably herein. Unless otherwise indicated, motor/pumps described herein are not required to be configured to function both as a motor and a pump if they are utilized during system operation only as a motor or a pump but not both. Gas expansions described herein may be performed in the absence of combustion (as opposed to the operation of an internal-combustion cylinder, for example).

BRIEF DESCRIPTION OF THE DRAWINGS

[0135] In the drawings, like reference characters generally refer to the same parts throughout the different views. Cylinders, rods, and other components are depicted in cross section in a manner that will be intelligible to all persons familiar with the art of pneumatic and hydraulic cylinders. Also, the drawings are not necessarily to scale, emphasis instead generally being placed upon illustrating the principles of the invention. In the following description, various embodiments of the present invention are described with reference to the following drawings, in which:

[0136] FIG. **1** is a schematic drawing of a compressed-gas energy storage system in accordance with various embodiments of the invention;

[0137] FIG. **2** is a schematic drawing of various components of a compressed-gas energy storage system in accordance with various embodiments of the invention;

[0138] FIG. **3** is a schematic drawing of the major components of a compressed air energy storage and recovery system in accordance with various embodiments of the invention;

[0139] FIG. **4** is a schematic drawing of various components of a multi-cylinder compressed-gas energy storage system in accordance with various embodiments of the invention;

[0140] FIG. **5** is a schematic drawing of a cylinder assembly with apparatus for the generation of foam external to the cylinder in accordance with various embodiments of the invention;

[0141] FIG. **6** is a schematic drawing of a cylinder assembly with apparatus for the generation of foam external to the cylinder and provision for bypassing the foam-generating apparatus in accordance with various embodiments of the invention;

[0142] FIG. **7** is a schematic drawing of a cylinder assembly with apparatus for the generation of foam in a vessel external to the cylinder in accordance with various embodiments of the invention;

[0143] FIG. **8** is a schematic drawing of a cylinder assembly with apparatus for the generation of foam internal to the cylinder in accordance with various embodiments of the invention;

[0144] FIG. **9** is a schematic drawing of a compressed-air energy storage system employing multiple pairs of high- and low-pressure cylinders in accordance with various embodiments of the invention;

[0145] FIG. **10** is an illustrative plot of pressure as a function of time for four different expansion scenarios in accordance with various embodiments of the invention;

[0146] FIG. **11** is a graphical display of experimental test data in accordance with various embodiments of the invention;

[0147] FIG. **12** is an illustrative plot of the ideal pressure-volume cycle in a cylinder operated as either a compressor or expander;

[0148] FIG. **13** is an illustrative plot of cylinder chamber pressure as a function of cylinder chamber volume for three different expansion scenarios in an illustrative CAES system in accordance with various embodiments of the invention;

[0149] FIGS. **14**A-**14**C are illustrative plots of cylinder chamber pressure as a function of cylinder chamber volume for different expansion scenarios in an illustrative CAES system in accordance with various embodiments of the invention;

[0150] FIG. **15** is an illustrative plot of cylinder chamber pressure as a function of cylinder chamber volume for three different compression scenarios in an illustrative CAES system in accordance with various embodiments of the invention;

[0151] FIG. **16** is an illustrative plot of cylinder chamber pressure as a function of cylinder chamber volume for three different compression scenarios in an illustrative CAES system in accordance with various embodiments of the invention;

[0152] FIG. **17**A is a schematic drawing of the major components of a low-side poppet valve in accordance with various embodiments of the invention;

[0153] FIG. **17**B is a schematic drawing of the valve of FIG. **17**A in a different state of operation;

[0154] FIG. **18**A is a schematic drawing of the major components of a high-side poppet valve in accordance with various embodiments of the invention;

[0155] FIG. **18**B is a schematic drawing of the valve of FIG. **18**A in a different state of operation;

[0156] FIG. **19**A is a schematic drawing of a cylinder assembly with a high-side valve and a low-side valve integrated into the head of the cylinder in accordance with various embodiments of the invention;

[0157] FIG. **19**B is a schematic drawing of the assembly of FIG. **19**A in a different state of operation;

[0158] FIG. **20** is a schematic drawing of various components of a hydraulic activation cylinder and a poppet valve in accordance with various embodiments of the invention;

[0159] FIG. **21** is a schematic drawing of a system incorporating a high-side valve having a member that is decelerated during closure by a time-varying hydraulic resistance in accordance with various embodiments of the invention;

[0160] FIG. **22** is a schematic drawing of a system incorporating a low-side valve having a member that is decelerated during closure by a time-varying hydraulic resistance in accordance with various embodiments of the invention;

[0161] FIG. **23**A is a schematic drawing of an activation cylinder with contrivances for governing fluid flow into and out of the cylinder in accordance with various embodiments of the invention;

[0162] FIGS. **23**B-**23**D are schematic drawings of the system of FIG. **23**A in different states of operation;

[0163] FIG. **24**A is a schematic drawing of various components of an activation cylinder having both a fixed orifice and occludable orifices in accordance with various embodiments of the invention;

[0164] FIGS. **24**B-**24**E are schematic drawings of the system of FIG. **24**A in different states of operation;

[0165] FIG. **24**F is a schematic drawing of various components of an activation cylinder having an occludable orifice in accordance with various embodiments of the invention;

[0166] FIG. **25** is a cross-section of an occludable orifice in accordance with various embodiments of the invention;

[0167] FIG. **26**A is a schematic drawing of a system incorporating a high-side valve and an activation cylinder with occludable orifices in accordance with various embodiments of the invention;

[0168] FIG. **26**B is an expanded view of the activation cylinder of FIG. **26**A;

[0169] FIG. **27** is a plot of the position of the spool of a hydraulic actuation cylinder in accordance with various embodiments of the invention;

[0170] FIG. **28** is a plot of the velocity of the spool of a hydraulic actuation cylinder in accordance with various embodiments of the invention;

[0171] FIG. **29** is a plot of the fluid pressure within one chamber of a hydraulic actuation cylinder in accordance with various embodiments of the invention;

[0172] FIG. **30**A is a schematic drawing of the major components of a high-side poppet valve in accordance with various embodiments of the invention;

[0173] FIG. **30**B is a depiction of an illustrative wave spring, such as might be employed in the valve of FIG. **30**A;

[0174] FIGS. **30**C and **30**D are schematic drawings of the valve of FIG. **30**A in different states of operation;

[0175] FIG. **31**A is an illustrative diagram of the position over time of the disc of a conventional valve;

[0176] FIG. **31**B is an illustrative diagram of the position over time of the disc of the valve of FIG. **30**A, in accordance with embodiments of the invention;

[0177] FIGS. **32**A-**32**C are schematic drawings of the valve of FIG. **30**A in different states of operation;

[0178] FIG. **33**A is a schematic drawing of the major components of the actuation mechanism of a high-side poppet valve in accordance with various embodiments of the invention;

[0179] FIGS. **33**B and **33**C are schematic drawings of the mechanism of FIG. **33**A in different states of operation;

[0180] FIG. **34**A is a schematic drawing of an electromagnetic valve in accordance with various embodiments of the invention;

[0181] FIG. **34**B is a schematic drawing of the valve of FIG. **34**A in a different state of operation;

[0182] FIG. **35**A is a schematic drawing of an electromagnetic valve in accordance with various embodiments of the invention; and

[0183] FIG. **35**B is a schematic drawing of the valve of FIG. **35**A in a different state of operation.

DETAILED DESCRIPTION

[0184] FIG. **1** depicts an illustrative system **100** that may be part of a larger system, not otherwise depicted, for the storage and release of energy. Subsequent figures will clarify the application of embodiments of the invention to such a system. The system **100** depicted in FIG. **1** features an assembly **101** for compressing and expanding gas. Expansion/compression assembly **101** may include or consist essentially of either one or more individual devices for expanding or compressing gas (e.g., turbines or cylinder assemblies that each may house a movable boundary mechanism) or a staged series of such devices, as well as ancillary devices (e.g., valves) not depicted explicitly in FIG. **1**.

[0185] An electric motor/generator **102** (e.g., a rotary or linear electric machine) is in physical communication (e.g., via hydraulic pump, piston shaft, or mechanical crankshaft) with the expansion/compression assembly **101**. The motor/generator **102** may be electrically connected to a source and/or sink of electric energy not explicitly depicted in FIG. **1** (e.g., an electrical distribution grid or a source of renewable energy such as one or more wind turbines or solar cells).

[0186] The expansion/compression assembly **101** may be in fluid communication with a heat-transfer subsystem **104** that alters the temperature and/or pressure of a fluid (i.e., gas, liquid, or gas-liquid mixture such as a foam) extracted from expansion/compression assembly **101** and, after alteration of the fluid's temperature and/or pressure, returns at least a portion of it to expansion/compression assembly **101**. Heat-transfer subsystem **104** may include pumps, valves, and other devices (not depicted explicitly in FIG. **1**) ancillary to its heat-transfer function and to the transfer of fluid to and from expansion/compression assembly **101**. Operated appropriately, the heat-transfer subsystem **104** enables substantially isothermal compression and/or expansion of gas inside expansion/compression assembly **101**.

[0187] Connected to the expansion/compression assembly **101** is a pipe **106** with a control valve **108** that controls a flow of fluid (e.g., gas) between assembly **101** and a storage reservoir **112** (e.g., one or more pressure vessels, pipes, and/or caverns). The storage reservoir **112** may be in fluid communication with a heat-transfer subsystem **114** that alters the temperature and/or pressure of fluid removed from storage reservoir **112** and, after alteration of the fluid's temperature and/or pressure, returns it to storage reservoir **112**. A second pipe **116** with a control valve **118** may be in fluid communication with the expansion/compression assembly **101** and with a vent **120** that communicates with a body of gas at relatively low pressure (e.g., the ambient atmosphere).

[0188] A control system **122** receives information inputs from any of expansion/compression assembly **101**, storage reservoir **112**, and other components of system **100** and sources external to system **100**. These information inputs may include or consist essentially of pressure, temperature, and/or other telemetered measurements of properties of components of system **101**. Such information inputs, here generically denoted by the letter "T," are transmitted to control system **122** either wirelessly or through wires. Such transmission is denoted in FIG. **1** by dotted lines **124**, **126**.

[0189] The control system **122** may selectively control valves **108** and **118** to enable substantially isothermal compression and/or expansion of a gas in assembly **101**. Control signals, here generically denoted by the letter "C," are transmitted to valves **108** and **118** either wirelessly or through wires. Such transmission is denoted in FIG. **1** by dashed lines **128**, **130**. The control system **122** may also control the operation of the heat-transfer assemblies **104**, **114** and of other components not explicitly depicted in FIG. **1**. The transmission of control and telemetry signals for these purposes is not explicitly depicted in FIG. **1**.

[0190] The control system **122** may be any acceptable control device with a human-machine interface. For example, the control system **122** may include a computer (for example a PC-type) that executes a stored control application in the form of a computer-readable software medium. More generally, control system **122** may be realized as software, hardware, or some combination thereof. For example, control system **122** may be implemented on one or more computers, such as a PC having a CPU board containing one or more processors such as the Pentium, Core, Atom, or Celeron family of processors manufactured by Intel Corporation of Santa Clara, Calif., the 680x0 and POWER PC family of processors manufactured by Motorola Corporation of Schaumburg, Ill., and/or the ATHLON line of processors manufactured by Advanced Micro Devices, Inc., of Sunnyvale, Calif. The processor may also include a main memory unit for storing programs and/or data relating to the methods described above. The memory may include random access memory (RAM), read only memory (ROM), and/or FLASH memory residing on commonly available hardware such as one or more application specific integrated circuits (ASIC), field programmable gate arrays (FPGA), electrically erasable programmable read-only memories (EEPROM), programmable read-only memories (PROM), programmable logic devices (PLD), or read-only memory devices (ROM). In some embodiments, the programs may be provided using external RAM and/or ROM such as optical disks, magnetic disks, or other storage devices.

[0191] For embodiments in which the functions of controller **122** are provided by software, the program may be written in any one of a number of high-level languages such as FORTRAN, PASCAL, JAVA, C, C++, C#, LISP, PERL, BASIC or any suitable programming language. Additionally, the software can be implemented in an assembly language and/or machine language directed to the microprocessor resident on a target device.

[0192] As described above, the control system **122** may receive telemetry from sensors monitoring various aspects of the operation of system **100**, and may provide signals to control valve actuators, valves, motors, and other electromechanical/electronic devices. Control system **122** may communicate with such sensors and/or other components of system **100** (and other embodiments described herein) via wired or wireless communication. An appropriate interface may be used to convert data from sensors into a form readable by the control system **122** (such as RS-232 or network-based interconnects). Likewise, the interface converts the computer's control signals into a form usable by valves and other actuators to perform an operation. The provision of such interfaces, as well as suitable control programming, is clear to those of ordinary skill in the art and may be provided without undue experimentation.

[0193] System **100** may be operated so as to compress gas admitted through the vent **120** and store the gas thus compressed in reservoir **112**. For example, in an initial state of operation, valve **108** is closed and valve **118** is open, admitting a quantity of gas into expansion/compression assembly **101**. When a desired quantity of gas has been admitted into assembly **101**, valve **118** may be closed. The motor/generator **102**, employing energy supplied by a source not explicitly depicted in FIG. **1** (e.g., the electrical grid), then provides mechanical power to expansion/compression assembly **101**, enabling the gas within assembly **101** to be compressed.

[0194] During compression of the gas within assembly **101**, fluid (i.e., gas, liquid, or a gas-liquid mixture) may be circulated between assembly **101** and heat-exchange assembly **104**. Heat-exchange assembly **104** may be operated in such a manner as to enable substantially isothermal compression of the gas within assembly **101**. During or after compression of the gas within assembly **101**, valve **108** may be opened to enable high-pressure fluid (e.g., compressed gas or a mixture of liquid and compressed gas) to flow to reservoir **112**. Heat-exchange assembly **114** may be operated at any time in such a manner as alter the temperature and/or pressure of the fluid within reservoir **112**.

[0195] That system **100** may also be operated so as to expand compressed gas from reservoir **112** in expansion/compression assembly **101** in such a manner as to deliver energy to the motor/generator **102** will be apparent to all persons familiar with the operation of pneumatic, hydraulic, and electric machines.

[0196] FIG. **2** depicts an illustrative system **200** that features a cylinder assembly **201** (i.e., an embodiment of assembly **101** in FIG. **1**) in communication with a reservoir **222** (**112** in FIG. **1**) and a vent to atmosphere **223** (**120** in FIG. **1**). In the illustrative system **200** shown in FIG. **2**, the cylinder assembly **201** contains a piston **202** slidably disposed therein. In some embodiments the piston **202** is replaced by a different boundary mechanism dividing cylinder assembly **201** into multiple chambers, or piston **202** is absent entirely, and cylinder assembly **201** is a "liquid piston." The cylinder assembly **201** may be divided into, e.g., two pneumatic chambers or one pneumatic chamber and one hydraulic chamber. The piston **202** is connected to a rod **204**, which may contain a center-drilled fluid passageway with fluid outlet **212** extending from the piston **202**. The rod **204** is also attached to, e.g., a mechanical load (e.g., a crankshaft or a hydraulic system) that is not depicted. The cylinder assembly **201** is in liquid communication with a heat-transfer subsystem **224** that includes or consists essentially of a circulation pump **214** and a spray mechanism **210** to enable substantially isothermal compression/expansion of gas. Heat-transfer fluid circulated by pump **214** may be passed through a heat exchanger **203** (e.g., tube-in-shell- or parallel-plate-type heat exchanger). Spray mechanism **210** may include or consist essentially of one or more spray heads (e.g., disposed at one end of cylinder assembly **201**) and/or spray rods (e.g., extending along at least a portion of the central axis of cylinder assembly **201**). In other embodiments, the spray mechanism **210** is omitted and a foam, rather than a spray of droplets, is created to facilitate heat exchange between liquid and gas during compression and expansion of gas within the cylinder assembly **201**, as described in the '128 application. Foam may be generated by foaming gas with heat-exchange liquid in a mechanism (not shown, described in more detail below) external to the cylinder assembly **201** and then injecting the resulting foam into the cylinder assembly **201**. Alternatively or additionally, foam may be generated inside the cylinder assembly **201** by the injection of heat-exchange liquid into cylinder assembly **201** through a foam-generating mechanism (e.g., spray head, rotating blade, one or more nozzles), partly or entirely filling the pneumatic chamber of cylinder assembly **201**. In some embodiments, droplets and foams may be introduced into cylinder assembly **201** simultaneously and/or sequentially. Various embodiments may feature mechanisms (not shown in FIG. **2**) for controlling the characteristics of foam (e.g., bubble size) and for breaking down, separating, and/or regenerating foam.

[0197] System **200** further includes a first control valve **220** (**108** in FIG. **1**) in communication with a storage reservoir **222** and cylinder assembly **201**, and a second control valve **221** (**118** in FIG. **1**) in communication with a vent **223** and cylinder assembly **201**. A control system **226** (**122** in FIG. **1**) may control operation of, e.g., valves **222** and **221** based on various system inputs (e.g., pressure, temperature, piston position, and/or fluid state) from cylinder assembly **201** and/or storage reservoir **222**. Heat-transfer fluid (liquid or circulated by pump **214** enters through pipe **213**. Pipe **213** may be (a) connected to a low-pressure fluid source (e.g., fluid reservoir (not shown) at the pressure to which vent **223** is connected or thermal well **242**); (b) connected to a high-pressure source (e.g., fluid reservoir (not shown) at the pressure of reservoir **222**); (c) selectively connected (using valve arrangement not shown) to low pressure during a compression process and to high pressure during an expansion process; (d) connected to changing-pressure fluid **208** in the cylinder **201** via connection **212**; or (e) some combination of these options.

[0198] In an initial state, the cylinder assembly **201** may contain a gas **206** (e.g., air introduced to the cylinder assembly **201** via valve **221** and vent **223**) and a heat-transfer fluid **208** (which may include or consist essentially of, e.g., water or another suitable liquid). When the gas **206** enters the cylinder assembly **201**, piston **202** is operated to compress the gas **206** to an elevated pressure (e.g., approximately 3,000 psi). Heat-transfer fluid (not necessarily the identical body of heat-transfer fluid **208**) flows from pipe **213** to the pump **214**. The pump **214** may raise the pressure of the heat-exchange fluid to a pressure (e.g., up to approximately 3,015 psig) somewhat higher than the pressure within the cylinder assembly **201**, as described in the '409 application. Alternatively or in conjunction, embodiments of the invention add heat (i.e., thermal energy) to, or remove heat from, the high-pressure gas in the cylinder assembly **201** by passing only relatively low-pressure fluids through a heat exchanger or fluid reservoir, as detailed in U.S. patent application Ser. No. 13/211,440, filed Aug. 17, 2011 (the '440 application), the entire disclosure of which is incorporated by reference herein.

[0199] Heat-transfer fluid is then sent through a pipe **216**, where it may be passed through a heat exchanger **203** (where its temperature is altered) and then through a pipe **218** to the spray mechanism **210**. The heat-transfer fluid thus circulated may include or consist essentially of liquid or foam. Spray mechanism **210** may be disposed within the cylinder assembly **201**, as shown; located in the storage reservoir **222** or vent **223**; or located in piping or manifolding around the cylinder assembly, such as pipe **218** or the pipes connecting the cylinder assembly to storage reservoir **222** or vent **223**. The spray mechanism **210** may be operated in the vent **223** or connecting pipes during compression, and a separate spray mechanism may be operated in the storage reservoir **222** or connecting pipes during expansion. Heat-transfer spray **211** from spray mechanism **210** (and/or any additional spray mechanisms), and/or foam from mechanisms internal or external to the cylinder assembly **101**, enable substantially isothermal compression of gas **206** within cylinder assembly **201**.

[0200] In some embodiments, the heat exchanger **203** is configured to condition heat-transfer fluid at low pressure (e.g., a pressure lower than the maximum pressure of a compression or expansion stroke in cylinder assembly **201**), and heat-transfer fluid is thermally conditioned between strokes or only during portions of strokes, as detailed in the '440 application. Embodiments of the invention are configured for circulation of heat-transfer fluid without the use of hoses that flex during operation through the use of, e.g., tubes or straws configured for non-flexure and/or pumps (e.g., submersible bore pumps, axial flow pumps, or other in-line style pumps) internal to the cylinder assembly (e.g., at least partially disposed within the piston rod thereof), as described in U.S. patent application Ser. No. 13/234,239, filed Sep. 16, 2011 (the '239 application), the entire disclosure of which is incorporated by reference herein.

[0201] At or near the end of the compression stroke, control system **226** opens valve **220** to admit the compressed gas **206** to the storage reservoir **222**. Operation of valves **220** and **221** may be controlled by various inputs to control system **226**, such as piston position in cylinder assembly **201**, pressure in storage reservoir **222**, pressure in cylinder assembly **201**, and/or temperature in cylinder assembly **201**.

[0202] As mentioned above, the control system **226** may enforce substantially isothermal operation, i.e., expansion and/or compression of gas in cylinder assembly **201**, via control over, e.g., the introduction of gas into and the exhausting of gas out of cylinder assembly **201**, the rates of compression and/or expansion, and/or the operation of the heat-exchange subsystem in response to sensed conditions. For example, control system **226** may be responsive to one or more sensors disposed in or on cylinder assembly **201** for measuring the temperature of the gas and/or the heat-exchange fluid within cylinder assembly **201**, responding to deviations in temperature by issuing control signals that operate one or more of the system components noted above to compensate, in real time, for the sensed temperature deviations. For example, in response to a temperature increase within cylinder assembly **201**, control system **226** may issue commands to increase the flow rate of spray **211** of heat-exchange fluid **208**.

[0203] Furthermore, embodiments of the invention may be applied to systems in which cylinder assembly **201** (or a chamber thereof) is in fluid communication with a pneumatic chamber of a second cylinder (e.g., as shown in FIG. **4**). That second cylinder, in turn, may communicate similarly with a third cylinder, and so forth. Any number of cylinders may be linked in this way. These cylinders may be connected in parallel or in a series configuration, where the compression and expansion is done in multiple stages.

[0204] The fluid circuit of heat exchanger **203** may be filled with water, a coolant mixture, an aqueous foam, or any other acceptable heat-exchange medium. In alternative embodiments, a gas, such as air or refrigerant, is used as the heat-exchange medium. In general, the fluid is routed by conduits to a large reservoir of such fluid in a closed or open loop. One example of an open loop is a well or body of water from which ambient water is drawn and the exhaust water is delivered to a different location, for example, downstream in a river. In a closed-loop embodiment, a cooling tower may cycle the water through the air for return to the heat exchanger. Likewise, water may pass through a submerged or buried coil of continuous piping where a counter heat-exchange occurs to return the fluid flow to ambient temperature before it returns to the heat exchanger for another cycle.

[0205] In various embodiments, the heat-exchange fluid is conditioned (i.e., pre-heated and/or pre-chilled) or used for heating or cooling needs by connecting the fluid inlet **238** and fluid outlet **240** of the external heat-exchange side of the heat exchanger **203** to an installation such as a heat-engine power plant, an industrial process with waste heat, a heat pump, and/or a building needing space heating or cooling, as described in the '731 patent. Alternatively, the external heat-exchange side of the heat exchanger **203** may be connected to a thermal well **242** as depicted in FIG. **2**. The thermal well **242** may include or consist essentially of a large water reservoir that acts as a constant-temperature thermal fluid source for use with the system. Alternatively, the water reservoir may be thermally linked to waste heat from an industrial process or the like, as described above, via another heat exchanger contained within the installation. This allows the heat-exchange fluid to acquire or expel heat from/to the linked process, depending on configuration, for later use as a heating/cooling medium in the energy storage/conversion system. Alternatively, the thermal well **242** may include two or more bodies of energy-storage medium, e.g., a hot-water thermal well and a cold-water thermal well, that are typically maintained in contrasting energy states in order to increase the exergy of system **200** compared with a system in which thermal well **242** includes a single body of energy-storage medium. Storage media other than water may be utilized in the thermal well **242**; temperature changes, phase changes, or both may be employed by storage media of thermal well **242** to store and release energy. Thermal or fluid links (not shown) to the atmosphere, ground, and/or other components of the environment may also be included in system **200**, allowing mass, thermal energy, or both to be added to or removed from the thermal well **242**. Moreover, as depicted in FIG. **2**, the heat-transfer subsystem **224** does not interchange fluid directly with the thermal well **242**, but in other embodiments, fluid is passed directly between the heat-transfer subsystem **224** and the thermal well **242** with no heat exchanger maintaining separation between fluids.

[0206] FIG. **3** is a schematic of the major components of an illustrative system **300** that employs a pneumatic cylinder **302** to efficiently convert (i.e., store) mechanical energy into the potential energy of compressed gas and, in another mode of operation, efficiently convert (i.e., recover) the potential energy of compressed gas into mechanical work. The pneumatic cylinder **302** may contain a slidably disposed piston **304** that divides the interior of the cylinder **302** into a distal chamber **306** and a proximal chamber **308**. A port or ports (not shown) with associated pipes **312** and a bidirectional valve **316** enables gas from a high-pressure storage reservoir **320** to be admitted to chamber **306** as desired. A port or ports (not shown) with associated pipes **322** and a bidirectional valve **324** enables gas from the chamber **306** to be exhausted through a vent **326** to the ambient atmosphere as desired. In alternate embodiments, vent **326** is replaced by additional lower-pressure pneumatic cylinders (or pneumatic chambers of cylinders). A port or ports (not shown) enables the interior of the chamber **308** to communicate freely at all times with the ambient atmosphere. In alternate embodiments, cylinder **302** is double-acting and chamber **308** is, like chamber **306**, equipped to admit and exhaust fluids in various states of operation. The distal end of a rod **330** is coupled to the piston **304**. The rod **330** may be connected to a crankshaft, hydraulic cylinder, or other mechanisms for converting linear mechanical motion to useful work as described in the '678 and '842 patents.

[0207] In the energy recovery or expansion mode of operation, storage reservoir **320** is filled with high-pressure air (or other gas) **332** and a quantity of heat-transfer fluid **334**. The heat-transfer fluid **334** may be an aqueous foam or a liquid that tends to foam when sprayed or otherwise acted upon. The liquid component of the aqueous foam, or the liquid that tends to foam, may include or consist essentially of water with 2% to 5% of certain additives; these additives may also provide functions of anti-corrosion, anti-wear (lubricity), anti-biogrowth (biocide), freezing-point modification (anti-freeze), and/or surface-tension modification. Additives may include a micro-emulsion of a lubricating fluid such as mineral oil, a solution of agents such as glycols (e.g. propylene glycol), or soluble synthetics (e.g. ethanolamines). Such additives tend to reduce liquid surface tension and lead to substantial foaming when sprayed. Commercially available fluids may be used at an approximately 5% solution in water, such as Mecagreen 127 (available from the Condat Corporation of Michigan), which consists in part of a micro-emulsion of mineral oil, and Quintolubric 807-WP (available from the Quaker Chemical Corporation of Pennsylvania), which consists in part of a soluble ethanolamine. Other additives may be used at higher concentrations (such as at a 50% solution in water), including Cryo-tek 100/A1 (available from the Hercules Chemical Company of New Jersey), which consists in part of a propylene glycol. These fluids may be further modified to enhance foaming while being sprayed and to speed defoaming when in a reservoir.

[0208] The heat-transfer fluid **334** may be circulated within the storage reservoir **320** via high-inlet-pressure, low-power-consumption pump **336** (such as described in the '731 patent). In various embodiments, the fluid **334** may be removed from the bottom of the storage reservoir **320** via piping **338**, circulated via pump **336** through a heat exchanger **340**, and introduced (e.g., sprayed) back into the top of storage reservoir **320** via piping **342** and spray head **344** (or other suitable mechanism). Any changes in pressure within reservoir **320** due to removal or addition of gas (e.g., via pipe **312**) generally tend to result in changes in temperature of the gas **332** within reservoir **320**. By spraying and/or foaming the fluid **334** throughout the storage reservoir gas **332**, heat may be added to or removed from the gas **332** via heat exchange with the heat-transfer fluid **334**. By circulating the heat-transfer fluid **334** through heat exchanger **340**, the temperature of the fluid **334** and gas **332** may be kept substantially constant (i.e., isothermal). Counterflow heat-exchange fluid **346** at near-ambient pressure may be circulated from a near-ambient-temperature thermal well (not shown) or source (e.g., waste heat source) or sink (e.g., cold water source) of thermal energy, as described in more detail below.

[0209] In various embodiments of the invention, reservoir **320** contains an aqueous foam, either unseparated or partially separated into its gaseous and liquid components. In such embodiments, pump **336** may circulate either the foam itself, or the separated liquid component of the foam, or both, and recirculation of fluid into reservoir **320** may include regeneration of foam by apparatus not shown in FIG. **3**.

[0210] In the energy recovery or expansion mode of operation, a quantity of gas may be introduced via valve **316** and pipe **312** into the upper chamber **306** of cylinder **302** when piston **304** is near or at the top of its stroke (i.e., "top dead center" of cylinder **302**). The piston **304** and its rod **330** will then be moving downward (the cylinder **302** may be oriented arbitrarily but is shown vertically oriented in this illustrative embodiment). Heat-exchange fluid **334** may be introduced into chamber **306** concurrently via optional pump **350** (alternatively, a pressure drop may be introduced in line **312** such that pump **350** is not needed) through pipe **352** and directional valve **354**. This heat-exchange fluid **334** may be sprayed into chamber **306** via one or more spray nozzles **356** in such a manner as to generate foam **360**. (In some embodiments, foam **360** is introduced directly into chamber **306** in foam form.) The foam **360** may entirely fill the entire chamber **306**, but is shown in FIG. **3**, for illustrative purposes only, as only partially filling chamber **306**. Herein, the term "foam" denotes either (a) foam only or (b) any of a variety of mixtures of foam and heat-exchange liquid in other, non-foaming states (e.g., droplets). Moreover, some non-foamed liquid (not shown) may accumulate at the bottom of chamber **306**; any such liquid is generally included in references herein to the foam **360** within chamber **306**.

[0211] System **300** is instrumented with pressure, piston position, and/or temperature sensors (not shown) and controlled via control system **362**. At a predetermined position of piston **304**, an amount of gas **332** and heat-transfer fluid **334** have been admitted into chamber **306** and valve **316** and valve **354** are closed. (Valves **316** and **354** may close at the same time or at different times, as each has a control value based on quantity of fluid desired.) The gas in chamber **306** then undergoes free expansion, continuing to drive piston **304** downward. During this expansion, in the absence of foam **360**, the gas would tend to decrease substantially in temperature. With foam **360** largely or entirely filling the chamber, the temperature of the gas in chamber **306** and the temperature of the heat-transfer fluid **360** tend to approximate to each other via heat exchange. The heat capacity of the liquid component of the foam **360** (e.g., water with one or more additives) may be much higher than that of the gas (e.g., air) such that the temperature of the gas and liquid do not change substantially (i.e., are substantially isothermal) even over a many-times gas expansion (e.g., from 250 psig to near atmospheric pressure, or in other embodiments from 3,000 psig to 250 psig).

[0212] When the piston **304** reaches the end of its stroke (bottom dead center), the gas within chamber **306** will have expanded to a predetermined lower pressure (e.g., near atmospheric). Valve **324** will then be opened, allowing gas from chamber **306** to be vented, whether to atmosphere through pipe **322** and vent **326** (as illustrated here) or, in other embodiments, to a next stage in the expansion process (e.g., chamber in a separate cylinder), via pipe **322**. Valve **324** remains open as the piston undergoes an upward (i.e., return) stroke, emptying chamber **306**. Part or substantially all of foam **360** is also forced out of chamber **306** via pipe **322**. A separator (not shown) or other means such as gravity separation is used to recover heat-transfer fluid, preferably de-foamed (i.e., as a simple liquid with or without additives), and to direct it into a storage reservoir **364** via pipe **366**.

[0213] When piston **304** reaches top of stroke again, the process repeats with gas **332** and heat-transfer fluid **334** admitted from vessel **320** via valves **316** and **354**. If additional heat-transfer fluid is needed in reservoir **320**, it may be pumped back into reservoir **320** from reservoir **364** via piping **367** and optional pump/motor **368**. In one mode of operation, pump **368** may be used to continuously refill reservoir **320** such that the pressure in reservoir **320** is held substantially constant. That is, as gas is removed from reservoir **320**, heat-transfer fluid **334** is added to maintain constant pressure in reservoir **320**. In other embodiments, pump **368** is not used or is used intermittently, the pressure in reservoir **320** continues to decrease during an energy-recovery process (i.e., involving removal of gas from reservoir **320**), and the control system **362** changes the timing of valves **316** and **354** accordingly so as to reach approximately the same ending pressure when the piston **304** reaches the end of its stroke. An energy-recovery process may continue until the storage reservoir **320** is nearly empty of pressurized gas **332**, at which time an energy-storage process may be used to recharge the storage reservoir **320** with pressurized gas **332**. In other embodiments, the energy-recovery and energy-storage processes are alternated based on operator requirements.

[0214] In either the energy-storage or energy-compression mode of operation, storage reservoir **320** is typically at least partially depleted of high-pressure gas **332**, as storage reservoir **320** also typically contains a quantity of heat-transfer fluid **334**. Reservoir **364** is at low pressure (e.g., atmospheric or some other low pressure that serves as the intake pressure for the compression phase of cylinder **302**) and contains a quantity of heat-transfer fluid **370**.

[0215] The heat-transfer fluid **370** may be circulated within the reservoir **364** via low-power-consumption pump **372**. In various embodiments, the fluid **370** may be removed from the bottom of the reservoir **364** via piping **367**, circulated via pump **372** through a heat exchanger **374**, and introduced (e.g., sprayed) back into the top of reservoir **364** via piping **376** and spray head **378** (or other suitable mechanism). By spraying the fluid **370** throughout the reservoir gas **380**, heat may be added or removed from the gas via the heat-transfer fluid **370**. By circulating the heat-transfer fluid **370** through heat exchanger **374**, the temperature of the fluid **370** and gas **380** may be kept near constant (i.e., isothermal). Counterflow heat-exchange fluid **382** at near-ambient pressure may be circulated from a near-ambient-temperature thermal well (not shown) or source (e.g., waste heat source) or sink (e.g., cold water source) of thermal energy. In one embodiment, counterflow heat-exchange fluid **382** is at high temperature to increase energy recovery during expansion and/or counterflow heat-exchange fluid **382** is at low temperature to decrease energy usage during compression.

[0216] In the energy-storage or compression mode of operation, a quantity of low-pressure gas is introduced via valve **324** and pipe **322** into the upper chamber **306** of cylinder **302** starting when piston **304** is near top dead center of cylinder **302**. The low-pressure gas may be from the ambient atmosphere (e.g., may be admitted through vent **326** as illustrated herein) or may be from a source of pressurized gas such as a previous compression stage. During the intake stroke, the piston **304** and its rod **330** will move downward, drawing in gas. Heat-exchange fluid **370** may be introduced into chamber **306** concurrently via optional pump **384** (alternatively, a pressure drop may be introduced in line **386** such that pump **384** is not needed) through pipe **386** and directional valve **388**. This heat exchange fluid **370** may be introduced (e.g., sprayed) into chamber **306** via one or more spray nozzles **390** in such a manner as to generate foam **360**. This foam **360** may fill the chamber **306** partially or entirely by the end of the intake stroke; for illustrative purposes only, foam **360** is shown in FIG. **3** as only partially filling chamber **306**. At the end of the intake stroke, piston **304** reaches the end-of-stroke position (bottom dead center) and chamber **306** is filled with foam **360** generated from air at a low pressure (e.g., atmospheric) and heat-exchange liquid.

[0217] At the end of the stroke, with piston **304** at the end-of-stroke position, valve **324** is closed. Valve **388** is also closed, not necessarily at the same time as valve **324**, but after a predetermined amount of heat-transfer fluid **370** has been admitted, creating foam **360**. The amount of heat-transfer fluid **370** may be based upon the volume of air to be compressed, the ratio of compression, and/or the heat capacity of the heat-transfer fluid. Next, piston **304** and rod **330** are driven upwards via mechanical means (e.g., hydraulic fluid, hydraulic cylinder, mechanical crankshaft) to compress the gas within chamber **306**.

[0218] During this compression, in the absence of foam **360**, the gas in chamber **306** would tend to increase substantially in temperature. With foam **360** at least partially filling the chamber, the temperature of the gas in chamber **306** and the temperature of the liquid component of foam **360** will tend to equilibrate via heat exchange. The heat capacity of the fluid component of foam **360** (e.g., water with one or more additives) may be much higher than that of the gas (e.g., air) such that the temperature of the gas and fluid do not change substantially and are near-isothermal even over a many-times gas compression (e.g., from near atmospheric pressure to 250 psig, or in other embodiments from 250 psig to 3,000 psig).

[0219] The gas in chamber **306** (which includes, or consists essentially of, the gaseous component of foam **360**) is compressed to a suitable pressure, e.g., a pressure approximately equal to the pressure within storage reservoir **320**, at which time valve **316** is opened. The foam **360**, including both its gaseous and liquid components, is then transferred into storage reservoir **320** through valve **316** and pipe **312** by continued upward movement of piston **304** and rod **330**.

[0220] When piston **304** reaches top of stroke again, the process repeats, with low-pressure gas and heat-transfer fluid **370** admitted from vent **326** and reservoir **364** via valves **324** and **388**. If additional heat-transfer fluid is needed in reservoir **364**, it may be returned to reservoir **364** from reservoir **320** via piping **367** and optional pump/motor **368**. Power recovered from motor **368** may be used to help drive the mechanical mechanism for driving piston **304** and rod **330** or may be converted to electrical power via an electric motor/generator (not shown). In one mode of operation, motor **368** may be run continuously, while reservoir **320** is being filled with gas, in such a manner that the pressure in reservoir **320** is held substantially constant. That is, as gas is added to reservoir **320**, heat-transfer fluid **334** is removed from reservoir **320** to maintain substantially constant pressure within reservoir **320**. In other embodiments, motor **368** is not used or is used intermittently; the pressure in reservoir **320** continues to increase during an energy-storage process and the control system **362** changes the timing of valves **316** and **388** accordingly so that the desired ending pressure (e.g., atmospheric) is attained within chamber **306** when the piston **304** reaches bottom of stroke. An energy-storage process may continue until the storage reservoir **320** is full of pressurized gas **332** at the maximum storage pressure (e.g., 3,000 psig), after which time the system is ready to perform an energy-recovery process. In various embodiments, the system may commence an energy-recovery process when the storage reservoir **320** is only partly full of pressurized gas **332**, whether at the maximum storage pressure or at some storage pressure intermediate between atmospheric pressure and the maximum storage pressure. In other embodiments, the energy-recovery and energy-storage processes are alternated based on operator requirements.

[0221] FIG. **4** depicts an illustrative system **400** that features at least two cylinder assemblies **402**, **406** (i.e., an embodiment of assembly **101** in FIG. **1**; e.g., cylinder assembly **201** in FIG. **2**) and a heat-transfer subsystem **404**, **408** (e.g., subsystem **224** in FIG. **2**) associated with each cylinder assembly **402**, **406**. Additionally, the system includes a thermal well **410** (e.g., thermal well **242** in FIG. **2**) which may be associated with either or both of the heat-transfer subsystems **404**, **408** as indicated by the dashed lines.

[0222] Assembly **402** is in selective fluid communication with a storage reservoir **412** (e.g., **112** in FIG. **1**, **222** in FIG. **2**) capable of holding fluid at relatively high pressure (e.g., approximately 3,000 psig). Assembly **406** is in selective fluid communication with assembly **402** and/or with optional additional cylinder assemblies between assemblies **402** and **406** as indicated by ellipsis marks **422**. Assembly **406** is in selective fluid communication with an atmospheric vent **420** (e.g., **120** in FIG. **1**, **223** in FIG. **2**).

[0223] System **400** may compress air at atmospheric pressure (admitted to system **400** through the vent **420**) stagewise through assemblies **406** and **402** to high pressure for storage in reservoir **412**. System **400** may also expand air from high pressure in reservoir **412** stagewise through assemblies **402** and **406** to a low pressure (e.g., approximately 5 psig) for venting to the atmosphere through vent **420**.

[0224] As described in U.S. Pat. No. 8,191,362, filed Apr. 6, 2011 (the '362 patent), the entire disclosure of which is incorporated by reference herein, in a group of N cylinder assemblies used for expansion or compression of gas between a high pressure (e.g., approximately 3,000 psig) and a low pressure (e.g., approximately 5 psig), the system will contain gas at N−1 pressures intermediate between the high-pressure extreme and the low pressure. Herein each such intermediate pressure is termed a "mid-pressure." In illustrative system **400**, N=2 and N−1=1, so there is one mid-pressure (e.g., approximately 250 psig during expansion) in the system **400**. In various states of operation of the system, mid-pressures may occur in any of the chambers of a series-connected cylinder group (e.g., the cylinders of assemblies **402** and **406**) and within any valves, piping, and other devices in fluid communication with those chambers. In illustrative system **400**, the mid-pressure, herein denoted "mid-pressure P**1**," occurs primarily in valves, piping, and other devices intermediate between assemblies **402** and **406**.

[0225] Assembly **402** is a high-pressure assembly: i.e., assembly **402** may admit gas at high pressure from reservoir **412** to expand the gas to mid-pressure P**1** for transfer to assembly **402**, and/or may admit gas at mid-pressure P**1** from assembly **406** to compress the gas to high pressure for transfer to reservoir **412**. Assembly **406** is a low-pressure assembly: i.e., assembly **406** may admit gas at mid-pressure P**1** from assembly **402** to expand the gas to low pressure for transfer to the vent **420**, and/or may admit gas at low pressure from vent **420** to compress the gas to mid-pressure P**1** for transfer to assembly **402**.

[0226] In system **400**, extended cylinder assembly **402** communicates with extended cylinder assembly **406** via a mid-pressure assembly **414**. Herein, a "mid-pressure assembly" includes or consists essentially of a reservoir of gas that is placed in fluid communication with the valves, piping, chambers, and other components through or into which gas passes. The gas in the reservoir is at approximately at the mid-pressure which the particular mid-pressure assembly is intended to provide. The reservoir is large enough so that a volume of mid-pressure gas approximately equal to that within the valves, piping, chambers, and other components with which the reservoir is in fluid communication may enter or leave the reservoir without substantially changing its pressure. Additionally, the mid-pressure assembly may provide pulsation damping, additional heat-transfer capability, fluid separation, and/or house one or more heat-transfer sub-systems such as part or all of sub-systems **404** and/or **408**. As described in the '362 patent, a mid-pressure assembly may substantially reduce the amount of dead space in various components of a system employing pneumatic cylinder assemblies, e.g., system **400** in FIG. **4**. Reduction of dead space tends to increase overall system efficiency.

[0227] Alternatively or in conjunction, pipes and valves (not shown in FIG. **4**) bypassing mid-pressure assembly **414** may enable fluid to pass directly between assembly **402** and assembly **406**. Valves **416**, **418**, **424**, and **426** control the passage of fluids between the assemblies **402**, **406**, **412**, and **414**.

[0228] A control system **428** (e.g., **122** in FIG. **1**, **226** in FIG. **2**, **362** in FIG. **3**) may control operation of, e.g., all valves of system **400** based on various system inputs (e.g., pressure, temperature, piston position, and/or fluid state) from assemblies **402** and **406**, mid-pressure assembly **414**, storage reservoir **412**, thermal well **410**, heat transfer subsystems **404**, **408**, and/or the environment surrounding system **420**.

[0229] It will be clear to persons reasonably familiar with the art of pneumatic machines that a system similar to system **400** but differing by the incorporation of one, two or more mid-pressure extended cylinder assemblies may be devised without additional undue experimentation. It will also be clear that all remarks herein pertaining to system **400** may be applied to such an N-cylinder system without substantial revision, as indicated by elliptical marks **422**. Such N-cylinder systems, though not discussed further herein, are contemplated and within the scope of the invention. As shown and described in the '678 patent, N appropriately sized cylinders, where N≧2, may reduce an original (single-cylinder) operating fluid pressure range R to $R^{1/N}$ and correspondingly reduce the range of force acting on each cylinder in the

N-cylinder system as compared to the range of force acting in a single-cylinder system. This and other advantages, as set forth in the '678 patent, may be realized in N-cylinder systems. Additionally, multiple identical cylinders may be added in parallel and attached to a common or separate drive mechanism (not shown) with the cylinder assemblies **402**, **406** as indicated by ellipsis marks **432**, **436**, enabling higher power and air-flow rates.

[0230] FIG. **5** is a schematic diagram showing components of a system **500** for achieving approximately isothermal compression and expansion of a gas for energy storage and recovery using a pneumatic cylinder **502** (shown in partial cross-section) according to embodiments of the invention. The cylinder **502** typically contains a slidably disposed piston **504** that divides the cylinder **502** into two chambers **506**, **508**. A reservoir **510** contains gas at high pressure (e.g., 3,000 psi); the reservoir **510** may also contain a quantity of heat-exchange liquid **512**. The heat-exchange liquid **512** may contain an additive that increases the liquid's tendency to foam (e.g., by lowering the surface tension of the liquid **512**). Additives may include surfactants (e.g., sulfonates), a micro-emulsion of a lubricating fluid such as mineral oil, a solution of agents such as glycols (e.g., propylene glycol), or soluble synthetics (e.g., ethanolamines). Foaming agents such as sulfonates (e.g., linear alkyl benzene sulfonate such as Bio-Soft D-40 available from Stepan Company of Illinois) may be added, or commercially available foaming concentrates such as fire-fighting foam concentrates (e.g., fluorosurfactant products such as those available from ChemGuard of Texas) may be used. Such additives tend to reduce liquid surface tension of water and lead to substantial foaming when sprayed. Commercially available fluids may be used at an approximately 5% solution in water, such as Mecagreen 127 (available from the Condat Corporation of Michigan), which consists in part of a micro-emulsion of mineral oil, and Quintolubric 807-WP (available from the Quaker Chemical Corporation of Pennsylvania), which consists in part of a soluble ethanolamine. Other additives may be used at higher concentrations (such as at a 50% solution in water), including Cryo-tek 100/A1 (available from the Hercules Chemical Company of New Jersey), which consists in part of a propylene glycol. These fluids may be further modified to enhance foaming while being sprayed and to speed defoaming when in a reservoir.

[0231] A pump **514** and piping **516** may convey the heat-exchange liquid to a device herein termed a "mixing chamber" (**518**). Gas from the reservoir **510** may also be conveyed (via piping **520**) to the mixing chamber **518**. Within the mixing chamber **518**, a foam-generating mechanism **522** combines the gas from the reservoir **510** and the liquid conveyed by piping **516** to create foam **524** of a certain grade (i.e., bubble size variance, average bubble size, void fraction), herein termed Foam A, inside the mixing chamber **518**.

[0232] The mixing chamber **518** may contain a screen **526** or other mechanism (e.g., source of ultrasound) to vary or homogenize foam structure. Screen **526** may be located, e.g., at or near the exit of mixing chamber **518**. Foam that has passed through the screen **526** may have a different bubble size and other characteristics from Foam A and is herein termed Foam B (**528**). In other embodiments, the screen **526** is omitted, so that Foam A is transferred without deliberate alteration to chamber **506**.

[0233] The exit of the mixing chamber **518** is connected by piping **530** to a port in the cylinder **502** that is gated by a valve **532** (e.g., a poppet-style valve) that permits fluid from piping **530** to enter the upper chamber (air chamber) **506** of the cylinder **502**. Valves (not shown) may control the flow of gas from the reservoir **510** through piping **520** to the mixing chamber **518**, and from the mixing chamber **518** through piping **528** to the upper chamber **506** of the cylinder **502**. Another valve **534** (e.g., a poppet-style valve) permits the upper chamber **506** to communicate with other components of the system **500**, e.g., an additional separator device (not shown), the upper chamber of another cylinder (not shown), or a vent to the ambient atmosphere (not shown).

[0234] The volume of reservoir **510** may be large (e.g., at least approximately four times larger) relative to the volume of the mixing chamber **518** and cylinder **502**. Foam A and Foam B are preferably statically stable foams over a portion or all of the time-scale of typical cyclic operation of system **500**: e.g., for a 120 RPM system (i.e., 0.5 seconds per revolution), the foam may remain substantially unchanged (e.g., less than 10% drainage) after 5.5 seconds or a time approximately five times greater than the revolution time.

[0235] In an initial state of operation of a procedure whereby gas stored in the reservoir **510** is expanded to release energy, the valve **532** is open, the valve **534** is closed, and the piston **504** is near top dead center of cylinder **502** (i.e., toward the top of the cylinder **502**). Gas from the reservoir **510** is allowed to flow through piping **520** to the mixing chamber **518** while liquid from the reservoir **510** is pumped by pump **514** to the mixing chamber **518**. The gas and liquid thus conveyed to the mixing chamber **518** are combined by the foam-generating mechanism **522** to form Foam A (**524**), which partly or substantially fills the main chamber of the mixing chamber **518**. Exiting the mixing chamber **518**, Foam A passes through the screen **526**, being altered thereby to Foam B. Foam B, which is at approximately the same pressure as the gas stored in reservoir **510**, passes through valve **532** into chamber **506**. In chamber **506**, Foam B exerts a force on the piston **504** that may be communicated to a mechanism (e.g., an electric generator, not shown) external to the cylinder **502** by a rod **536** that is connected to piston **504** and that passes slideably through the lower end cap of the cylinder **502**.

[0236] The gas component of the foam in chamber **506** expands as the piston **504** and rod **536** move downward. At some point in the downward motion of piston **504**, the flow of gas from reservoir **510** into the mixing chamber **518** and thence (as the gas component of Foam B) into chamber **506** may be ended by appropriate operation of valves (not shown). As the gas component of the foam in chamber **506** expands, it will tend, unless heat is transferred to it, to decrease in temperature according to the Ideal Gas Law; however, if the liquid component of the foam in chamber **506** is at a higher temperature than the gas component of the foam in chamber **506**, heat will tend to be transferred from the liquid component to the gas component. Therefore, the temperature of the gas component of the foam within chamber **506** will tend to remain constant (approximately isothermal) as the gas component expands.

[0237] When the piston **504** approaches bottom dead center of cylinder **502** (i.e., has moved down to approximately its limit of motion), valve **532** may be closed and valve **534** may be opened, allowing the expanded gas in chamber **506** to pass from cylinder **502** to some other component of the system **500**, e.g., a vent or a chamber of another cylinder for further expansion.

[0238] In some embodiments, pump **514** is a variable-speed pump, i.e., may be operated so as to transfer liquid **512** at a slower or faster rate from the reservoir **510** to the foam-generating mechanism **522** and may be responsive to signals from the control system (not shown). If the rate at which liquid **512** is transferred by the pump **514** to the foam-mechanism **522** is increased relative to the rate at which gas is conveyed from reservoir **510** through piping **520** to the mechanism **522**, the void fraction of the foam produced by the mechanism **522** may be decreased. If the foam generated by the mechanism **522** (Foam A) has a relatively low void fraction, the foam conveyed to chamber **506** (Foam B) will generally also tend to have a relatively low void fraction. When the void fraction of a foam is lower, more of the foam consists of liquid, so more thermal energy may be exchanged between the gas component of the foam and the liquid component of the foam before the gas and liquid components come into thermal equilibrium with each other (i.e., cease to change in relative temperature). When gas at relatively high density (e.g., ambient temperature, high pressure) is being transferred from the reservoir **510** to chamber **506**, it may be advantageous to generate foam having a lower void fraction, enabling the liquid fraction of the foam to exchange a correspondingly larger quantity of thermal energy with the gas fraction of the foam.

[0239] All pumps shown in subsequent figures herein may also be variable-speed pumps and may be controlled based on signals from the control system. Signals from the control system may be based on system-performance (e.g., gas temperature and/or pressure, cycle time, etc.) measurements from one or more previous cycles of compression and/or expansion.

[0240] Embodiments of the invention increase the efficiency of a system **500** for the storage and retrieval of energy using compressed gas by enabling the surface area of a given quantity of heat-exchange liquid **512** to be greatly increased (with correspondingly accelerated heat transfer between liquid **512** and gas undergoing expansion or compression within cylinder **502**) with less investment of energy than would be required by alternative methods of increasing the surface of area of the liquid, e.g., the conversion of the liquid **512** to a spray.

[0241] In other embodiments, the reservoir **510** is a separator rather than a high-pressure storage reservoir as depicted in FIG. **5**. In such embodiments, piping, valves, and other components not shown in FIG. **5** are supplied that allow the separator to be placed in fluid communication with a high-pressure gas storage reservoir as well as with the mixing chamber **518**, as shown and described in the '128 application.

[0242] FIG. **6** is a schematic diagram showing components of a system **600** for achieving approximately isothermal compression and expansion of a gas for energy storage and recovery using a pneumatic cylinder **604** (shown in partial cross-section) according to embodiments of the invention. System **600** is similar to system **500** in FIG. **5**, except that system **600** includes a bypass pipe **638**. Moreover, two valves **640**, **642** are explicitly depicted in FIG. **6**. Bypass pipe **638** may be employed as follows: (1) when gas is being released from the storage reservoir **610**, mixed with heat-exchange liquid **612** in the mixing chamber **618**, and conveyed to chamber **606** of cylinder **604** to be expanded therein, valve **640** will be closed and valve **642** open; (2) when gas has been compressed in chamber **606** of cylinder **604** and is to be conveyed to the reservoir **610** for storage, valve **640** will be open and valve **642** closed. Less friction will tend to be encountered by fluids passing through valve **640** and bypass pipe **638** than by fluids passing through valve **642** and screen **626** and around the foam-generating mechanism **622**. In other embodiments, valve **642** is omitted, allowing fluid to be routed through the bypass pipe **638** by the higher resistance presented by the mixing chamber **618**, and valve **640** is a check valve preventing fluid flow when gas is being released in expansion mode. The direction of fluid flow from chamber **606** to the reservoir **610** via a lower-resistance pathway (i.e., the bypass pipe **638**) will tend to result in lower frictional losses during such flow and therefore higher efficiency for system **600**.

[0243] In other embodiments, the reservoir **610** is a separator rather than a high-pressure storage reservoir as depicted in FIG. **6**. In such embodiments, piping, valves, and other components not shown in FIG. **6** are supplied that allow the separator to be placed in fluid communication with a high-pressure gas storage reservoir as well as with the mixing chamber **618** and bypass pipe **638**.

[0244] FIG. **7** is a schematic diagram showing components of a system **700** for achieving approximately isothermal compression and expansion of a gas for energy storage and recovery using a pneumatic cylinder **702** (shown in partial cross-section) according to embodiments of the invention. System **700** is similar to system **500** in FIG. **5**, except that system **700** omits the mixing chamber **518** and instead generates foam inside the storage reservoir **710**. In system **700**, a pump **714** circulates heat-exchange liquid **712** to a foam-generating mechanism **722** (e.g., one or more spray nozzles) inside the reservoir **710**. The reservoir **710** may, by means of the pump **714** and mechanism **722**, be filled partly or entirely by foam of an initial or original character, Foam A (**724**). The reservoir **710** may be placed in fluid communication via pipe **720** with a valve-gated port **744** in cylinder **702**. Valves (not shown) may govern the flow of fluid through pipe **720**. An optional screen **726** (or other suitable mechanism such as an ultrasound source), shown in FIG. **7** inside pipe **720** but locatable anywhere in the path of fluid flow between reservoir **710** and chamber **706** of the cylinder **702**, serves to alter Foam A (**724**) to Foam B (**728**), regulating characteristics such as bubble-size variance and average bubble size.

[0245] In other embodiments, the reservoir **710** is a separator rather than a high-pressure storage reservoir as depicted in FIG. **7**. In such embodiments, piping, valves, and other components not shown in FIG. **7** will be supplied that allow the separator to be placed in fluid communication with a high-pressure gas storage reservoir as well as with the cylinder **702**. In other embodiments, a bypass pipe similar to that depicted in FIG. **6** is added to system **700** in order to allow fluid to pass from cylinder **702** to reservoir **710** without passing through the screen **726**.

[0246] FIG. **8** is a schematic diagram showing components of a system **800** for achieving approximately isothermal compression and expansion of a gas for energy storage and recovery using a pneumatic cylinder **802** (shown in partial cross-section) according to embodiments of the invention. System **800** is similar to system **500** in FIG. **5**, except that system **800** omits the mixing chamber **518** and instead generates foam inside the air chamber **806** of the cylinder **802**. In system **800**, a pump **814** circulates heat-exchange liquid **812** to a foam-generating mechanism **822** (e.g., one or more spray nozzles injecting into cylinder and/or onto a screen through which admitted air passes) either located within, or communicating with (e.g., through a port), chamber **806**. The chamber **806** may, by means of the pump **814** and mechanism **822** (and by means of gas supplied from reservoir **810** via pipe **820** through a port **844**), be filled partly or substantially entirely by foam. The reservoir **810** may be placed in fluid communication via pipe **820** with valve-gated port **844** in cylinder **802**. Valves (not shown) may govern the flow of fluid through pipe **820**.

[0247] FIG. **9** is a schematic drawing of an illustrative CAES system **900** employing pairs of high- and low-pressure cylinders in which air is compressed and expanded. Half of the cylinders are high-pressure cylinders (HPCs, indicated in FIG. **900** by block **902**) and half of the cylinders are low-pressure cylinders (LPCs, indicated in FIG. **900** by block **904**), resulting in a two-stage compression process. Block **902** represents some number N of high-pressure cylinders (not shown) and block **904** represents an equal number N of low-pressure cylinders (not shown). The HPCs and LPCs jointly drive a crankshaft that in turn drives an electric generator or, in some states of operation of system **900**, is driven by an electric motor. Systems employing principles of operation similar to those of **900** but including other subsystems, other mechanisms, other arrangements of parts, other numbers of stages (i.e., a single stage or more than one stage), and unequal numbers of high- and low-pressure cylinders, are also contemplated and within the scope of the invention.

[0248] Separating the LPCs from the HPCs is a mid-pressure vessel (MPV) **906** that buffers and decouples the HPCs **902** and LPCs **904** during either compression or expansion processes. This allows each cylinder assembly (i.e., each high- or low-pressure cylinder and the valves that control the entry or exit of gas from the cylinder) to operate independently from all the other cylinder assemblies within system **900**. Independent operation of cylinder assemblies allows, in turn, for optimization of the performance (e.g., optimization of valve timing) of each cylinder assembly. A system controller (not shown), e.g., a computerized controller, coordinates the operation of individual cylinders with each other and with the other pneumatic components and processes within system **900**.

[0249] In addition to the cylinders **902**, **904** and MPV **906**, system **900** also includes a spray reservoir **908** that holds a heat-transfer fluid (e.g., treated water) at low (e.g., atmospheric) pressure, a low-pressure spray chamber **910** that creates foam and/or spray at atmospheric pressure for intake into the LPCs for compression, and a high-pressure spray chamber **912** that creates foam and/or spray at storage pressure (i.e., the pressure at which gas and/or heat-transfer fluid is stored after compression and/or before expansion) for expansions in the HPCs. Finally, system **900** includes one or more storage reservoirs (not shown) that are connected to the HPCs **902** via the high-pressure spray chamber **912**. The storage reservoirs typically contain compressed air, e.g., air compressed by system **900** and stored for future expansion to drive electricity generation.

[0250] Each of the cylinder assemblies in the HPC group **902** and LPC group **904** typically includes a cylinder similar to cylinder **201** in FIG. **2**, a high-side valve similar to valve **220** in FIG. **2**, and a low-side valve assembly similar to valve **221** in FIG. **2**. Each high-side valve includes or consists essentially of one or more poppet elements that open out of the cylinder, connecting the expansion/compression chamber of the cylinder to a volume that is generally at higher pressure than the chamber. For a low-pressure cylinder, the high-side valve connects the cylinder's expansion/compression chamber to the MPV **906**; for a high-pressure cylinder, the high-side valve connects the cylinder's expansion/compression chamber to the high-pressure spray chamber **912**. Because these high-side valves open out of the cylinder rather than into the cylinder, they passively check open under an over-pressure condition in the cylinder, reducing the risk of a hydrolocking event with possible attendant damage to system components or interference with system operation.

[0251] Each low-side valve includes or consists essentially of one or more poppet elements that open into the cylinder, connecting the expansion/compression chamber of the cylinder to a volume that is generally at lower pressure than the chamber. For a low-pressure cylinder, the low-side valve connects the cylinder's expansion/compression chamber to the spray reservoir **908**; for a high-pressure cylinder, the low-side valve connects the cylinder's expansion/compression chamber to the MPV **906**. All of the valves, both low-side and high-side, may be hydraulically actuated. Other actuated valves such as variable cam-driven valves, electromagnetically actuated valves, mechanically actuated valves, and pneumatically actuated valves are also considered and may be utilized.

[0252] The system **900** may cyclically perform a normal compression process (or "compression cycle") or a normal expansion process (or "expansion cycle"). In a normal compression process, each low-pressure and high-pressure cylinder progresses through a series of four conditions or phases, each of which has an associated a valve configuration. The four phases are (1) compression stroke, (2) direct fill, (3) regeneration or expansion stroke, and (4) breathe, intake, or auxiliary stroke. The numbering of the phases is arbitrary in the sense that when the phases are performed in a repeating cycle, no one phase is "first" other than by convention. It is assumed in this description that all high-side and low-side valves in system **900** activate instantaneously and ideally; the implications of non-ideal valve actuation will be described subsequently. The four phases are described in detail below.

[0253] The compression stroke begins with the cylinder's piston at the bottom of its stroke range. The cylinder's expansion/compression chamber (herein also referred to simply as "the chamber") is filled with air at relatively low pressure (e.g., atmospheric). For example, the cylinder may have previously drawn in air through its low-side valve from a source on its low side (e.g., an HPC draws air in from the MPV **906**, or an LPC draws air in from the ambient intake/exhaust port). With the piston at bottom of stroke, the cylinder's low-side valve and high-side valve both close, if they were not already closed, and the piston begins to move upward, compressing the air within the chamber: i.e., the compression stroke begins. The compression stroke continues as the piston moves up from bottom of stroke with both valves closed, compressing the air inside. The compression phase nominally ends when the pressure in the cylinder is approximately equal to the pressure in the component (e.g., MPV **906** or HP spray chamber and thence to high-pressure storage reservoir) to which the cylinder is connected on its high side. At this point, the direct-fill stroke or phase begins.

[0254] The direct-fill stroke or phase occurs while the piston is still moving upward and involves pushing the compressed air within the chamber out of the cylinder and into the high-side component. Direct fill begins when the pressures in the cylinder and the high-side vessel are approximately equal and the high-side valve is actuated to open. The low-side valve remains closed. Once the high-side valve is open, the cylinder pushes compressed air from its chamber into the high-side component as the piston continues to travel toward top of stroke. Direct fill ends when the cylinder reaches top of stroke, whereupon the high-side valve is closed.

[0255] The regeneration stroke occurs with both valves closed as the cylinder piston moves downward, away from top of stroke. Each cylinder has some amount of clearance volume, which is the physical space within the cylinder—above the piston and below the valves and in all the connections and crevices—that is present when the piston is at top of stroke. Moreover, in a CAES system that utilizes a liquid/water mixture to effect heat transfer within the cylinder (e.g., system **900**), some fraction of the clearance volume will be occupied by liquid and some by air. That portion of the clearance volume occupied by air during a particular state of operation of the cylinder is the air dead volume (also herein termed simply the "dead volume") of the cylinder in that state of operation. This is the portion of air that was compressed during the compression stroke that was not then subsequently pushed into the high-side component. This compressed air contains energy (i.e., both thermal and elastic potential), and the regeneration stroke allows this energy to be recaptured. The regeneration stroke starts at top of stroke with both valves closed and continues as the piston moves downward, expanding the dead volume air. The regeneration stroke ends when the pressure in the cylinder has dropped to the low-side vessel pressure and the low-side valve is commanded open.

[0256] As the cylinder piston is moving downward, once the low-side valve is opened the intake stroke begins. The intake stroke continues, drawing in new air to be compressed on the next stroke, until the piston reaches bottom of stroke. At this point, the low-side valve is closed and the next compression stroke may begin.

[0257] Each of the four compression stages is separated from preceding and subsequent stages by a valve transition event, i.e., the opening or closing of one or more valves. In the descriptions of the stages above, the valve transition points were clearly defined as top of stroke, bottom of stroke, or pressure equalization, but this assumes that the valves of system **900** respond ideally and instantaneously. However, because of finite valve response time, each valve transition event is an opportunity for system optimization.

[0258] The first valve transition event mentioned above is the transition from intake stroke to compression stroke, which nominally occurs at bottom dead center (BDC; the condition where the piston is at its nethermost point of motion). With finite (nonzero) valve response time, the low-side valve will need to be commanded closed slightly before BDC and will likely be full seat closed slightly after BDC. Transitioning this valve too early means that less air is drawn in than could have been, resulting in less air to be compressed during the following compression stroke (reduced capacity). Transitioning the valve too late means that some of the air drawn in will be re-exhausted before the valve fully seats, also resulting in reduced capacity.

[0259] The second valve event is at the end of compression, transitioning into direct-fill, where the high-side valve is opened to end admission of air into the chamber and start pushing air into the high-side vessel. Nominally, the high-side valve opens when the pressure in the chamber is equal to the pressure in the high-side component. However, because of finite valve response time, if the valve is commanded open when the pressures are equal, then the pressure in the chamber will spike significantly as the flow from the chamber is limited and throttled through the high-side valve while the valve is transitioning open. The pressure spike may be avoided by commanding the high-side valve to open before pressures are equal in the chamber and the high-side component. However, if the high-side valve starts to open when the chamber pressure is still lower than that within the high-side component, then some amount of back-flow into the chamber will occur as fluid from the high-side component flows backward through the high-side valve into the chamber. (This backflow will be throttled, since the high-side valve is partially open.) If the high-side valve is opened too early, then the pressure within the chamber will jump quickly to the high-side component pressure, and the piston will need to perform additional work to push the air back out of the chamber into the high-side component. Thus, this valve transition entails a tradeoff between backflow and pressure-spike, both of which impact the pressure profile and the work that needs to be performed by the piston upon the air.

[0260] The valve transition closing the high-side valve at the end of the direct fill impacts system capacity. If the high-side valve is closed too early, then less air is pushed into the high-side component than could have been, and the pressure will momentarily spike before dropping. If the high-side valve is closed too late, then some of the air pushed into the high-side component will be pulled back out again as the piston moves away from top dead center (TDC; the condition where the piston is at its uppermost point of motion), and as the high-side valve finishes closing the flow will be throttled so the energy of the air flowing back into the cylinder performs less work on the piston. Potential work lost to throttling is not, in general, recovered.

[0261] Finally, the transition at the end of the regeneration stroke that opens the low-side valve to start the intake stroke impacts the work done on the piston. If the low-side valve is opened too early, then the remaining air at higher pressure from the dead volume will no longer expand, doing work on the piston, but will expand (throttled) through the opening low-side valve. If the low-side valve opens too late, then the pressure in the chamber will drop below the low-side component pressure and the piston will have to do additional work to pull the piston down and pull air through the partially open low-side valve.

[0262] Similarly, in a normal expansion process, each low-pressure and high-pressure cylinder progresses through a series of four conditions or sub-processes, each of which has an associated a valve configuration. The four conditions are (1) vent (or exhaust, or auxiliary) stroke, (2) pre-compression stroke, (3) direct drive, and (4) expansion stroke. The numbering of the phases is arbitrary in the sense that when the phases are performed in a repeating cycle, no one phase is "first" other than by convention. It is assumed in this description that all high-side and low-side valves in system **900** activate instantaneously and ideally; the implications of non-ideal valve actuation will be described subsequently. The four phases are described in detail below.

[0263] An expansion cycle begins at the bottom of stroke with the piston beginning to move upward from BDC and the low-side valve open, commencing a vent stroke. As the piston moves upward away from BDC, air in the chamber (e.g., air that was expanded in a previous cycle) is exhausted through the low-side valve to the low-side component (e.g., exhausted from an HPC to the MPV **906** or from an LPC to the ambient intake/exhaust port). A vent stroke ends when the low-side valve is closed and a pre-compression stroke begins. In general, the piston continues to move upward without interruption as a vent stroke ends and a pre-compression stroke begins.

[0264] As the cylinder piston continues to move upward, before it reaches TDC the low-side valve is closed to begin the pre-compression stroke. In pre-compression, both the high-side valve and low-side valve are closed and the air volume trapped within the chamber is compressed. In controlling the events of pre-compression, a goal is to close off the low-side valve at such a time that the air trapped in the chamber is compressed as nearly as possible to the pressure of the cylinder's adjoining high-side component when the piston reaches TDC. This allows the high-side valve to open at top of stroke with equal pressures on either side, resulting in approximately zero throttled flow through that valve and approximately zero loss of exergy due to non-work-performing loss of pressure of a quantity of gas. Timing of the start of pre-compression (closure of the low-side valve) greatly impacts the achievement of the equal-pressure goal.

[0265] Once the piston is at TDC and the high-side valve is opened, the direct-drive stroke begins. In direct drive, the piston is moving down, away from TDC, and air in the high-side component is expanding and flowing through the high-side valve into the chamber, directly driving the piston downward. The direct drive stroke continues until an appropriate mass of air has been added to the cylinder, at which point the high-side valve closes and the expansion stroke begins. In general, the piston continues to move downward without interruption as a direct-drive stroke ends and an expansion stroke begins.

[0266] In an expansion stroke, both valves are closed, and the air that was admitted to the chamber during the direct-drive phase expands, continuing to perform work upon the piston as it drives it downward. If a correct mass of air was added to the cylinder during direct drive, then the air pressure at the end of the expansion stroke will be approximately equal to the end-of-stroke target pressure at the moment when the piston reaches BDC. For a high-pressure cylinder, the end-of-stroke target pressure is the pressure in the MPV; for a low-pressure cylinder, the end-of-stroke target pressure is the vent pressure, which is typically slightly higher than atmospheric pressure. Once the piston reaches BDC, the low-side valve is opened and the cylinder begins to move up in the next vent stroke.

[0267] Valve actuation timings during a compression or expansion cycle may have a significant impact on the efficiency of the cycle. Such impact tends, in some embodiments, to be greater during an expansion cycle than during a compression cycle. During an expansion cycle, the first valve transition, as described hereinabove, is the closing of the low-side valve to begin the pre-compression. Incorrect or suboptimal timing of this valve transition may have significant consequences for the cycle. First, this valve transition is associated with the need for rapid high-side valve transition (short actuation time): as the valve is closing, the pressure in the chamber is rising quickly, resulting in throttled flow through the valve during the transition. Thus, the slower the transition, the greater the flow losses. Second, if the valve is closed later than is ideal, there will be less air in the cylinder to compress and less stroke length during which to compress it, resulting in a pressure at TDC less than the pressure in the high-side vessel. If this difference is large enough, then the pressure will be below the minimum coupling pressure and the high-side valve will be physically unable to open (i.e., the actuator will not be able to provide enough force to open the valve against the pressure difference) and the cylinder will not be able to complete the expansion cycle. If the pressure at TDC is above the minimum coupling pressure but below the pressure within the high-side component, then the high-side valve will be able to open, but gas will flow into the cylinder during the opening event without performing useful work on the piston. Contrariwise, if the low-side valve closing at the start of the pre-compression occurs too early, then the pressure within the chamber will reach the pressure within the high-side component before the piston reaches TDC, and surpass the pressure within the high-side component by the time the piston reaches TDC. In this case, more air would have been re-compressed than necessary (and less air would have been exhausted), resulting in a capacity reduction. If the pressure in the chamber reaches the high-side vessel before TDC, then the high-side valve should open when the pressures are equal rather than waiting to TDC, in order to prevent over-pressurization of the cylinder.

[0268] Once the high-side valve is open at TDC and the cylinder is in direct drive, with the piston moving down, the next transition is the closing of the high-side valve at the end of direct drive. This transition may impact both capacity and efficiency. Under perfect valve timing (i.e., closure of the high-side valve occurs at a time such that the exactly correct mass of air is drawn into the cylinder), the pressure will decrease during expansion phase and be equal to the target pressure approximately at BDC. If the valve is closed too early, the chamber will reach target pressure before BDC and will have expanded less air during the stroke than it could have (i.e., there will be a loss of capacity). If the high-side valve is closed too late, then the chamber pressure will still be above target at BDC, and the pressure difference will entail a loss once the valve is opened.

[0269] The last valve transition is at the end of the expansion stroke, when the low-side valve is opened. Ideally, the pressure at the end of the expansion stroke is equal to the target pressure exactly at BDC. If the chamber pressure drops to the target pressure before BDC, then the valve is opened to prevent the crankshaft from having to work to pull the piston down. If the piston reaches BDC before the pressure has decreased to the target, then the valve should also open. (An exception may occur if the chamber pressure is above the maximum allowable vent pressure.) This valve opening event is also impacted by the finite valve response time, in a manner similar to that described hereinabove for other valve actuation events.

[0270] It will be evident to persons familiar with pneumatics and hydraulic devices that similar considerations apply to the timing and non-ideality of valve actuation events during a compression mode of system **900**. For example, at the transition between a compression stroke and a direct-fill stroke, as described hereinabove, the high-side valve of a cylinder opens. If the high-side valve is opened too late, the pressure within the chamber will exceed that within the high-side component (e.g., high-pressure storage vessel), and gas will expand from the chamber into the high-side component in a non-work-performing manner upon valve actuation. If the high-side valve is opened too soon, the pressure within the chamber will be less than that within the high-side component, and gas will expand from the high-side component into the chamber in a non-work-performing manner upon valve actuation. Valve actuation timing is similarly constrained at other transitions between the other phases of cylinder operation during a compression process.

[0271] FIG. **10** is an illustrative plot of cylinder pressure as a function of time for four different expansion scenarios in an illustrative CAES system similar or even identical to the system **200** shown in FIG. **2**. Points A, B, C, D, and E in FIG. **10**, marked by dots, correspond to operating states of one or more components of system **200**, or to changes in such operating states, as described below. In the illustrative plot shown, Point A represents an initial state of the pneumatic cylinder assembly (**201** in FIG. **2**) during which the piston slidably disposed therein (**202** in FIG. **2**) is at top dead center and the high-side valve (**220** in FIG. **2**) between the storage reservoir (**222** in FIG. **2**) and the lower-pressure cylinder assembly **201** is opened. Point B represents the end of a direct-drive phase of operation during which the high-side valve **220** between the storage reservoir **222** and the lower-pressure cylinder assembly **201** is closed and the pressure inside the cylinder assembly **201** is approximately equal to the bottle pressure P_b (i.e., the pressure of gas inside storage reservoir **222** in FIG. **2** (e.g., 300 psi)). Point C represents the end of an expansion stage or phase, during which the quantity of gas admitted into the cylinder assembly **201** performs work on the piston **202** slidably disposed therein, from top dead center at a bottle pressure P_b to bottom dead center at an exhaust pressure $P_{exhaust}$. Point C also represents the opening actuation of the low-side valve (**221** in FIG. **2**) between the vent to atmosphere (**223** in FIG. **2**) and the cylinder assembly **201** when the piston **202** is at bottom dead center. Points A, B, and C represent approximately the same operating states in all four scenarios of operation of system **200** described hereinbelow.

[0272] Four versions of Point D (labeled D_1, D_2, D_3, and D_4 to correspond to the four different valve-actuation scenarios) represent the end of the exhaust stage and beginning of a pre-compression stage; event D thus corresponds to the closing actuation of the low-side valve **221** between the vent **223** (or a lower-pressure stage) and the cylinder assembly **201**. Four versions of event E (E_1, E_2, E_3, and E_4 for the four different valve-actuation scenarios) represent the end of the pre-compression stage, at which time the piston **202** is again at top dead center and the pressure inside the cylinder assembly **201** is approximately equal to the bottle pressure P_b. If the system is operated cyclically, Point E (any version) immediately precedes Point A and the expansion cycle may be repeated. The pressure inside the cylinder assembly **201** at the end of the pre-compression stroke (i.e., at any version of Point E) is determined by the relative timing of the closing of the low-side valve **221** (i.e., at any version of Point D).

[0273] In an idealized expansion scenario (Scenario **1**, represented by a solid line in FIG. **10**), there is no dead volume in the cylinder assembly **201** and all valve actuations occur instantaneously. The low-side valve **221** closes at Point D_1 when the piston **202** is at top dead center, instantaneously followed by the opening of the high-side valve **220** at Point E_1. In Scenario **1**, pressurization of the cylinder assembly **201** occurs immediately and with no coupling loss, as there is no volume to pressurize at the top of stroke: this instantaneous pressurization of cylinder assembly **201**, simultaneous with D_1 and E_1, is represented by the perfect verticality of the Scenario **1** line in FIG. **10** at Point D_1/E_1. Scenario **1** is shown as a reference line in FIG. **10**.

[0274] In a second scenario (Scenario **2**, represented by a bold dashed line in FIG. **10**), dead volume exists in the cylinder assembly **201**. Between Point C and Point D_1, the piston **202** performs a return stroke within the cylinder assembly **201**, with the low-side valve **221** open to allow venting of gas from the upper chamber of cylinder assembly **201**. At Point D_1, the low-side valve **221** is closed so that the gas remaining in the upper chamber of cylinder assembly **201** will be pressurized during the remainder of the return stroke. However, in Scenario **2** the low-side valve **221** is closed too late, trapping an insufficient amount of gas in the upper chamber of cylinder assembly **201**, so when the piston **202** reaches top dead center the gas inside the upper chamber of cylinder assembly **201** is at a pressure P_2 that is lower than the storage bottle pressure P_b. In other words, operational Point D_2 occurs too late in time (or piston position) to allow adequate pre-compression by Point E_2 of the gas remaining in the cylinder assembly **201**. Adequate pre-compression would be to approximately reservoir pressure P_b. When the high-side valve is opened at Point E_2, a coupling loss occurs when pressurized gas at reservoir pressure P_b is admitted from the storage reservoir **222** (or, in some other embodiments, the previous cylinder stage) to the lower-pressure cylinder assembly **201**.

[0275] In a third scenario (Scenario **3**, represented by a short-dash line in FIG. **10**), point D_3, i.e., closure of the valve **221** between the vent **223** and the cylinder assembly **201**, is timed correctly to enable the gas in the upper chamber of cylinder assembly **201** to reach a pressure P_3 when the piston **202** is at top dead center (point E_3), where P_3 substantially equal to the stored bottle pressure P_b. In other words, operational Point D_3 is at the correct time (or piston position) so that pre-compression of the gas remaining in the cylinder assembly **201** is to a pressure P_3 approximately equal to the reservoir pressure P_b at Point E_3. When the pressure P_3 inside the upper chamber of the cylinder assembly **202** is approximately equal to the pressure P_b of gas from the storage reservoir **222** (or previous cylinder stage), then there will be little or no coupling loss when valve **220** is opened, and overall system efficiency and performance will be improved.

[0276] In a fourth scenario (Scenario **4**, represented by a bold dotted line in FIG. **10**), the valve **221** between the vent **223** and the cylinder assembly **201** is closed (point D_4) too soon and the pressure inside the upper chamber of cylinder assembly **201** reaches a pressure P_4 higher than the stored bottle pressure P_b when the piston **202** reaches top dead center (Point E_4). In other words, operational Point D_4 is too early in time (or piston position) and the pre-compression of the remaining air in the upper chamber of the cylinder assembly **201** exceeds reservoir pressure P_b at Point E_4. When the valve **220** between the storage vessel **222** (or earlier cylinder stage) and the cylinder assembly **201** is opened (Point E_4), the difference in pressure between results in a coupling loss.

[0277] The system controller (**226** in FIG. **2**) may be programmed in such a manner as to receive feedback (e.g., information from measurements) from previous expansion cycles and the present state of system **200**. Such feedback, whether informational or mechanical, may be used to adjust the timing of Point D, the closing of the low-side valve **221** to commence pre-compression. For example, a lookup table may be employed to set valve actuation times in response to measurements of conditions in the system. In one embodiment, the controller **226** utilizes timing information from previous expansion strokes and pressure measurements of the cylinder **201** and reservoir **222** at the completion of the pre-compression process (Point E) to set the next time of closure of valve **221**. Such feedback may provide optimal performance of the expander/compressor, improving efficiency, performance, and system component lifetime. The time of opening of valve **220** and other events in the expansion cycle may also be adjusted by the system controller **226** based on feedback.

[0278] Thus, in accordance with embodiments of the invention, efficiency is maximized during a gas compression or expansion by a combination of feedforward and feedback control of the valve timing where either early or late actuation of the valves would reduce overall efficiency of the compression or expansion process. This efficiency of the valve timing may be calculated mathematically by comparing the work required with ideal valve timing to the actual measured work with the experimental or sub-optimal valve timing. Other factors that are measurable or may be calculated via measurable values and impact efficiency are the rate of pressure decrease of the storage system during the expansion process, the rate of mass storage during the compression process, and the degree of under- or over-pressurization during either process. For both the compression and expansion processes, there is typically a known ideal pressure profile that may be approached by optimizing valve timing. The ideal pressure profile may be approached by determining valve timing that minimizes or maximizes the integrated work about key points in the pressure-volume curve. Deriving and subjecting the system to such timing values constitutes the feedforward component of the valve timing controller. Correcting for modeling uncertainties, system disturbances, quickly occurring system changes, or longer-term system drift is performed by incorporating representative measurements in the valve timing controller, and this constitutes the feedback component of the valve timing controller. Each valve transition event may be optimized for efficiency as described herein. For example, in opening a high-side valve at the end of a compression stroke to begin direct fill, an early actuation would cause gas to travel backwards from the high-side reservoir into the cylinder, reducing efficiency of the compression process, and late actuation would result in a pressure spike, increasing work required to complete the compression and causing a loss of useful energy when the valve opens and the air in the cylinder pressure equalizes with storage. Thus, in short, as utilized herein, "maximizing efficiency" of a compression or expansion process entails valve-timing optimization to minimize or eliminate lost work during an expansion of a particular amount of gas or minimizing or eliminating additional work required to compress a particular amount of gas.

[0279] FIG. **11** is a graphical display of experimental test data from an expansion process involving expansion of gas first in a high-pressure pneumatic cylinder and then in a low-pressure pneumatic cylinder. That is, in the physical system from which the data in FIG. **11** were drawn, a first, high-pressure cylinder expanded gas from a high pressure P_H to an intermediate pressure P_I while a second, low-pressure cylinder either (a) did not pre-compress the gas in its expansion chamber from a preexisting default pressure P_L (PLOT A in FIG. **11**) or (b) pre-compressed the gas in its expansion chamber from the preexisting default pressure P_I, to the intermediate pressure P_I (PLOT B in FIG. **12**).

[0280] In PLOT A, the pressure in the expansion chamber of the first, high-pressure cylinder as a function of time during expansion of the chamber's contents from P_H to P_I is indicated by curves **1100** and **1104**, and the pressure in the expansion chamber of the second, low-pressure cylinder as a function of time is indicated by curves **1102**, **1106**, and **1108**. Expansion of the gas in the high-pressure cylinder is indicated by curve **1100**. The approximately constant pressure of the gas in the low-pressure cylinder, being exhausted by a return stroke concurrently with the expansion recorded by curve **1100**, is indicated by curve **1102**.

[0281] At the moment corresponding to labeled point A_1, a valve is opened to place the expansion chamber of the first, high-pressure cylinder in fluid communication with the expansion chamber of the second, low-pressure cylinder. Because the two chambers are at different pressures at that time (i.e., the gas in the high-pressure cylinder chamber is at P_1 and the gas in the low-pressure cylinder chamber is at P_L), after point A_1 (valve opening) the pressure within the chamber of the high-pressure cylinder decreases rapidly to an intermediate pressure P_{I2} (curve **1104**) while the pressure within the chamber of the low-pressure cylinder increases rapidly to the intermediate pressure P_{I2} (curve **1106**). By point A_2, shortly after A_1, the pressures in the two cylinder chambers have equilibrated. The rapid expansion indicated by curve **1104** performs no work on any mechanical component of the system and therefore entails a loss of available energy (i.e., a dead-volume loss). At point A_2, an expansion occurs in the expansion chamber of the low-pressure cylinder, from P_{I2} to some low end pressure P_{EI} (curve **1108**).

[0282] In PLOT B, the pressure in the expansion chamber of the first, high-pressure cylinder as a function of time during expansion of the chamber's contents from P_H to I_2 is indicated by curve **1110**, and the pressure in the expansion chamber of the second, low-pressure cylinder as a function of time during pre-compression of the gas in the low-pressure cylinder chamber from P_L to approximately P_I is indicated by curve **1112**. Prior to labeled point B, the low-pressure cylinder is performing an exhaust stroke, and gas is being expelled from the expansion chamber of the low-pressure chamber at approximately constant pressure P_L through an open exhaust valve. At the moment corresponding to point B, the valve permitting gas to exit the expansion chamber of the low-pressure cylinder is closed, trapping a fixed quantity of gas within the chamber at pressure P_L. This quantity of gas is then compressed to pressure P_I as indicated by curve **1112**.

[0283] At point C, the pressure in the expansion chamber of the low-pressure cylinder is approximately equal to the pressure P_I in the expansion chamber of the high-pressure cylinder and a valve is opened to place the two chambers in fluid communication with each other. Because the two chambers are at approximately equal pressures, there is no significant equilibration upon valve opening (i.e., there is no curve in PLOT B corresponding to the expansion of gas in the high-pressure cylinder indicated by curve **1104** in PLOT A) and thus little or no energy loss due to equilibration. Subsequent to point C, an expansion occurs in the expansion chamber of the low-pressure cylinder, from P_I to some low end pressure P_{E2} (curve **1114**). In FIG. **11**, end pressure P_{E2} is not equal to end pressure P_{E1}, but this is not a necessary result of the pre-compression process illustrated in FIG. **11** and end pressure P_{E2} may have any of a range of values in accordance with embodiments of the present invention.

[0284] FIG. **12** is an illustrative plot of the ideal pressure-volume cycle in a cylinder operated as either a compressor or expander. FIG. **12** provides explanatory context for subsequent figures. Instantaneous and perfectly timed valve actuations are presumed for the system whose behavior is represented in FIG. **12**. The horizontal axis represents volume (increasing rightward) and the vertical axis represents pressure (increasing upward). In FIG. **12**, the volume represented by the horizontal axis is the volume of the expansion/compression chamber of a cylinder assembly that is similar or identical to cylinder **201** in FIG. **2** and is being operated as either an expander or compressor. The four curves in FIG. **12** (labeled **1**, **2**, **3**, and **4**) form a cyclic loop; each curve represents one of the four distinct phases of operation described above for both compression and expansion. For a cylinder operating as a compressor, Curves **1** through **4** are traversed counterclockwise (e.g., in order **1**, **4**, **3**, **2**), where Curve **1** represents the direct-fill phase; Curve **2** represents the compression stroke; Curve **3** represents the intake stroke; and Curve **4** represents the regeneration stroke. For a cylinder operating as an expander, Curves **1** through **4** are traversed clockwise (e.g., in order **1**, **2**, **3**, **4**). Curve **1** represents the direct-fill phase; Curve **2** represents the expansion stroke; Curve **3** represents the exhaust stroke; and Curve **4** represents the precompression stroke. Points A, B, C, and D in FIG. **12** represent valve transition events. As each Event (A, B, C, or D) is traversed during cyclic operation of the cylinder, the valve actuations that occur at each point depend on whether the cylinder is being operated as an expander or compressor. Specifically, if the cylinder is being operated as a compressor, at Event A a high-side valve V**1** (**220** in FIG. **2**) is closed and a low-side valve V**2** (**221** in FIG. **2**) remains closed; at Event D, V**1** remains closed and V**2** is opened; at Event C, V**1** remains closed and V**2** is opened; and at Event B, V**1** is opened while V**2** remains closed. If the cylinder is being operated as an expander, at Event A, V**1** is opened and V**2** remains closed; at Event B, V**1** is closed and V**2** remains closed; at Event C, V**1** remains closed and V**2** is opened; at Event D, V**1** remains closed and V**2** is closed.

[0285] As will be made clear by subsequent figures, the effects of finite (non-ideal, nonzero) actuation times for valves V**1** and V**2** at all valve transitions in FIG. **12** tend to decrease system capacity and/or efficiency and to alter the shapes of Curves **1**, **2**, **3**, and **4**. Also, mistiming of valve transitions in FIG. **12** may decrease system capacity and/or efficiency. An optimal actuation timing exists for each valve actuation event under any given conditions of operation; this optimal time will, in general, change as the conditions under which the system is being operated change (e.g., as the pressure in a high-pressure gas storage reservoir gradually increases or decreases).

[0286] FIG. **13** is an illustrative plot of cylinder chamber pressure as a function of cylinder chamber volume for three different expansion scenarios in an illustrative CAES system similar or even identical to the system **200** shown in FIG. **2**. FIG. **13** shows the effects of early, correctly timed, and tardy closure of valve V**2** in the transition (Event D in FIG. **12**; not shown in FIG. **13**) from intake phase to pre-compression phase (i.e., from Curve **3** to Curve **4** in FIG. **12**). The three scenarios depicted in a pressure-volume plot in FIG. **13** greatly resemble the scenarios depicted in a pressure-time plot in FIG. **10**, as shall be explained below.

[0287] The region of the expander's pressure-volume cycle portrayed in FIG. **13** corresponds to Point A in FIG. **12** as defined for an expansion process. (In a non-ideal system, events occurring during a valve transition event do not occur without changes of pressure and volume, and so cannot be represented by a single point in a pressure-volume plot. The instant that an actuated valve is commanded to transition is one representation of the start of a valve transition.) The dashed curve **1302** represents the pressure-volume history of the gas within the cylinder chamber during the latter part of the pre-compression phase (Curve **4** in FIG. **12**) for a scenario (the Correct V**2**(D) Closure scenario) in which closure of V**2** (the low-pressure valve, **221** in FIG. **2**) in the transition from vent phase to pre-compression phase occurs at an optimal time. The Correct V**2**(D) Closure scenario corresponds to the curve passing through points D_2 and E_2 in FIG. **10**.

[0288] The thick solid curves **1304**, **1306** represent the pressure-volume history of the gas during the latter part of the pre-compression phase for a scenario (the Late V**2**(D) Closure scenario) in which closure of V**2** in the transition from vent phase to pre-compression phase is tardy. The Late V**2**(D) Closure scenario corresponds to the curve passing through points D_3 and E_3 in FIG. **10**. The thin solid curves **1308**, **1310** represent the pressure-volume history of the gas during the latter part of the pre-compression phase for a scenario (the Early V**2**(D) Closure scenario) in which closure of V**2** in the transition from vent phase to pre-compression phase occurs too early. The Early V**2**(D) Closure scenario corresponds to the curve passing through points D_4 and E_4 in FIG. **10**.

[0289] All curves in FIGS. **13-16** are traversed, in time, in the sense shown by the arrowheads attached to each curve.

[0290] In the systems whose behavior is partly represented by FIGS. **13-16**, the gas volume of the high-side component (e.g., high-pressure storage reservoir **222** in FIG. **2**) that is connected to the cylinder through V**1** is presumed to be sufficiently large that exchanges of air between the high-side component and the cylinder chamber do not substantially change the pressure P_H of the gas within the high-side component.

[0291] In the Correct V**2**(D) Closure Scenario, closure of V**2** traps the correct amount of air in the cylinder chamber to produce at TDC a chamber pressure approximately equal to that the pressure P_H within the high-side component. In FIG. **13**, P_H is approximately 21.5 megapascals (MPa). At point **1312**, when the gas in the chamber approximates pressure P_H, V**1** is opened; since both the gas in the high-side component and the gas in the chamber are at or near to P_H, the pressure of the gas in the chamber does not change significantly. Subsequently, gas is transferred during direct-drive phase at approximately constant P_H from the high-side component to the chamber as the piston descends in the cylinder (solid curve **1314**). The effects of dead volume during the transition from pre-compression phase to direct-drive phase are minimized or even nonexistent in the Correct V**2**(D) Closure Scenario.

[0292] In the Late V**2**(D) Closure Scenario, closure of V**2** traps insufficient air in the chamber to produce at TDC a chamber pressure approximately equal to P_H. Instead, the gas in the chamber achieves some lower pressure P_{H2}; in FIG. **13**, P_{H2} is approximately 15 MPa. At point **1316**, V**1** opens; since the gas in the high-side component is at a higher pressure (P_H) than the gas in the chamber (P_{H2}), gas from the high-side component rapidly enters the chamber, raising the pressure of the gas in the chamber while the volume of the chamber does not change significantly. This pressure-volume change at near-constant volume is represented by curve **1306**. Potentially useful pressure energy is lost during this non-work-performing expansion of gas from the high-pressure component into the chamber, i.e., a dead-volume loss occurs. At the end of curve **1306**, the gas in the chamber reaches point **1312**, after which the Late V**2**(D) Closure Scenario coincides with the Correct V**2**(D) Closure Scenario (curve **1314**).

[0293] In the Early V**2**(D) Closure Scenario, closure of V**2** traps more air in the chamber than is needed to produce at

TDC a pressure approximately equal to P_H. Instead, the gas achieves some higher pressure P_{H3}; in FIG. **13**, P_{H3} is approximately 25 MPa. At point **1318**, V**1** opens; since the gas in the high-side component is at a lower pressure (P_H) than the gas in the chamber (P_{H3}), gas from chamber rapidly enters the high-side component, lowering the pressure of the gas in the chamber while its volume does not change significantly. This pressure-volume change at near-constant volume is represented by curve **1310**. Potentially useful pressure energy is lost during this non-work-performing expansion of gas from the chamber into the high-pressure component, i.e., a dead-volume loss occurs. At the end of curve **1318**, the gas in the chamber is at point **1312**, after which the Early V**2**(D) Closure Scenario coincides with the Correct V**2**(D) Closure Scenario (curve **1314**).

[0294] FIGS. **14**A-**14**C are illustrative plots of cylinder chamber pressure as a function of cylinder chamber volume for two different expansion scenarios in an illustrative CAES system similar or even identical to the system **200** shown in FIG. **2**. The two scenarios depicted in FIGS. **14**A-**14**C are the Correct V**1**(A) Opening Scenario and the Late V**1**(A) Opening Scenario. The region of the expander's pressure-volume cycle portrayed in FIGS. **14**A-**14**C corresponds to Point A in FIG. **12** as defined for an expansion process. Valves V**1** and V**2** are defined as for FIG. **13**. FIGS. **14**A-**14**C show the effects of correctly timed and tardy opening of valve V**1** in the transition from pre-compression phase to direct-drive phase (i.e., from Curve **4** to Curve **1** in FIG. **12**). It is presumed that in the scenarios depicted in FIGS. **14**A-**14**C, the previous valve transition (Event D in FIG. **12**) was optimally made. Three separate figures, FIGS. **14**A-**14**C, depicting curves **1402**, **1404**, and **1406** respectively, are employed to avoid partial obscuration of curve **1402** by curve **1406**.

[0295] Curve **1402** of FIG. **14**A represents the volume-pressure history of both scenarios until point **1408** is reached. Thereafter, the two scenarios diverge. In the Correct V**1**(A) Opening Scenario, V**1** is opened at point **1408**, just as the cylinder reaches TDC. Because pre-compression was optimally performed, the pressure in the chamber approximates P_H and there is little or no gas exchange between the chamber and the high-pressure component when V**1** is opened, and subsequent to point **1408** gas is vented during a direct-drive phase at approximately constant P_H from the high-side component to the chamber as the piston descends in the cylinder (curve **1404** in FIG. **14**B). In FIGS. **14**A-**14**C, P_H is approximately 1.82 MPa. The effects of dead volume during the transition from pre-compression phase to direct-drive phase are minimal or nonexistent in the Correct V**1**(A) Opening Scenario.

[0296] In the Late V**1**(A) Opening Scenario, V**1** is not opened at point **1408**, when the piston is at TDC, but remains closed for a time thereafter. As the piston descends, the pressure-volume state of the gas in the chamber thus begins to retrace curve **1402** in the opposite direction (left-hand portion of curve **1406**, FIG. **14**C): that is, the gas trapped in the chamber simply begins to re-expand. At point **1410** (FIG. **14**C), at which the gas in the chamber has achieved some pressure P_{H4} significantly lower than P_H, V**1** is opened. Gas then enters the chamber from the high-side component, raising the pressure of the gas in the chamber to P_H while the volume of the chamber is increasing (rising portion of curve **1406**). Potential energy is lost and less work is done during this non-work-performing expansion of gas from the high-pressure component into the chamber, i.e., a dead-volume loss occurs.

[0297] FIG. **15** is an illustrative plot of cylinder chamber pressure as a function of cylinder chamber volume for three different compression scenarios in an illustrative CAES system similar or even identical to the system **200** shown in FIG. **2**. FIG. **15** shows the effects of early, correctly timed, and tardy opening of valve V**1** in the transition (Event B in FIG. **12**, defined for compression mode) from compression phase to direct-fill phase (i.e., from Curve **2** to Curve **1** in FIG. **12**). The region of the expander's pressure-volume cycle portrayed in FIG. **15** corresponds to Point B in FIG. **12** as defined for a compression process.

[0298] The dotted curve **1502** (partly obscured by thick solid curve **1504**) represents the pressure-volume history of the gas within the cylinder chamber during the latter part of the compression phase (Curve **2** in FIG. **12**) for a scenario (the Correct V**1**(B) Opening scenario) in which opening of V**1** in the transition from compression phase to direct-fill phase occurs at an optimal time. The thick solid curve **1504** represents the pressure-volume history of the gas in the chamber during the latter part of the compression phase for a scenario (the Late V**1**(B) Opening scenario) in which opening of V**1** in the transition from compression phase to direct-fill phase is tardy. The thin solid curve **1506** represents the pressure-volume history of the gas during the latter part of the compression phase for a scenario (the Early V**1**(B) Opening scenario) in which opening of V**1** in the transition from compression phase to direct-fill phase occurs too early.

[0299] In the system whose behavior is partially depicted in FIG. **15**, V**1** and V**2** have nonzero actuation times. Therefore, the optimal time of opening of V**1** (i.e., the timing of V**1** opening for the Correct V**1**(B) Opening scenario) occurs at point **1508**, before the pressure in the chamber reaches P_H. At point **1508** the gas in the cylinder chamber has not yet achieved the pressure P_H of the gas in the high-pressure component, but only a small amount of gas is throttled through the partly-open valve into the chamber as the pressure-volume state of the gas in the chamber evolves from point **1508** to point **1510**, at which time the pressure in the chamber approximates P_H. A small amount of pressure overshoot may occur (represented by the small hump in curve **1502**); subsequently, gas is exhausted during direct-fill phase at approximately constant P_H from the cylinder chamber to the high-side component as the piston continues to ascend in the cylinder (curve **1512**). The effects of dead volume during the transition from pre-compression phase to direct-drive phase are minimal or nonexistent in the Correct V**1**(B) Opening Scenario.

[0300] In the Late V**1**(B) Opening scenario, V**1** is opened at point **1514**, by which time the gas in the chamber has reached a pressure P_{H5}, significantly higher than P_H. After the tardy opening of V**1**, gas in the chamber transfers into the high-pressure component as the pressure in the chamber decreases (left-hand side of large hump in curve **1514**). Energy is lost during this non-work-performing expansion of gas from the chamber into the high-pressure component, i.e., a dead-volume loss occurs.

[0301] A similar curve would be traced even if the valve started to transition open at point **1510** when the pressures are equal, due to the nonzero time to open the valve and pressure rise that occurs with a partially open valve. Notably, in systems employing a pressure-driven check valve for V**1** rather than an actuated valve, a pressure-volume history similar to that of the Late V1(B) Opening scenario (curve **1504**), although not necessarily so extreme, typically occurs in every compression cycle, as an overpressure (e.g., P_{H5} or some other pressure significantly higher than P_H) must be achieved on the chamber side of V**1** with respect to the high-pressure-component side of V**1** in order for V**1** to be actuated. The use of actuated valves rather than check valves in CAES systems is thus advantageous in this, as well as in other, respects.

[0302] In the Early V**1**(B) Opening scenario, V1 is opened at point **1516**, by which time the gas in the chamber has reached a pressure of only P_{H6}, significantly lower than P_H. After the early opening of V**1**, gas in the high-pressure component flows into the chamber as the pressure in the chamber increases (right-hand side of curve **1506**). Energy is lost during this non-work-performing expansion of gas from the high-pressure component into the chamber, i.e., a dead-volume loss occurs.

[0303] FIG. **16** is an illustrative plot of cylinder pressure as a function of cylinder volume for three different compression scenarios in an illustrative CAES system similar or even identical to the system **200** shown in FIG. **2**. FIG. **16** shows the effects of early, correctly timed, and tardy opening of valve V2 in the transition (Event D in FIG. **12**, defined for compression mode) from regeneration phase to intake phase (i.e., from Curve **4** to Curve **3** in FIG. **12**). The region of the expander's pressure-volume cycle portrayed in FIG. **16** corresponds to Point D in FIG. **12** as defined for a compression process.

[0304] The dotted curve **1602** represents the pressure-volume history of the gas within the cylinder chamber during the latter part of the regeneration phase (Curve **4** in FIG. **12**) for a scenario (the Correct V2(D) Opening scenario) in which opening of V**2** in the transition from regeneration phase to intake phase occurs at an optimal time. The thick solid curve **1604** represents the pressure-volume history of the gas in the chamber during the latter part of the compression phase for a scenario (the Late V**2**(D) Opening scenario) in which opening of V**2** in the transition from regeneration phase to intake phase is tardy. The thin solid curve **1606** represents the pressure-volume history of the gas during the latter part of the compression phase for a scenario (the Early V**2**(D) Opening scenario) in which opening of V2 in the transition from regeneration phase to intake phase occurs too early.

[0305] In the system whose behavior is partially depicted in FIG. **16**, V**1** and V**2** have nonzero actuation times. Therefore, the optimal time of opening of V**2** (the timing of V**2** opening for the Correct V**2**(D) Opening scenario) occurs at point **1608**, before the pressure in the chamber approximates P_L (the pressure of the low-pressure component that communicates with the cylinder through V**2**, e.g., vent **223** in FIG. **2**). The gas in the cylinder chamber has not yet decreased to the pressure P_L of the gas in the low-pressure component, but only a small amount of gas is throttled through the partly-open valve V**2** into the chamber as the pressure-volume state of the gas in the chamber evolves from point **1608** to point **1610**, at which time the pressure in the chamber approximates P_L. A small amount of pressure overshoot may occur (dip in curve **1602**); subsequently, gas is admitted to the chamber during the intake phase at approximately constant P_L from the low-side component to the chamber as the piston continues to descend in the cylinder. The effects of dead volume during the transition from regeneration phase to intake phase are minimal or nonexistent in the Correct V**2**(D) Opening Scenario.

[0306] In the Late V**2**(D) Opening scenario, V**2** is opened at point **1612**, by which time the gas in the chamber has decreased to a pressure of P_{L2}, significantly lower than P_L. After the tardy opening of V**2**, gas from the low-pressure component flows into the chamber as the pressure in the chamber increases (right-hand side of large dip in curve **1604**). Energy is lost during this non-work-performing expansion of gas into the chamber from the low-pressure component: i.e., a dead-volume loss occurs.

[0307] Notably, in systems employing a pressure-driven check valve for V**2** rather than an actuated valve, a pressure-volume history similar to that of the Late V**2**(D) Opening scenario (curve **1604**) typically occurs in every compression cycle, as an underpressure (e.g., P_{L2} or some other pressure significantly lower than P_L) must be achieved on the chamber side of V**2** with respect to the low-pressure-component side of V**2** in order for V**2** to be actuated. The use of actuated valves rather than check valves in CAES systems is thus advantageous in this, as well as in other, respects.

[0308] In the Early V**2**(D) Opening scenario, V**2** is opened at point **1614**, by which time the gas in the chamber has only declined to a pressure of P_{L3}, significantly higher than P_L. After the early opening of V**2**, gas in the chamber exits to the low-pressure component as the pressure in the chamber decreases (left-hand side of curve **1606**). Energy is lost during this non-work-performing expansion of gas from chamber into the low-pressure component, i.e., a dead-volume loss occurs.

[0309] It will be clear to persons familiar with the sciences of hydraulics and pneumatics that considerations similar to those described above with reference to FIGS. **10** and **13-16** also pertain to optimal, early, and late valve actuation at valve transition events not explicitly depicted herein, for both compression and expansion modes of operation of a CAES system. Optimization by the means described of valve actuations at all valve transition events in a CAES system is contemplated and within the scope of the invention. In brief, wherever two or more volumes of gas are to be brought into fluid communication with each other in the course of operating a CAES system, optimally timed valve actuations will generally be those that occur at moments calculated to bring the volumes of gas into fluid communication with each other when their pressures are approximately equal.

[0310] FIG. 17A is a schematic cross-sectional drawing of the major components of an illustrative poppet valve **1700** that employs a hydraulic actuation mechanism **1702** to open and close a port (or opening) **1704** by moving a disc (or valve member) **1706** connected to a stem (or rod) **1708**. In other embodiments, the valve **1700** is actuated by electrical and/or mechanical actuation systems. The valve may include a mechanical or pneumatic spring (not shown) to bias the valve towards closing, cushion opening forces, and/or replace or supplement the closing actuation mechanism. The valve **1700** shown in FIG. **17**A is a low-side valve, as defined above.

[0311] As depicted in FIG. **17**A, the actuation mechanism **1702** features a hydraulic cylinder **1710** containing a piston **1712**. The piston **1712** is connected to stem **1708** that passes out of the actuation mechanism **1702** through a gasket **1714**, passes into the body **1716** of the valve **1700**, and passes through a ring **1718** and additional gaskets **1720**. Exiting the ring **1718**, the stem **1708** passes into a flow chamber **1722** and through port **1704**. The stem **1708** is connected to disc **1706**. The port **1704** is surrounded by a lip or flange **1724** termed the "seat." The seat is shaped and sized so that the entire periphery of the disc **1706** may make tight contact with the surface of the seat **1724**. A second port **1726** is typically permanently open and may be connected to piping (not shown). The stem **1708**, piston **1712**, disc **1706**, port **1704**, and seat **1724** may be circular in cross-section or may have some other cross-sectional form.

[0312] As depicted in FIG. **17**A, the low-side valve **1700** is closed. That is, the stem **1708**, actuator piston **1712**, and disc **1706** are in a position that places the disc **1706** in firm contact with the seat **1724**, occluding the port **1704**. If greater force is exerted by fluid on the outside of the disc **1706** than by fluid within the flow chamber **1722**, the valve will remain closed even if no force is applied to the stem **1708** by the actuation mechanism **1702**. (The disc **1706** is too large to pass through the port **1704**.) A drain **1728** is provided for fluid leakage that may occur from the actuation mechanism **1702** through gasket **1714** or from chamber **1722** through gasket **1720**.

[0313] The valve **1700** may be designed to open at a predetermined pressure differential determined by the area ratios on either side of disc **1706**. The valve **1700** may be responsive to a control system (e.g., control system **122** or control system **226**) that actuates the valve at a time just prior to the valve checking open due to the predetermined pressure differential such that the pressure drop across the valve **1700** stays below a threshold value (e.g., <2% of the absolute pressure), improving the efficiency of the energy storage system. Further, the actuation of the valve **1700** may be such as to bias the valve towards opening or closing, and the actual hydraulic actuation may need not occur at the precise time of valve opening or closing. The control system may operate on a feedback loop that adjusts valve timing based on pressure drop across the valve **1700** on a previous valve opening or closing occurrence or based on another feedback measurement such as actuation time of a previous occurrence. A pneumatic spring (not shown) may be included in the valve **1700** to further bias the valve **1700** towards closing. The pressure within the pneumatic spring may be adjusted during operation of the system and may even be vented for part of a cylinder stroke in order to achieve optimal valve performance.

[0314] FIG. **17**B depicts the high-side valve **1700** in a fully open position. That is, the stem **1708**, actuator piston **1712**, and disc **1706** are in a position that places the disc **1706** as far out of contact with the seat **1724** as the dimensions of the mechanism permit, opening the port **1704**.

[0315] FIG. **18**A is a schematic cross-sectional drawing of the major components of an illustrative poppet valve **1800** that employs a hydraulic actuation mechanism **1802** to open and close a port **1804** by moving a disc **1806** connected to a stem **1808**. In other embodiments, the valve **1800** is actuated by electrical and/or mechanical actuation systems. The valve **1800** shown in FIG. **18**A is a high-side valve, as defined above.

[0316] As depicted in FIG. **18**A, the actuation mechanism **1802** features a hydraulic cylinder **1810** containing a piston **1812**. The piston **1812** is connected to stem **1808** that passes out of the actuation mechanism **1802** through a gasket **1814**, passes into the body **1816** of the valve **1800**, and passes through a ring **1818** and additional gaskets **1820**. Exiting the ring **1818**, the stem **1808** passes into a flow chamber **1822** and through port **1804**. The stem **1808** is connected to disc **1806**. The port **1804** is surrounded by a lip or flange **1824** termed the "seat." The seat is shaped and sized so that the entire periphery of the disc **1806** may make tight contact with the surface of the seat **1824**. A second port **1826** is typically permanently open and may be connected to piping (not shown). The stem **1808**, piston **1812**, disc **1806**, port **1804**, and seat **1824** may be circular in cross-section or may have some other cross-sectional form.

[0317] As depicted in FIG. **18**A, the high-side valve **1800** is closed. That is, the stem **1808**, actuator piston **1812**, and disc **1806** are in a position that places the disc **1806** in firm contact with the seat **1824**, occluding the port **1804**. If less force is exerted by fluid on the outside of the disc **1806** than by fluid within the flow chamber **1822**, the valve will remain closed even if no force is applied to the stem **1808** by the actuation mechanism **1802**. (The disc **1806** is too large to pass through the port **1804**.) A drain **1828** is provided for fluid leakage that may occur from the actuation mechanism **1802** through gasket **1814** or from the chamber **1822** through gasket **1820**.

[0318] The valve **1800** may be designed to open at a predetermined pressure differential determined by the area ratios on either side of disc **1806**. The valve **1800** may be responsive to a control system (e.g., control system **122** or control system **226**) that actuates the valve at a time just prior to the valve checking open due to the predetermined pressure differential such that the pressure drop across the valve **1800** stays below a threshold value (e.g. <2% of the absolute pressure), improving the efficiency of the energy storage system. Further, the actuation of the valve **1800** may be such as to bias the valve towards opening or closing, and the actual hydraulic actuation may need not occur at the precise time of valve opening or closing. The control system may operate on a feedback loop that adjusts valve timing based on pressure drop across the valve **1800** on a previous valve opening or closing occurrence or based on another feedback measurement such as actuation time of a previous occurrence. A pneumatic spring (not shown) may be included in the valve **1800** to further bias the valve **1800** towards closing. The pressure within the pneumatic spring may be adjusted during operation of the system and may even be vented for part of a cylinder stroke in order to achieve optimal valve performance.

[0319] FIG. **18**B depicts the high-side valve **1800** in a fully open position. That is, the stem **1808**, actuator piston **1812**, and disc **1806** are in a position that places the disc **1806** as far out of contact with the seat **1824** as the dimensions of the mechanism permit, opening the port **1804**.

[0320] FIG. **19**A is a schematic cross-sectional drawing of several components of a cylinder assembly **1900**. FIG. **19**A depicts one end of a pneumatic or pneumatic-hydraulic cylinder **1902**. Portions of the cylinder **1902** and cylinder assembly **1900** are not depicted in FIG. **19**A, as indicated by the irregular dashed line **1904**. A high-side valve **1906** and a low-side valve **1908** are integrated with the head **1910** (end cap) of the cylinder **1902**. That is, the valves **1906**, **1908** are in this embodiment not connected to a chamber **1912** within the cylinder **1902** by piping, but communicate directly with the chamber **1912**. High-side valve **1906** is substantially identical to the valve depicted in FIGS. **18**A and **18**B. Low-side valve **1908** is substantially identical to the valve depicted in FIGS. **17**A and **17**B. The valves may be sized in a manner to allow low pressure drop (e.g., <2% of absolute pressure) when passing two-phase flow (i.e., both gas and liquid) including a substantial volume fraction of liquid (e.g., >20% of the total volume is liquid). The mass of the valves and actuation forces may be sized such that actuation time is rapid with respect to cylinder stroke time (e.g., <5% of total stroke time).

[0321] As depicted in FIG. **19**A, a port **1914** of high-side valve **1906** communicates with a channel **1913** within the cylinder head **1910**. The channel **1913** may in turn be connected with piping that places channel **1913** in fluid communication with a store of gas at high pressure (e.g., 3,000 psi). Port 1916 of low-side valve **1908** communicates with a channel **1918** within the cylinder head **1910**. The channel **1918** may in turn be connected with piping that places channel **1918** in communication with a vent to the atmosphere (not shown), with a store of pressurized gas (not shown), or with the inlet of another pneumatic or pneumatic-hydraulic cylinder (not shown).

[0322] In the state of operation depicted in FIG. **19**A, high-side valve **1906** is open to admit gas from a high-pressure store (not shown) into chamber **1912** of the cylinder **1902**. Low-side valve **1908** is closed, and, barring the application of sufficient force by the actuation mechanism **1920** of the valve **1908**, will remain closed by the pressure within the chamber **1912**, which is high relative to the pressure within the channel **1918**.

[0323] In a state of operation (not shown) subsequent to the state depicted in FIG. **19**A, valves **1906** and **1908** are both closed. In this state, gas within the chamber **1912** may be expanded, performing work on a piston (not shown) within the cylinder **1902**. Valve **1906** may be configured so that if for any reason pressure of the fluid in chamber **1912** exceeds that of the gas in the high-pressure store by some predetermined amount, valve **1906** opens, acting as a pressure-relief to prevent overpressurization of the cylinder **1902**.

[0324] FIG. **19**B depicts the cylinder assembly of FIG. **19**A in another state of operation. In the state of operation depicted in FIG. **19**B, high-side valve **1906** is closed and, barring the application of sufficient force by the actuation mechanism **1922** of the valve **1906**, will be kept closed by the pressure within the channel **1913**, which is high relative to the pressure within the chamber **1912**. Low-side valve **1908** is open to allow transfer (e.g., venting) of gas from chamber **1912**. When sufficient gas is transferred from chamber **1912** in this state of operation, the cylinder assembly **1900** may be returned to the state of operation depicted in FIG. **19**A in order to admit another installment of high-pressure gas to chamber **1912**.

[0325] FIG. **20** is a schematic drawing of components of an illustrative system **2000** that includes a hydraulic activation cylinder **2002** and a poppet valve **2004**. Various components of system **2000** have been omitted for clarity. System **2000** does not employ the invention but displays a context in which various embodiments of the invention may be employed. System **2000** is shown in a vertical orientation for illustrative purposes in FIG. **20**, but other orientations may be employed.

[0326] Activation cylinder **2002** includes a cylinder **2006**, a piston **2008** that divides the interior of cylinder **2006** into a distal chamber **2010** and a proximal chamber **2012**, and a stem **2014** connected to piston **2008**. The chambers **2010**, **2012** are fluid-filled; appropriate ports, valves, piping, and other devices (not shown) enable the controlled entry and exit of a substantially incompressible fluid from the chambers **2010**, **2012**, the pressure of which may drive the piston **2008** in the distal or proximal direction. The stem **2014** passes through the proximal end-cap **2016** of the cylinder **2006** and into the body (not shown) of the poppet valve **2004**. Within the poppet valve **2004**, the stem **2014** is attached to a disc **2018**. The valve **2004** comprises a beveled seat **2020** of a suitable material recessed into an annular groove or channel surrounding an opening **2021** in a cylinder end-cap **2022** (shown in part). The valve **2004** is closed when the disc **2018** is in contact with the seat **2020**, and fully open when the disc **2018** is distance h from the seat **2020**. When valve **2004** is open, fluid may move through the opening **2021** and the gap between the disc **2018** and seat **2020**. The piston **2008**, stem **2014**, and disc **2018** move in unison; movement of the disc **2018** through distance h is driven by movement of the piston **2008** through distance h.

[0327] FIG. **21** is a schematic drawing of various components of an illustrative assembly **2100** embodying aspects of the invention. Assembly **2100** is depicted in a vertical orientation in FIG. **21** but may be oriented otherwise (e.g., horizontally) in various other embodiments. Assembly **2100** includes a high-side poppet valve **2102** that controls fluid communication between the interior of a pneumatic cylinder (not shown) and some source or destination for gas (not shown) exterior to the pneumatic cylinder. Valve **2102** features a disc **2104**; when the disc **2104** is in contact with the seat **2106**, valve **2102** is closed. Disc **2104** is connected to a stem **2108**, which passes through two gaskets **2110**, **2112**. Between gaskets **2110** and **2112** is a catchment chamber **2114** whose function shall be explained below. The catchment chamber **2114** communicates with a connection **2116**, e.g., a connection to a source of fluid at high pressure, low pressure, or variable pressure.

[0328] The poppet valve **2102** is actuated by an actuation mechanism **2118**. The actuation mechanism **2118** includes a piston **2120** that divides its interior into two chambers, an upper chamber **2122** and a lower chamber **2124**. A stem **2126** is connected to the piston **2120** and also to the stem **2108** of the poppet valve **2102** (as shown in FIG. **21**, in some embodiments stems **2126**, **2108** are portions of a single stem extending through actuation mechanism **2118** and valve **2102**). The stem **2126** is extended upward into an equalization chamber **2128** in order that the effective area of piston **2120** presented to hydraulic pressure in the upper chamber **2122** may be approximately equal to the effective area of piston **2120** presented to hydraulic pressure in the lower chamber **2124**. Equalization chamber **2128** communicates with a connection **2130** to a body of fluid whose pressure may either be constant or may vary according to the state of operation of assembly **2100**. For example, connection **2130** may communicate with a fluid at atmospheric pressure or a fluid set to (1) a pressure that balances some balance pressure acting on the disc **2104** and stem **2108** of the poppet valve **2102**, or (2) a high pressure in order to assist in accelerating the piston **2120** of the actuation mechanism **2118** downward.

[0329] By means of a connection **2116** to a relatively low-pressure body of fluid (not shown), the pressure within the catchment chamber **2114** is typically near atmospheric pressure and is preferably kept less than or (at most) approximately equal to both (a) the pressure within the poppet valve **2102** and (b) the pressure within the lower chamber **2124** of the actuation cylinder **2118**. Fluids (e.g., gas, hydraulic fluid, heat-transfer liquid) that manage to bypass the gaskets **2110** and **2112** therefore do not pass from the poppet valve **2102** to the actuator cylinder **2118** or vice versa (and hence causing undesired mixing of different bodies of fluids and/or fluid types within assembly **2100** and any system including assembly **2100**), and instead accumulate in the catchment chamber **2114** and may be removed therefrom through the connection **2116**.

[0330] In the illustrative embodiment depicted in FIG. **21**, the areas A_1-A_5 on the lower face of the poppet-valve disc **2104** (A_1), the upper face of the disc **2104** (A_2), the lower face of the actuator piston **2120** (A_3), the upper face of the actuator piston **2120** (A_4), and the upper end of the actuator stem **2126** (A_5, exposed to fluid in chamber **2128**), upon which pressure is exerted by fluid, are chosen to allow for efficient actuation of the poppet valve **2102** through appropriate sizing and balancing of forces acting along the stem **2126** in various states of operation. In general, the lower face of the poppet-valve disc area, A_1, will be equivalent to the valve opening area. For a valve opening diameter of 7.5 cm, the poppet valve disc area would be approximately $A_1=4.42\times10^{-3}$ m^2. The actuator stem area, A_5, is typically large enough to withstand actuation, impact, and other forces, typically 5% to 10% of the poppet-valve disc area, A_1. The area A_2 is equal to the difference between the poppet-valve disc area and the poppet-valve stem area, or A_1-A_5 in the case where the poppet-valve stem area is equal to the actuator stem area. The actuator piston areas, A_3 and A_4, are sized (considering actuator hydraulic fluid pressure) to provide sufficient force, and thus acceleration, of the poppet-valve in order to achieve the desired poppet-valve actuation time; the actuator piston areas are typically on the order to 5% to 15% of the area of poppet-valve disc area, A_1. In the examples shown in FIGS. **27-29**, $A_3=0.095A_1$ and $A_5=0.064A_1$.

[0331] Assembly **2100** features contrivances for closing poppet valve **2102** rapidly while not allowing the disc **2104** to impact the seat **2106** at unacceptably high velocity, as well as for opening poppet valve **2102** rapidly while not allowing actuator piston **2120** to impact the end-cap of upper chamber **2122** at unacceptably high velocity. Below, the state of operation of assembly **2100** depicted in FIG. **21** is described in detail. In FIG. **22**, an assembly **2200** similar to assembly **2100** but featuring a low-side poppet valve is depicted. In FIGS. **23**A-**23**D and **24**A-**24**E, the principles of operation of contrivances for achieving rapid valve operation with controlled impact velocities within assemblies **2100** and **2200**, as well as within various other embodiments, are elucidated.

[0332] Piping **2132** conducts fluid between the lower actuation chamber **2124** and the outflow side of a pressure-relief valve **2134**, which remains closed in ordinary states of operation of the assembly **2100**. Piping **2132** also conducts fluid to and from a master valve **2136**, which governs the direction of operation of the actuator **2118**, i.e., opening (upward motion) or closing (downward motion). In the position of valve **2136** depicted in FIG. **21**, actuator **2118** is closing, and fluid may be directed from lower chamber **2124** through piping **2132** and valve **2136** to a reservoir of low-pressure fluid **2138**.

[0333] Lower chamber **2124** is also connected to piping **2140**, which conducts fluid between lower chamber **2124** and piping **2132** through (a) a variable or fixed flow resistance **2142** and (b) an optional connection **2146**, e.g., a connection to a source of fluid at high pressure, low pressure, or variable pressure. Fluid may flow from piping **2132** to piping **2140** and chamber **2124** via check valve **2144**.

[0334] Piping **2132** communicates with lower actuator chamber **2124** through an opening in the side surface (lateral interior surface) of chamber **2124**. This opening is sized, shaped, and located in the side of chamber **2124** in such a manner that when piston **2120** has reached its downward limit of motion, the opening is occluded, and fluid essentially cannot enter or leave the lower chamber **2124** through piping **2132**. Moreover, the sizing, shaping, and location of the occludable opening in chamber **2124** may be such as to contribute to controlled deceleration of piston **2120** when the piston **2120** approaches the lower end cap of chamber **2124**. The manner in which the character of an occludable opening may contribute to controlled deceleration of piston **2120** will be elucidated in reference to FIGS. **23**A-**23**D and **24**A-**24**E.

[0335] Piping **2140** typically communicates with chamber **2124** through an opening in the end cap (lower surface) of chamber **2124**; thus, fluid may typically enter or leave the lower chamber **2124** regardless of the position of piston **2120**.

[0336] Components of assembly **2100** associated with operation of the upper actuator chamber **2122** are similar to those just described in association with operation of the lower actuator chamber **2124**. For example, piping **2148** conducts fluid between the upper actuation chamber **2122** and the inflow side of the pressure-relief valve **2134**. When the pressure at the inflow side of valve **2134** exceeds that at the outflow side of valve **2134** by some specified opening difference, valve **2134** opens and remains open until the difference in pressure between its two sides drops below some specified closing difference. Relief valve **2134** allows poppet valve **2102** to open without damage to components of assembly **2100** when pressure within the pneumatic cylinder (not shown), i.e., pressure exerted upward on the disc **2104**, exceeds some predetermined threshold. For example, an attempt to compress a relatively incompressible liquid in the upper chamber of the pneumatic cylinder, i.e., a possible hydrolock condition, may produce a pressure difference between the two chambers of the actuator cylinder **2118** sufficient to cause relief valve **2134** to open. Valve **2134** prevents hydrolock in the pneumatic cylinder and other conditions of overpressure in the pneumatic cylinder.

[0337] Piping **2148** also conducts fluid between upper actuator chamber **2122** and the master valve **2136**. In the position of valve **2136** depicted in FIG. **21**, actuator **2118** is closing, and fluid at high pressure may be directed to upper chamber **2122** through check valve **2152**, master valve **2136**, and piping **2148** from a source of high-pressure fluid **2154**. The source of high-pressure fluid **2154** (e.g., a hydraulic pump) may be located distant from valve **2136**, thus, a hydraulic accumulator **2150** may be located closer to valve **2136** in order to maintain high pressure during fluid flow through valve **2136**.

[0338] Upper chamber **2122** is also connected to piping **2156**, which conducts fluid between upper chamber **2122** and piping **2148** through (a) a variable or fixed flow resistance **2160** and (b) an optional connection **2162**, e.g., a connection to a source of fluid at high pressure, low pressure, or variable pressure. Fluid may flow from piping **2148** to piping **2156** and chamber **2122** via check valve **2158** and through flow resistance **2160**.

[0339] Piping **2148** communicates with upper actuator chamber **2122** through an opening (not depicted) in the side surface (lateral interior surface) of chamber **2122**. This opening is sized, shaped, and located in the side of chamber **2122** in such a manner that when piston **2120** has reached its upward limit of motion, the opening is occluded, and fluid essentially cannot enter or leave the upper chamber **2122** through piping **2148**. Moreover, the sizing, shaping, and location of the occludable opening in chamber **2122** may be such as to contribute to controlled deceleration of piston **2120** when the piston **2120** approaches the upper end cap of chamber **2122**. The manner in which the character of an occludable opening may contribute to controlled deceleration of piston **2120** will be elucidated in FIGS. **23**A-**23**D and **24**A-**24**E.

[0340] Piping **2156** communicates with chamber **2122** through a non-occludable opening in the end cap (upper surface) of chamber **2122**; thus, fluid may typically enter or leave the upper chamber **2122** regardless of the position of piston **2120**.

[0341] In the state of operation depicted in FIG. **21**, the high-side poppet valve **2102** is being closed. High-pressure fluid from the source **2154** flows through check valve **2152**, master valve **2136**, and piping **2148** into the upper actuator chamber **2122**. Fluid at high pressure also flows from piping **2148** through check valve **2158** and piping **2156** into the upper chamber **2122**. Simultaneously, fluid at relatively low pressure exits the lower actuator chamber **2124** through piping **2132** and master valve **2136**. Fluid at relatively low pressure also exits the lower chamber **2124** through piping **2140** to piping **2132** through flow resistance **2142**. A net downward force is being applied on the actuator piston **2120**, and the stem **2108** and disc **2104** are moving downward.

[0342] FIG. **22** is a schematic drawing of various components of an illustrative assembly **2200** embodying aspects of the invention. Assembly **2200** differs from assembly **2100** in FIG. **21** primarily in comprising a low-side poppet valve **2202** rather than a high-side poppet valve (e.g., valve **2102** in FIG. **21**). Assembly **2200** also differs from assembly **2100** in the orientation of the relief valve **2234**. That is, the inflow side of valve **2234** is connected to piping **2232** (which corresponds to piping **2132** in FIG. **21**) and the outflow side of valve **2234** is connected to piping **2248** (which corresponds to piping **2148** in FIG. **21**). Relief valve **2234** remains closed in ordinary states of operation of the assembly **2200**. When the pressure at the inflow side of valve **2234** exceeds that at the outflow side of valve **2234** by some specified opening difference, valve **2234** opens and remains open until the difference in pressure between its two sides drops below some specified closing difference. Relief valve **2234** allows poppet valve **2202** to open without damage to components of assembly **2200** when pressure within poppet valve **2202**, i.e., pressure exerted downward on the disc **2204**, exceeds some predetermined threshold.

[0343] In the state of operation depicted in FIG. **22**, opening of the high-side poppet valve **2202** has just commenced. Much as in FIG. **21**, a net downward force is being applied on the actuator piston **2220** and the disc **2204** is moving downward.

[0344] FIG. **23**A is a schematic drawing of various components of an illustrative two-chamber hydraulic activation cylinder assembly **2300** whose components correspond functionally to certain components of assembly **2100** in FIG. **21** and assembly **2200** in FIG. **22**, as shall be made clear hereinbelow. Assembly **2300** includes an activation cylinder **2302**, a hydraulic connection **2304** selectably connected to either a low-pressure fluid reservoir or a high-pressure fluid source, an adjustable flow resistance **2306**, and a check valve **2308**, all interconnected by piping as shown. Activation cylinder **2302** includes a piston **2310** and a stem **2312**. Stem **2312** is connected to the disc of a poppet valve (not shown), which is opened and closed by the assembly **2300** in an arrangement similar to that depicted in FIGS. **20**-**22**. References herein to the movements of the piston **2310** describe the movements in unison of the piston **2310**, stem **2312**, and poppet disc. Piping **2314** connects the connection **2304** to an occludable orifice **2316** in the wall of the proximal chamber **2318** of the cylinder **2302**. The shape of the occludable orifice **2316** is not depicted in FIG. **23**A, but its distal (rightward) and proximal (leftward) limits are indicated by dotted lines A and B. A fixed orifice **2320** allows fluid to enter or leave chamber **2318** whether or not the occludable orifice **2316** is occluded. The orifices **2316**, **2320** and other characteristics of assembly **2300** are chosen so that the piston **2310** decelerates from a high velocity V_{max}, achieved before the piston **2310** reaches position A, to an acceptably low final velocity V_{end}, achieved by the time the piston **2310** reaches position B. Assembly **2300** is shown in a horizontal orientation for illustrative purposes in FIGS. **23**A-**23**D but other orientations (e.g., vertical) may be employed in various embodiments.

[0345] Functionally, cylinder **2302** corresponds to activation cylinder **2118** in FIG. **21**; piston **2310** to piston **2120**; stem **2312** to stem **2108**; chamber **2318** to chamber **2124**; piping **2320** to piping **2140**; flow resistance **2306** to flow resistance **2142**; check valve **2308** to check valve **2144**; piping **2314** to **2132**; and connection **2304** to either high-pressure source **2154** or low-pressure reservoir **2138** (selectable by master valve **2136**).

[0346] In the state of operation depicted in FIG. **23**A, the activation cylinder **2302** is performing a closing stroke, i.e., the piston **2310** and stem **2312** are moving to the left. The piston **2310** has not yet begun to occlude orifice **2316** (i.e., the proximal surface of piston **2310** has not yet reached position A). It is preferable, during this portion of the closing stroke, that the fluid pressure in the proximal chamber **2318** be as low as possible, in order to minimize the work that must be performed to moving the piston **2310**. During this portion of the closing stroke, therefore, the connection **2304** is connected to a low-pressure fluid reservoir and the total orifice area presented to fluid in chamber **2318** is maximal: e.g., fluid undergoes a relatively small pressure drop in flowing through orifice **2316** and piping **2314** to connection **2304**. Some fluid also flows through the adjustable flow resistance **2306** and thence to connection **2304**.

[0347] FIG. **23**B depicts the system **2300** of FIG. **23**A in a state of operation subsequent to that shown in FIG. **23**A. Piston **2310** has occluded all of orifice **2316** and has been decelerated to V_{end}. Fluid has ceased to flow through piping **2314** but continues to flow through the adjustable flow resistance **2306** and thence to connection **2304**. As piston **2310** subsequently reaches and/or overshoots position B (due to, e.g., compression or spring-cushioning of the seat of the activated poppet valve), the pressure in chamber **2318** and therefore the decelerating force acting on piston **2310** is determined by the size of fixed orifice **2320** and by the setting of the adjustable flow resistance **2306**. (It is here assumed, for simplicity, that the piping in system **2300** presents negligible resistance to fluid flow.) If orifice **2320** were not present, when piston **2310** completely occluded orifice **2316** fluid would be unable to exit chamber **2318** and pressure in chamber **2318** would spike to some relatively very high value (i.e., a hydrolock condition would occur). With an appropriately sized orifice **2320** and appropriately adjusted resistance **2306**, at no time in the closing stroke (e.g., even during overshoot of position B) does the pressure in chamber **2318** exceed the design pressure limit P_{max} (i.e., the maximum pressure tolerable without mechanical damage or failure).

[0348] FIG. **23**C depicts the system **2300** of FIG. **23**A in a different state of operation. In the state of operation depicted in FIG. **23**C, the activation cylinder **2302** is commencing an opening stroke, i.e., the piston **2310** and stem **2312** are moving to the right. The piston **2310** has not yet begun to unocclude orifice **2316** (i.e., the proximal surface of the piston **2310** has not yet passed position B). It is preferable, during this portion of the opening stroke, that the fluid pressure in the proximal chamber **2318** be as high as possible (e.g., P_{max}) in order to accelerate the piston **2310** rapidly in the distal direction and so open the poppet valve rapidly, minimizing disc-proximity losses. During this operating state, therefore, the connection **2304** is connected to a fluid source at high pressure (e.g., P_{max}). No fluid can yet flow through piping **2314** to occludable orifice **2316**. However, fluid flows freely through the low-resistance check valve **2308** to fixed orifice **2320**, enabling rapid acceleration of the piston **2310**.

[0349] FIG. **23**D shows the system **2300** of FIG. **23**C in a state of operation subsequent to that depicted in FIG. **23**C. Piston **2310** has passed position A and the occludable orifice **2316** is completely unoccluded. Fluid now passes relatively freely from connection **2304** into chamber **2318** through the check valve **2308** (and orifice **2320**) and piping **2314** (and occludable orifice **2316**).

[0350] The arrangement of orifices, valves, and piping depicted in FIGS. **23**A-**23**D are advantageous because they provide automatically, i.e., without the operation of active valves or other complex or energy-consuming devices, for modulated resistance to fluid flow out of chamber **2318** during states of operation of system **2300** when modulated resistance is desirable, and for relatively low resistance to fluid flow into chamber **2318** during states of operation when low resistance is desirable.

[0351] It will be clear to persons familiar with the principles of hydraulic devices that an arrangement (not depicted) of orifices, valves, and piping similar to that depicted in FIGS. **23**A-**23**D may be connected to the distal chamber **2322** of the cylinder **2302** in order to control the deceleration of the piston **2310** during the latter portion of an opening stroke.

[0352] FIG. **24**A is a schematic drawing of components of an illustrative activation cylinder **2400** that may be part of a larger system, not otherwise depicted, for the storage and release of energy, and that incorporates aspects of certain embodiments of the invention. The cylinder **2400**, which is connected to and activates a poppet valve (not shown) in an arrangement similar to those depicted in FIG. **20** or FIG. **30**A, features arrangements for rapid, controlled deceleration of the piston of the activation cylinder **2400** during valve closure in order to enable rapid, efficient closure of the poppet valve (with lessened throttling losses during the final phase of closure) while avoiding high disc-to-seat impact velocity in the poppet valve.

[0353] The downward direction in FIG. **24**A is also herein termed the proximal direction, and the upward direction in FIG. **24**A is also herein termed the distal direction. The activation cylinder **2400** includes a tubular cylinder body **2402** (not necessarily circular in cross section), a piston **2404**, a proximal end-cap **2406**, a distal end-cap **2408**, an orifice of fixed cross-section (i.e., a fixed orifice) **2410** in the proximal end-cap **2406**, and occludable lateral orifices **2412**, **2414** (e.g., perforations in the wall of the cylinder body **2402**). (The two orifices **2412**, **2414** may alternatively be described as, and may be equivalent to, two portions of a single orifice, i.e., the upper and lower portions of a single orifice.) Herein, the upper orifice **2412** is also termed the "free flow zone" or "fixed orifice" and the lower orifice **2414** is also termed the "cushion zone" or "shaped orifice." In various embodiments, the lower orifice **2414** will tend to be significantly smaller than the upper orifice **2412**.

[0354] The volume between the proximal face of the piston **2404** and the inside surface of the proximal end-cap **2406** constitutes the proximal chamber **2416** of the activation cylinder **2400**. The volume between the distal face of the piston **2404** and the distal end cap **2408** constitutes the distal chamber **2418** of the activation cylinder **2400**. The proximal chamber **2416** and distal chamber **2416** are both filled with a substantially incompressible fluid.

[0355] The activation cylinder **2400** also includes a stem (not shown, for clarity) that connects the proximal portion of the piston **2404** to the disc of a poppet valve (not shown) that is located proximally to and is aligned with cylinder **2400**, in an arrangement similar to that depicted in a horizontal orientation in FIG. **23**. The stem and the poppet-valve disc move in unison with the piston **2404**. Until the piston **2404** (i.e., the proximal surface of piston **2404**) reaches position B, the poppet valve (not depicted) is deemed fully open. When the piston **2404** reaches position D, the poppet valve is closed (i.e., the disc touches the seat; see FIG. **20**). The distance between position B and position D is h_3.

[0356] In the state of operation depicted in FIG. **24**A, the activation valve **2400** is closing the poppet valve; that is, the piston **2404** is moving downward, and so, impelled by the stem, is the disc of the poppet valve (not shown). As piston **2404** moves downward, fluid is expelled from the proximal chamber **2416** through the fixed orifice **2410** and the occludable orifices **2412**, **2414**.

[0357] The upper occludable orifice **2412** is rectangular in the embodiment depicted in FIG. **24**A, with height h_1 and width w. The shaped orifice **2414** has height D_{prox} and a variable width. The trapezoidal form of the shaped orifice **2414** in FIG. **24**A is illustrative only; other forms for the shaped orifice **2414** are contemplated and within the scope of the invention. An exemplary calculation of a form for the shaped orifice **2414** that is optimal under certain assumptions will be provided hereinbelow.

[0358] For ease of illustration, the transverse cross-sectional shape of cylinder body **2402** in FIG. **24**A is presumed to be rectangular; thus, the shapes of the orifices **2412**, **2414** as depicted in FIG. **24**A are undistorted by projection on the page. However, other cross-sectional forms for cylinder body **2402** (e.g., circular) are contemplated and within the scope of the invention.

[0359] Until the proximal or lower surface of the piston **2404** reaches position C (marked by a dotted horizontal line in FIG. **24**A), whereupon the piston **2404** begins to occlude the shaped orifice **2414**, it is desirable that minimal force be required to move the piston **2404**. Minimal force exerted on piston **2404** entails the expenditure of minimal work. The purpose of upper orifice **2412** is to minimize hydraulic resistance to the downward movement of the piston **2404** until position C is reached. Therefore, upper orifice **2412** is generally made as large as is feasible. This implies that orifice **2412** will preferably be made as large as possible while still allowing the piston **2404** to simultaneously completely occlude both orifices **2412**, **2414**. This entails that, given a piston of height h_2 and a shaped orifice **2414** of height D_{prox}, we have $h_1+D_{prox}\leqq h_2$. This restriction on height h_1 of upper orifice **2412** may be restated as $h_1\leqq h_2-D_{prox}$. Likewise, the width w of orifice **2412** is preferably substantially large given the dimensions of cylinder body **2402** to prevent restriction of fluid entering or exiting orifice **2412**. For a cylinder body **2402** having internal circumference c, the width w of orifice **2412** can be no greater than c (i.e., w≦c). In FIG. **24**A, the width w of orifice **2412** is illustratively depicted as substantially equal to the projected diameter of the cylinder body **2402**; however, this is illustrative only and typically orifice **2412** would be circular in cross-section and sized large enough (e.g., only large enough) to prevent restriction of entering or exiting fluid.

[0360] In an ideal case, resistance to the downward movement of piston **2404** would be zero until the proximal surface of piston **2404** reached position C, the top of the shaped orifice, whereupon the piston **2404** would begin to occlude the shaped orifice **2414** and deceleration of the piston **2404** would proceed as shall be described in detail hereinbelow. The provision of the large vent or orifice **2412** is a viable contrivance for minimizing resistance to the downward movement of piston **2404** until the piston **2404** reaches position C and begins to occlude the shaped orifice **2414**. The use of other contrivances to this end is contemplated and within the scope of the invention. For example, in some alternative embodiments (not depicted), shaped orifice **2414** is retained but upper orifice **2412** is omitted. In these alternative embodiments, a channel passes longitudinally through the body of piston **2404**, of sufficient width to allow fluid to pass with minimal (e.g., near-zero) resistance from proximal chamber **2416** to distal chamber **2418** as the piston **2404** moves downward. This internal piston channel may be closed off when the proximal surface of piston **2404** approaches or reaches position C (the top of the shaped orifice): e.g., a butterfly valve or other valve mechanism within the body of the piston **2404** may close off the internal channel; or, a plug cylinder may be positioned within the proximal chamber **2416**, of diameter approximately equal to the internal piston channel and as tall as the shaped orifice **2414**, so that the plug's upper end enters the lower end of the internal piston channel as the proximal surface of piston **2404** passes position C. These and other viable alternative arrangements would allow rapid transition from low resistance to downward movement of piston **2404** to resistance governed by expulsion of fluid from proximal chamber **2416** through the shaped orifice **2414** and fixed orifice **2410**. By contrast, the arrangement depicted in FIG. **24**A entails a substantially linear decrease in total orifice area from the time piston **2404** reaches position A until it reaches position C, with a concomitant increase in decelerating force on piston **2404**.

[0361] In a physical realization of the system of which portions are depicted in FIG. **24**A, some deceleration of piston **2404** due to progressive occlusion of upper orifice **2412** and to throttling losses as the poppet disc (not shown) approaches the seat would occur by the time the proximal surface of the piston **2404** reaches position C (the top of the shaped orifice **2414**.) As explained above, alternative mechanisms could approximate the assumption of negligible resistance to movement of piston **2404** until piston **2404** reaches the top of shaped orifice **2414**. In this discussion, it is presumed that some deceleration of the piston **2404** from its maximum velocity V_{max} to some slightly lesser velocity V'_{max} occurs as the piston passes from position A to position C. At the moment depicted in FIG. **24**A, the piston **2404** is moving at a maximum velocity V_{max} and has not yet reached position A.

[0362] At the moment depicted in FIG. **24**B, the piston **2404** has passed position A. The upper orifice **2412** is partly occluded by the piston **2404**. The disc of the poppet valve (not shown) is still so far from the seat that the poppet valve is effectively fully open (i.e., throttling losses around the poppet disc are still negligible). When the piston **2404** passes position B, functional closure of the poppet valve begins. That is, the poppet valve is no longer presenting minimal resistance to flow; throttling losses begin to be significant and increase until valve closure is complete (i.e., when piston **2404** reaches position D, whereupon the poppet valve disc contacts the seat). Preferred embodiments of the present invention are designed to traverse the distance from position B to position D as quickly as possible, thus minimizing total energy losses from throttling, while (a) maintaining the fluid pressure within chamber **2416** at or below some specified limit P_{max} and (b) having the piston **2404** arrive at position D at an acceptably low final velocity V_{end}. This is also the impact velocity of the activated poppet valve's disc upon the poppet valve's seat.

[0363] At the moment depicted in FIG. **24**C, the piston **2404** has reached position C; that is, it has completely occluded the upper orifice **2412** and has not yet begun to occlude the shaped orifice **2414**. The piston **2404** has decelerated to V'_{max}.

[0364] As piston **2404** moves downward from position C to position D, the fluid within chamber **2416** is pressurized to some pressure P(t) that may vary with time, the volume of proximal chamber **2416** is decreased, and a volumetric flow of incompressible fluid Q(t) that is equal to the decrease in volume of chamber **2416** is expelled from chamber **2416** through the fixed orifice **2410** and shaped orifice **2414**. Herein, it is assumed for simplicity that the fluid exiting chamber **2416** encounters negligible flow resistance (e.g., in piping or valves) other than that presented by passage through the orifices **2410**, **2414**. In this case, the pressure within chamber **2416** is determined entirely by the rate of flow Q(t) (which is determined by the velocity of the piston **2404**) and by the total opening area O(t) of the orifices **2410**, **2414**. Total opening area O(t) is the sum of the areas of the unoccluded portions of the two orifices **2410**, **2414**. Slower motion of the piston **2404** entails lower Q(t), which tends to entail lower P(t); smaller orifice area O(t) tends to entail higher P(t). Since the velocity of piston **2404** tends to decrease as the piston **2404** moves proximally, and the opening area O(t) also tends to decrease as the piston **2404** moves proximally, there tends to be a balance or offsetting between the effects on P(t) of changing velocity and O(t).

[0365] Fluid pressure in chamber **2416** produces an upward-acting force on the proximal face of piston **2404**. If that pressure exceeds the closing force and there no other forces are acting on piston **2404** (as is assumed here for simplicity), piston **2404** will thus decelerate as it moves downward. Also, when piston **2404** passes position C, piston **2404** begins to occlude the shaped orifice **2414**, decreasing the total opening area O(t). Thus, as the piston **2404** moves downward past position C, the piston **2404** will decelerate, tending to entail lower P(t), while the unoccluded area of the orifice **2414** will decrease, tending to entail higher P(t). If the occludable orifice **2414** and fixed orifice **2410** are appropriately sized and shaped, these two effects (piston deceleration and orifice narrowing) will offset each other in such a manner that P(t) retains a constant value, preferably close (or even equal) to the maximum cushioning pressure, $P(t)=P_{max}$, as the piston **2404** decelerates from position C to position D. At position D, the piston **2404** will preferably be moving with velocity V_{end} (i.e., an acceptably slow disc-to-seat impact velocity).

[0366] As previously noted, the orifice **2414** is D_{prox} in height. The transverse profile of the orifice **2414** as depicted in FIG. **24**A (i.e., linear widening as one proceeds in the proximal direction) is illustrative only, and does not necessarily correspond to the shape that orifice **2414** would have in various physical realizations of system **2400** or various other embodiments. In various other embodiments, not depicted, more than one occludable orifice and more than one fixed orifice are employed, where the various orifices may differ from one another in shape and size, and contrivances external to the activation valve (e.g., valves with time-variable flow resistance) may also be employed, alternatively or additionally, to modulate the resistance encountered by fluid exiting chamber **2416** during piston deceleration and thus the relationship between piston velocity and pressure within the chamber **2416** during deceleration. In various other embodiments, contrivances internal to the activation valve may be employed additionally or alternatively to those depicted in FIGS. **24**A-**24**E in order to modulate the resistance encountered by fluid exiting chamber **2416** during piston deceleration: e.g., one or more channels within the body of piston **2404** may permit fluid flow (possibly modulated by valves or other devices) from the proximal surface of the piston **2404** to one or more orifices on the lateral surface of the piston **2404** that may communicate with occludable orifice **2414** in some positions or states of motion of the piston **2404**. All such alternative or additional contrivances for modulating the relationship between piston velocity and pressure within the proximal chamber **2416**, though not depicted, are contemplated and within the scope of the invention.

[0367] FIG. **24**D shows the activation cylinder **2400** of FIG. **24**A in a state of operation subsequent to that depicted in FIG. **24**C. In FIG. **24**D, the piston **2404** has passed position C and is partly occluding the shaped orifice **2414**. The piston **2404** is decelerating at an approximately constant rate A, the rate of fluid outflow Q(t) from chamber **2416** is decreasing, total opening area O(t) is decreasing, and pressure P(t) in chamber **2418** is at a constant P_{max}.

[0368] FIG. **24**E shows the activation cylinder **2400** of FIG. **24**A in a state of operation subsequent to that depicted in FIG. **24**D. In FIG. **24**E, the piston **2404** has reached position D and is entirely occluding the shaped orifice **2414**. The piston **2404** has decelerated to velocity V_{end} and the disc of the poppet valve (not depicted) is just making contact with the seat.

[0369] The shape of occludable orifice **2414** required for optimal closure of the activation cylinder of FIGS. **24**A-**24**E, in accordance with some embodiments of the invention, may be calculated under the following illustrative and idealized assumptions:

[0370] 1) The piston **2404**, the stem (not shown, but see FIG. **4**), and the disc of the poppet valve (not shown, but see FIG. **20**) move in unison, constituting a rigid mechanical unit herein termed the "piston-disc assembly."

[0371] 2) The piston-disc assembly is moving proximally at some maximum feasible velocity V'_{max} when the piston-disc assembly begins to decelerate (i.e., when the piston **2404** reaches point C in FIG. **24**A).

[0372] 3) The changing gravitational potential energy of the piston-disc assembly may be neglected. (Alternatively, activation cylinder **2400** may be operated in a horizontal position, in which case the gravitational potential energy of the piston-disc assembly is constant.)

[0373] 4) The changing gravitational potential energy and momentum of fluid in the distal chamber of the cylinder **2402** may be neglected.

[0374] 5) Turbulence and other complicating fluid-mechanical effects may be neglected.

[0375] 6) Cylinder **2400** is contrived so that the piston-disc assembly decelerates with constant deceleration of magnitude A to a final impact velocity V_{end}. Pressure in the proximal chamber **2412** is a constant P_{max} during deceleration.

[0376] 7) The only force acting on the piston-disc assembly during deceleration is the hydraulic force F_{decel} exerted in the distal direction on the piston **2404** by the fluid in the proximal chamber **2416**: i.e., $F_{decel}=P_{max}S_{pist}$, where S_{pist} is the area of the proximal surface of piston **2404**. F_{decel} is a constant because P_{max} and S_{pist} are both constant by definition. By Newton's Second Law, $A=F_{decel}/M_{PD}$, where M_{PD} is the total mass of the piston-disc assembly. Pressure in the distal chamber **2418** is assumed to be zero during deceleration. (Alternatively, one may assume constant nonzero pressure in the distal chamber **2418** during deceleration, which merely scales the transverse width of the occludable shaped orifice **2414** by a constant factor.)

[0377] 7) The square root of the pressure P(t) within the proximal chamber **2416** is proportional to the time-varying volumetric flow Q(t) of fluid out of the chamber **2416** through the orifices **2410**, **2414** divided by the total area O(t) of the orifices: $P(t)^{1/2} \propto KQ(t)/O(t)$, where K is some constant. This is a simplification of the relationship that would actually hold between P(t), Q(t), and O(t) in non-ideal valve. During deceleration, since $P(t)=P_{max}$, it follows that $P_{max}^{1/2} \propto KQ(t)/O(t)$.

[0378] 8) Below position C, the proximal chamber includes a single occludable orifice **2414** in the side-wall of the chamber **2416** and a single fixed orifice **2410** in the proximal end-cap **2406** of the chamber **2416**.

[0379] 9) The height of the occludable orifice **2414** is D_{prox}. Its distal end is at position C and its proximal end is at position D.

[0380] 10) The portion of the wall of cylinder **2402** that is perforated by the occludable orifice **2414** is planar (flat) or so nearly planar that its non-planarity may be neglected.

[0381] As will be clear to persons familiar with the calculus, the foregoing ten conditions allow the calculation of a unique solution for the transverse profile of the occludable orifice **2414**. Labeling the vertical direction in FIGS. **24**A-**24**E as x, where x equals 0 at position C and increases in the proximal direction, the transverse profile y(x) of one side of the orifice **2414** is described by the function $y(x)=C(V'^{2}_{max}-2Ax)^{1/2}$, where C is a constant. Analysis based on the foregoing ten assumptions shows that if a final velocity V_{end} of 0 is specified, y(x) goes to infinity at $x=D_{prox}$ that is, the solution is nonphysical (i.e., orifice **2414** cannot become infinitely wide in a real cylinder). However, the solution is physical if $V_{end}>0$ (i.e., the width of orifice **2414** is finite everywhere).

[0382] Under assumptions differing from those listed above, including more realistic fluid-mechanical assumptions, the optimal shape of the occludable orifice **2414** will differ from that specified by y(x). Additionally, the foregoing result is not altered by assuming that $V'_{max}=V_{max}$ (i.e., no deceleration of the piston prior to reaching position C).

[0383] The height of the occludable orifice **2414** is not limited to D_{prox} in all embodiments; some portion of the occludable orifice may, for example, extend all the way to the proximal end cap **2406** of the cylinder **2400**. The shape of any such extended occludable orifice **2414** may be adjusted to tune pressure and flow within the proximal chamber **2416** during any overshoot of position D during a closing stroke (e.g., during compression of the seat **2020** in FIG. **20**, or of a helical spring supporting the seat **2020**, after impact of the disc **2018**).

[0384] FIG. **24**F shows the activation cylinder **2400** of FIGS. **24**A-**24**E with the illustrative trapezoidal orifice **2414** of FIGS. **24**A-**24**E replaced by an occludable orifice **2420** having a transverse profile (for each side of the symmetrical orifice) described by the function $y(x)=C(V_{max}^2-2Ax)^{1/2}$. As shown hereinabove, this profile is optimal under certain assumptions. A symmetrical orifice **2420** is preferred in order to eliminate unbalanced transverse forces due to asymmetric fluid flow through the orifice **2420**. The scale of all aspects of FIG. **24**F is arbitrary and illustrative only, including the scaling of the transverse profile of the orifice **2420**.

[0385] The benefits of an idealized system (e.g., that in FIG. **24**F) may be realized to some degree by systems not possessing optimally shaped occludable orifices. FIG. **24**F is a drawing of an occludable orifice in the wall of the proximal chamber **2416** in one realization of the activation cylinder **2400**. This orifice form has the advantage of being manufacturable by two simple drillings.

[0386] FIG. **25** is a drawing of portions of one illustrative realization of aspects of the invention. Specifically, FIG. **25** shows the outline of two orifices **2502**, **2504** (which may also be considered as two portions of a single orifice). Functionally, orifice **2502** corresponds to the free-flow zone or upper orifice **2412** of FIGS. **24**A-**24**F and orifice **2504** corresponds to the cushion zones, lower orifices, or shaped orifices **2414**, **2420** of FIGS. **24**A-**24**F. The outlines of upper orifice **2502** and lower orifice **2504** correspond approximately to the non-straight portion of a circle segment perimeter: each may be manufactured as a circular drilling through the wall of the lower actuator chamber **2124** in FIG. **21**A. The circle whose perimeter corresponds in part to the outline of upper orifice **2502** has a radius of approximately 3.2 mm; the circle whose perimeter corresponds in part to the outline of lower orifice **2504** has a radius of approximately 1 mm. The lowermost edge of the lower orifice **2504** is approximately 0.75 mm from the inner surface of the lower end cap **2506** of the actuator cylinder (not otherwise depicted in FIG. **25**). In general, the lower orifice **2504** is substantially smaller than the upper orifice **2502**; the area of orifice **2504** is, in some embodiments, 5% to 15% of the area of the upper orifice **2502**. The upper orifice **2504** is sized to provide nearly free-flow, i.e., flow with a low pressure drop (e.g., <20% of actuation pressure), of the exiting fluid during valve actuation. The orifices **2502**, **2504** of FIG. **25** exemplify the non-uniqueness of the free-flow-zone and cushion-zone orifice shapes depicted herein, e.g., in FIGS. **24**A-**24**F.

[0387] FIGS. **26**A and **26**B are drawings of portions of one illustrative realization of aspects of the invention. Assembly **2600** includes a high-side poppet valve **2602** (corresponding to poppet valve **2102** in FIG. **21**) and actuator piston **2618** (corresponding to actuator **2118** in FIG. **21**). For clarity, the actuator **2618** has also been represented in FIG. **26**B in an expanded view, as indicated by thick dashed lines. Chamber **2628** corresponds to equalization chamber **2128** in FIG. **21**, upper actuator chamber **2622** to chamber **2122**, piston **2620** to piston **2120**, stem **2626** to stem **2126**, catchment chamber **2614** to catchment chamber **2114**, piping **2640** to piping **2140**, piping **2632** to piping **2132**, piping **2648** to piping **2148**, and piping **2656** to piping **2156**. Pipings or channels **2640**, **2632**, **2648**, and **2656** are only partially represented in FIG. **26**B. For clarity, as indicated by thin dashed lines, the cross-section of piping **2632** has been represented in FIG. **26**B in an expanded view **2660** and the cross-section of piping **2648** has been represented in FIG. **26**B in an expanded view **2662**. The cross-sections of pipings **2632** and **2648** correspond, in this exemplary realization, to the cross-section of the combined orifices **2502**, **2504** in FIG. **25**. Thus, the realization partly depicted in FIGS. **26**A and **26**B incorporates aspects of the invention to allow for rapid, cushioned opening and closure of the poppet valve **2602**. In FIG. **26**B, the actuator **2618** at its lower limit of motion and the poppet valve **2602** is closed.

[0388] FIG. **27** is a plot of data acquired from a physical realization of aspects of the invention closely resembling that partly depicted in FIGS. **21**, **26**A, and **26**B. FIG. **27** plots the position of the actuator piston **2620** in FIG. **26**B as a function of time (solid line), time averaged for a closing actuation. Also plotted in FIG. **27** is the position-vs.-time function for the assembly **2600** in FIG. **26**B as predicted using the software tool Simscape™ (dashed line). Simulated and observed piston velocity closely agree.

[0389] FIG. **28** is a plot of data acquired from a physical realization of aspects of the invention closely resembling that partly depicted in FIGS. **21**, **26**A, and **26**B. FIG. **28** plots the velocity of the actuator piston **2620** in FIG. **26**B as a function of time (solid line) for a single closing actuation. Also plotted in FIG. **27** is the position-vs.-time function for the assembly **2600** in FIG. **26**B as predicted using the software tool Simscape (dashed line). Simulated and observed piston velocity agree except for a "bounce" in the actual device before closure. Impact velocities of simulation and measurement closely agree at ~0.2 msec.

[0390] FIG. **29** is a plot of data acquired from a physical realization of aspects of the invention closely resembling that partly depicted in FIGS. **21**, **26**A, and **26**B. FIG. **29** plots the fluid pressure within the lower chamber of the actuator piston **2620** in FIG. **26**B as a function of time (solid line) for a single closing actuation. Also plotted in FIG. **27** is the pressure-vs.-time function for lower actuator chamber of the assembly **2600** in FIG. **26**B as predicted using the software tool Simscape (dashed line). Notably, pressure peaks during the latter portion of the closing interval, e.g., during deceleration of the piston-disc assembly.

[0391] FIG. **30**A is a schematic cross-sectional drawing of an illustrative poppet valve **3000**, in accordance with various embodiments of the present invention, that employs a hydraulic or other type of actuation mechanism (not shown) to open and close a port **3002** by moving a disc **3004**. The valve **3000** shown in FIG. **30**A is a high-side valve. Valve **3000** is depicted in a vertical orientation for illustrative purposes; other orientations (e.g., horizontal) may be employed in various embodiments.

[0392] The valve **3000** includes a beveled contact ring **3006** of a suitable material (e.g., polyether ether ketone [PEEK]), recessed into an annular groove or channel in a cylinder end-cap **3008** (shown in part). In FIG. **30**A, the beveling of the disc **3004** complements that of the contact ring **3006**, so that when the disc **3004** is in contact with the ring **3006**, the beveled surfaces of the disc **3004** and ring **3006** are in flush contact with each other, completely occluding port the **3002**.

[0393] The contact ring **3006** rests upon an annular (ring-shaped) wave spring **3010**. The upper surface of the wave spring presses against the contact ring **3006**, and the lower surface of the wave spring presses against the lower surface of the annular groove in the end cap **3008** into which the ring **3006** is recessed. To clarify the schematic cross-sectional view of the wave spring **3010** in FIG. **30**A, an illustrative wave spring is shown in FIG. **30**B. Wave springs of designs other than that shown in FIG. **30**B may be employed in the mechanism of FIG. **30**A. Non-wave-spring mechanisms (e.g., coil spring, pneumatic spring) may be employed, additionally or alternatively, in the mechanism of FIG. **30**A or in other valves embodying the present invention; the use of a wave spring in FIG. **30**A is illustrative only. Fluid flow between the contact ring **3006** and the end cap **3008** is prevented by one or more gaskets **3012**.

[0394] The vertical distance between the lower surface of the valve disc **3004** and the contact ring **3006** is herein termed the disc displacement h and is indicated by a two-headed arrow with letter h in FIG. **30**A. In FIG. **30**A, a vertical coordinate of 0 (zero) is defined by the plane of the lower surface **3014** of the disc **3004** when the disc **3004** is in contact with the ring **3006** and the ring **3006** is in the most distal (i.e., upward, in FIG. **30**A) position it can attain. Herein, this most-distal position of the ring **3006** is termed the neutral position of the ring **3006**. Proximal (i.e., downward, in FIG. **30**A) displacements from the h=0 plane are denoted by negative numbers. Herein, wave spring **3010** is said to be "neutrally compressed" when the ring **3006** is in its neutral position. The lower surface **3014** of the illustrative disc **3004** has radius R. The port **3002** also has radius R.

[0395] FIG. **30**A depicts valve **3000** in a state of operation that occurs during closure of the valve **3000**. In phases of valve closure prior to that depicted in FIG. **30**A, the disc **3004**, originally at rest at full-open displacement h_{FO} (substantially greater than the sufficiently-open distance h_{SO}), has accelerated proximally to maximum closing velocity V_{MC}. Maximum closing velocity V_{MC} was attained by disc **3004** before disc displacement h was reduced to the sufficiently-open distance h_{SO}. Effectual valve closure—i.e., significant reduction of capacity for flow through valve **3000**—may be said to begin at the moment (not depicted) that valve displacement h becomes less than h_{SO}.

[0396] In the state of operation depicted in FIG. **30**A, the wave spring **3010** is in its state of neutral compression, the ring **3006** is in its neutral position, and the disc **3004** is moving downward at the maximum closing velocity (V_{MC}) attained during operation of the valve mechanism. (A constant V_{MC} is assumed illustratively in discussion of FIG. **30**A and elsewhere herein, but non-constant disc velocities may occur in other embodiments of the invention.) At the moment depicted in FIG. **30**A, h is less than h_{SO}. Thus, at the moment depicted in FIG. **30**A, valve **3000** is no longer sufficiently open, but is in the process of closing.

[0397] FIG. **30**C depicts valve **3000** in later phase of closure of valve **3000**. The disc **3004** is still moving downward at velocity V_{MC}. The disc **3004** has made contact with the ring **3006**, occluding the port **3002**, but the ring **3006** is still in its neutral position and the wave spring **3010** is still in its state of neutral compression. FIG. **30**C thus depicts valve **3000** at the instant when sufficient closure is attained. The lower surface **3014** of the disc **3004** is at h=0.

[0398] Subsequent to the moment of sufficient closure depicted in FIG. **30**C, the ring **3006** moves downward from its neutral position, impelled by the momentum of disc **3004**, stem **3016**, and possibly other components connected thereto (herein collectively referred to as the "traveling mass of closure"), as well as by any net downward fluid pressure acting on disc **3004** or forces exerted on the stem **3016** by the actuation mechanism (not shown). Downward displacement of the ring **3006** compresses the wave spring **3010**, which exerts an upward, decelerating force on the disc **3004** and thus the whole traveling mass of closure. The spring constant and dimensions of the wave spring **3010** are chosen so that the velocity of the traveling mass of closure is reduced to an acceptably low final closing velocity V_{CV} (e.g., 0) by the time the spring **3010** has been maximally compressed.

[0399] FIG. **30**D depicts the valve **3000** of FIGS. **30**A and **30**C at the moment when the wave spring **3010** has been maximally compressed and the velocity of the traveling mass of closure has been reduced to V_{CV}. At the moment depicted in FIG. **30**D, the ring **3006** has been displaced by $-h_{SD}$ from its neutral position to its substantially-depressed position.

[0400] Subsequently to the moment depicted in FIG. **30**D, the spring **3010** will tend to restore ring **3006** to its neutral position. Herein, the time interval between the moment depicted in FIG. **30**D and the stable restoration of ring **3006** to its neutral position is termed the settling interval. Any oscillations or other motions of the ring **3006**, disc **3004**, and other components of valve **3000** that may occur during the settling interval depend on the details of construction of valve **3000** and/or other embodiments, and are not discussed further herein. Valve **3000** and other embodiments may be designed so that from the moment of sufficient closure depicted in FIG. **30**C and throughout the settling interval, the disc **3004** and ring **3006** remain in flush contact with each other (i.e., the valve does not bounce, but remains closed from the moment of sufficient closure until the commencement of an opening cycle).

[0401] In some embodiments, sufficient downward force is maintained upon the disc **3004** after the ring **3006** achieves its substantially depressed position at $-h_{SD}$ (as depicted in FIG. **30**D) to maintain the ring **3006** in its substantially-depressed position. In such embodiments, the disc **3004** and disc **3006** may be kept in substantially-depressed position until opening of the valve **3000** is initiated at some later time. In embodiments where the disc **3004** and ring **3006** are maintained in substantially depressed position after closure, work performed upon the spring **3010** during deceleration of the traveling mass of closure is stored in the spring **3010** as elastic potential energy and is available for acceleration of the ring **3006**, disc **3004**, and possibly other components during the early phases of an opening cycle.

[0402] The components and openings depicted in FIG. **30**A and FIGS. **30**C-**30**D are circular in cross-axial cross-section; however, other cross-sectional shapes are contemplated and within the scope of the invention.

[0403] The closure of valve **3000**, as described hereinabove and partly depicted in FIG. **30**A and FIGS. **30**C-**30**D, is more rapid and efficient than closure of an otherwise similar poppet valve (herein termed a "conventional valve") in which (a) during closure, the valve disc begins at rest at an approximate displacement of h_{SO}, (b) downward acceleration of the valve and stem by the actuation mechanism is approximately equal to that of disc **3004** in valve **3000**, and (c) the maximum velocity of closure V_{MC} is approximately the same as that of valve **3000**.

[0404] In embodiments where the disc **3004** and ring **3006** remain in a fully depressed position after closure, FIG. **30**D also depicts the state of valve **3000** at the initiation of an opening cycle. In this state, the spring **3010** is exerting an upward force on the disc **3004**, stem **3016**, and any components attached thereto (herein termed the "traveling mass of opening"). At a moment early in the opening cycle (e.g., the moment depicted in FIG. **30**D or shortly thereafter), the downward force that has held the disc **3004** and ring **3006** in fully depressed position during the closed state of valve **3000** is reversed or significantly reduced. Thereupon, ring **3006** is accelerated upward by the spring **3010**. If an upward force is exerted on the disc **3004** by the actuation mechanism during this period of upward acceleration, and if that upward force is not large enough to accelerate the disc **3004** faster than the spring **3010** accelerates, the ring **3006** and disc **3004** will remain in contact with each other during this interval of upward acceleration, i.e., the valve **3000** will remain sufficiently closed during this interval of upward acceleration. During upward acceleration, the spring **3010** performs work upon the ring **3006** and disc **3004**, restoring to the disc **3006** in the form of kinetic energy a portion of the energy stored in the spring **3010**, during the deceleration phase of the previous valve closure, as elastic potential energy. Thus, some energy typically dissipated in the actuation mechanism during closure of a conventional poppet valve is restored during valve opening in these and other embodiments of the invention, reducing valve actuation energy and increasing overall system efficiency.

[0405] Reference is now made to FIGS. **31**A and **31**B, in which the rapidity of effective closure of valve **3000** as compared to closure of a conventional valve is made clear. The plot in FIG. **31**A, "Conventional Valve," is an illustrative, schematic plot of the position over time of the disc of a conventional valve during closure. The plot in FIG. **31**B, "Valve **3000**," is an illustrative, schematic plot of the position over time of the disc **3004** of valve **3000** during closure.

[0406] In FIG. **31**A, valve closure commences at time T_1. Disc displacement h is equal to h_{SO} (e.g., approximately equal to R/2) prior to time T_1: that is, the disc is stationary at $h=h_{SO}$. Over interval A_1, the disc is accelerated downward until it reaches its maximum closure velocity V_{MC}. At time T_2, deceleration of the disc begins. Deceleration of the disc to an acceptable final closing velocity V_{CV} occurs over interval A_2. At time T_3, the disc and seat of the conventional valve are in contact with each other and the conventional valve is sufficiently closed. The interval from time T_1 (after which time the valve ceases to be sufficiently open) to time T_3 (at which time the valve achieves sufficient closure) is the effective closing time C_1 of the conventional valve.

[0407] In FIG. **31**B, valve closure commences at time T_0. Disc displacement h is equal to h_{FO} (full-open position) prior to T_0: that is, the disc **3004** is stationary at $h=h_{FO}$. Over interval A_3 (approximately equal in duration to interval A_1 in FIG. **31**A), the disc **3004** is accelerated downward until it reaches its maximum closure velocity V_{MC}. At time T_1, the disc has reached the maximum closure velocity V_{MC}. At time T_4, the disc **3004**, still moving at V_{MC}, makes contact with the ring **3006** and the valve **3000** is sufficiently closed. Deceleration of the disc to an acceptable final closing velocity V_{CV} occurs over interval A_4 (approximately equal in duration to interval A_2 in FIG. **31**A). By time T_5, the disc has been decelerated to V_{CV}. The interval from time T_1 (after which the valve ceases to be sufficiently open) to time T_4 (when the valve achieves sufficient closure) is the effective closing time C_2 of valve **3000**. FIGS. **31**A and **31**B make clear that the effective closing time C_2 of valve **3000** is less than the effective closing time C_1 of the conventional valve.

[0408] Moreover, it is apparent from FIGS. **31**A and **31**B that, because the disc **3004** of valve **3000** moves at higher average velocity between the point of sufficient openness (e.g., h approximately equal to R/2) and the point of sufficient closure (h=0), the curtain area $A_{curtain}$ of valve **3000** is restricted (e.g., $0<A_{curtain}<\pi R^2$) for a shorter period of time during closure of valve **3000** than during closure of the conventional valve. Restriction of $A_{curtain}$ entails heightened throttling losses (i.e., dissipation of pressure potential energy as heat in turbulent fluid) as fluid passes through the valve opening **3002**. Therefore, shortening the time interval during which heightened throttling losses occur (i.e., shortening the effective closure time), as the design of valve **3000** does, increases the overall efficiency of the energy conversion system that includes valve **3000**.

[0409] Reference is now made to FIG. **32**A, which depicts a state of the valve **3000** in FIG. **30**D. (Part numbers in FIGS. **32**A-**32**C correspond in their last two digits to numbers of identical parts in FIG. **30**A and FIGS. **30**C-**30**D.) The state depicted in FIG. **32**A occurs during an opening cycle that begins with the ring **3206** in fully depressed position (as depicted in FIG. **30**D) and continues with upward acceleration of the ring **3206** and disc **3204**. Valve **3200** is arranged so that by the time the ring **3206** has reached its neutral position (i.e., can travel no farther upward), and the disc **3204** is at displacement h=0, the disc **3204** has attained its maximum opening velocity, V_{MO}. FIG. **32**A depicts the moment at which ring **3206** reaches its neutral position, and the disc **3204** is at displacement h=0 and is traveling at V_{MO}. Up to this moment, the valve **3200** has remained sufficiently closed during the opening cycle (i.e., the disc **3204** and ring **3206** have remained in flush contact). After the disc **3204** ceases to be in contact with the ring **3206**, the valve **3200** is no longer sufficiently closed; however, it is not sufficiently open until the displacement h of the disc **3204** is approximately equal to or greater than the sufficiently open distance h_{SO}.

[0410] FIG. **32**B depicts a moment in the opening stroke subsequent to that depicted in FIG. **32**A. The displacement of the disc **3204** is approximately h_{SO} and the velocity of the disc **3204** remains V_{MO}, as in FIG. **32**A. At or after the moment depicted in FIG. **32**B, deceleration of the disc **3204** begins. By the time the disc **3204** has reached the full-open displacement h_{FO}, the disc **3204** will have been decelerated to an acceptable final opening velocity, V_{OV} (e.g., 0).

[0411] FIG. **32**C depicts the final position, at the end of an opening stroke, of selected components of valve **3200**. The disc is at rest at the full-open displacement h_{FO}.

[0412] The advantages of an opening cycle of the illustrative embodiment partially depicted in FIGS. **32**A-**32**C, as compared to the opening cycle of a conventional poppet valve (i.e., a poppet valve constructed according to the prior art), are comparable to those described hereinabove for a closing cycle of valve **3200** as compared to the closing cycle of a conventional poppet valve, as partially depicted in FIG. **30**A and FIGS. **30**C-**30**D. That is, accelerating the disc **3200** to its maximum transit speed (i.e., maximum opening velocity V_{MO}) before breaking contact between the disc and ring, and decelerating the disc **3200** after passing the displacement h_{SO} at which the valve becomes sufficiently open, shortens effective valve-opening time as compared to an otherwise similar conventional valve and decreases throttling losses during opening. The plots of FIGS. **31**A and **31**B, with their time axes reversed, would approximately represent the sequence of events for opening valve **3200** as compared to the sequence of events for opening a conventional valve.

[0413] In other embodiments, an opening cycle of valve assembly **3200** begins at a point comparable to the state depicted in FIG. **32**A (i.e., with the ring **3206** in its neutral position), except that the disc **3204** is at rest. In such embodiments, restoration of kinetic energy from the closure phase by the spring **3210** to the disc **3204** does not occur. However, the advantages already described for valve **3200** still accrue to such embodiments: i.e., the effective closure time of valve **3200** is shorter, and throttling losses are lower, than for an otherwise similar conventional valve.

[0414] In yet other embodiments, disc **3204** and ring **3206** may be moved from the neutral position to the fully depressed position in the early phase of an opening cycle, before commencement of upward acceleration.

[0415] FIGS. **33**A-**33**C refer to embodiments of a high-side valve. It will be apparent to persons reasonably familiar with the mechanics of valves that similar arrangements, realizing similar advantages, may be contrived for low-side valves. Such arrangements are contemplated and within the scope of the invention.

[0416] FIG. **33**A is a schematic drawing of components of an illustrative hydraulic actuation assembly **3300**, in accordance with various embodiments of the present invention. The actuator employs a hydraulic cylinder **3302** to open and close the port (not shown) of a poppet valve (e.g., that depicted in FIG. **20** or FIG. **30**A) by moving a disc **3304**. As depicted in FIG. **33**A, the hydraulic cylinder **3302** contains a piston **3306** that divides the interior of the cylinder **3302** into two chambers **3308**, **3310**, both of which are typically filled with an approximately incompressible liquid (herein termed "hydraulic fluid" or simply "fluid"). The piston **3306** is connected to a stem **3312** that passes out of the cylinder **3302** and into the body of the poppet valve (not shown) that is actuated by assembly **3300**. The disc **3304** is comparable to that depicted in FIG. **20** and other figures discussed above. In an alternative embodiment, stem **3312** may extend out of the cylinder **3302** to maintain substantially equal piston areas in chambers **3308** and **3310**.

[0417] Assembly **3300** features a three-way directional control valve (DCV) **3314** having two output ports (A, B) and two input ports (C, D). The three possible settings of DCV **3314** are as follows: (1) Closure setting, in which port C is connected to (i.e., placed in fluid communication with) port A and port D is connected to port B, (2) Deceleration setting, in which port A is connected to port B and ports C and D are closed off, and (3) Opening setting, in which port D is connected to port A and port C is connected to port B.

[0418] Assembly **3300** also features a high-pressure fluid accumulator **3316**, a lower-pressure fluid accumulator **3318**, a low-pressure tank or fluid reservoir **3320**, check valves **3322**, **3324**, **3326**, **3328**, **3330**, a pressure relief valve **3332**, and a pump **3334** that produces fluid at a relatively high pressure (e.g., 3000 psig). Pipes enable various of the components of assembly **3300** to exchange hydraulic fluid. As shall be made clear below, the arrangements of assembly **3300** enable the storage of energy from the deceleration of the piston **3306**, stem **3312**, and disc **3304** during opening or closing of the poppet valve (not shown) by assembly **3300**, and the application of a portion of that stored energy to the acceleration of the piston **3306**, stem **3312**, and disc **3304** during the next closing or opening cycle. Such recuperation or regeneration of actuation energy which would typically (e.g., in a conventional valve actuation mechanism) be dissipated increases the overall efficiency of the energy conversion system including actuation assembly **3300**.

[0419] In the state of operation of assembly **3300** depicted in FIG. **33**A, DCV **3314** is in the Closure setting. Hydraulic fluid at a baseline high pressure p_1 from the output of pump **3334** passes through check valve **3330**, into port D of DCV **3314**, out port B of DCV **3314**, through piping **3336**, and into chamber **3310** of the cylinder **3302**. The fluid in chamber **3310** will exert a force on piston **3306**, pushing the piston **3306**, stem **3312**, and disc **3304** to the left (i.e., toward closure).

[0420] Fluid at p_1 may also pass through piping **3338** into the high-pressure accumulator **3316** (e.g., if the pressure in the high-pressure accumulator **3316** is at a pressure p_{1-} slightly lower than p_1). The pressure of the fluid within the high-pressure accumulator typically never falls significantly below p_1, since fluid at pressure p_1 may always pass through check valve **3330** into the high-pressure accumulator **3316**. Alternatively, if the pressure of the fluid within the high-pressure accumulator **3316** is at a pressure p_{1+} higher than p_1, some of the fluid within the high-pressure accumulator will also pass through DCV **3314** into chamber **3310** of the cylinder **3302**, contributing to the force accelerating the piston **3306**, stem **3312**, and disc **3304** to the left.

[0421] As the piston **3306** moves leftward, fluid in chamber **3308** exits chamber **3308** through piping **3340**, passes through ports A and C of the DCV **3314**, and is conveyed by piping **3342** to the low-pressure accumulator **3318**. If the pressure in the low-pressure accumulator **3318** and/or cylinder chamber **3308** exceeds a predetermined threshold, fluid from the low-pressure accumulator **3318** and/or cylinder chamber **3308** is released to the low-pressure reservoir **3320** (via pressure relief valve **3332**) until the pressure in the low-pressure accumulator **3318** and/or cylinder chamber **3308** no longer exceeds the threshold.

[0422] FIG. **33**B depicts assembly **3300** in a state of operation later in the closure stroke of actuation cylinder **3302**. At a certain fraction of the stroke length of actuation cylinder **3302** (e.g., 80% of stroke length), the DCV **3314** is moved to the Deceleration position, in which chamber **3310** is placed in fluid communication with chamber **3308** through piping **3336**, DCV **3314**, and piping **3340**. This state of operation is depicted in FIG. **33**B. In this state of operation, fluid from chamber **3310** (initially approximately at pressure p_1) tends to move through the restricted passage offered by piping **3336**, DCV **3314**, and piping **3340** to chamber **3308**. The pressure in chamber **3310** thus tends to fall and the pressure in chamber **3308** tends to rise. Moreover, the momentum of the piston **3306**, stem **3312**, and disc **3304** (moving leftward at the maximum closure velocity V_{MC} of the valve) tends to exert force on the fluid in chamber **3308**, raising the pressure of the fluid in chamber **3308** to some peak value. The peak pressure in chamber **3310**, chamber **3308**, and piping connected thereto will depend in part on the total mass of the piston **3306**, stem **3312**, and disc **3304**, as well as V_{MC}. Typically, the peak pressure in peak pressure in chamber **3310**, chamber **3308**, and piping connected thereto, including pipe **3340**, is a pressure p_{1+} higher than pressure p_1. When fluid in pipe **3340** is at p_{1+} and the pressure in the high-pressure accumulator **3316** is less than p_{1+}, check valve **3328**

permits the passage of some of the fluid in pipe **3340** into the high-pressure accumulator **3316**, raising the pressure of the fluid within the high-pressure accumulator **3316**. Thus, in effect, placing DCV **3314** in Deceleration position causes some of the kinetic energy imparted to the piston **3306**, stem **3312**, and disc **3304** during valve closure to be recovered and stored as pressure potential energy in the high-pressure accumulator **3316**.

[0423] Assembly **3300** may be operated (e.g., by placing the DCV **3314** in Deceleration position at an appropriate point in a closing stroke of the actuation cylinder **3302**) so that by the end of the stroke, the piston **3306**, stem **3312**, and disc **3304** are moving at an acceptably low final closing velocity V_{CV}.

[0424] Similarly, to initiate an opening stroke of the actuation cylinder **3302**, the DCV **3314** is placed in Opening position. FIG. **33**C depicts assembly **3300** in a state of operation early in an opening stroke of actuation cylinder **3302**. In the state of operation of assembly **3300** depicted in FIG. **33**C, DCV **3314** is in the Opening setting. Hydraulic fluid at a baseline high pressure p_1 from the output of pump **3334** passes through check valve **3330**, into port D of DCV **3314**, out port A of DCV **3314**, through piping **3340**, and into chamber **3308** of the cylinder **3302**. The fluid in chamber **3308** exerts a force on piston **3306**, pushing the piston **3306**, stem **3312**, and disc **3304** to the right (i.e., opening the valve).

[0425] As during a closure stroke, fluid at p_1 may also pass through piping **3338** into the high-pressure accumulator **3316**. Alternatively, if the pressure of the fluid within the high-pressure accumulator **3316** is at a pressure p_{1+} higher than p_1 (e.g., as a result of the storage in high-pressure accumulator **3316** of pressure potential energy collected during the deceleration phase of a closure stroke), some of the fluid within the high-pressure accumulator **3316** will also pass into chamber **3308** of the cylinder **3302** when the DCV **3314** is first placed in Opening position, contributing to the force accelerating the piston **3306**, stem **3312**, and disc **3304** to the right.

[0426] As the piston **3306** moves rightward, fluid in chamber **3310** exits chamber **3310** through piping **3336**, passes through ports B and C of the DCV **3314**, and is conveyed by piping **3342** to the low-pressure accumulator **3318**.

[0427] At a certain fraction of the stroke length of actuation cylinder **3302** (e.g., 80% of stroke length), the DCV **3314** is moved to the Deceleration position, in which chamber **3310** is placed in fluid communication with chamber **3308** through piping **3336**, DCV **3314**, and piping **3340**. Deceleration of the piston **3306**, stem **3312**, and disc **3304** occurs as described above for deceleration during a closing stroke, with the roles of chamber **3310** and chamber **3308** reversed (i.e., during open-stroke deceleration, pressure drops in chamber **3308** and rises in chamber **3310**). Similarly to deceleration during a closing stroke, deceleration during an opening stroke causes some of the kinetic energy imparted to the piston **3306**, stem **3312**, and disc **3304** during valve opening to be recovered and stored as pressure potential energy in the high-pressure accumulator **3316**.

[0428] Thus, the illustrative embodiment depicted in FIGS. **33**A-**33**C permits the storage and re-use of a portion of the energy required to operate the actuation assembly **3300** during valve opening or closure. Consequently, an energy conversion system featuring poppet-valve actuators similar to assembly **3300** may operate at higher overall efficiency than an energy conversion system featuring conventional poppet-valve actuators.

[0429] FIGS. **33**A-**33**C refer to embodiments of a high-side valve. It will be apparent to persons reasonably familiar with the science of hydraulics that similar arrangements, realizing similar advantages, can be contrived for low-side valves. Such arrangements are contemplated and within the scope of the invention. Moreover, the horizontal orientation of the actuation cylinder **3302** in FIGS. **33**A-**33**C is illustrative only; other orientations (e.g., vertical) are contemplated and within the scope of the invention.

[0430] FIG. **34**A is a schematic cross-sectional drawing of major components of an illustrative electromagnetic valve **3400** in accordance with embodiments of the present invention. For clarity, components of the valve **3400**, including an outside port and the walls of the valve body, are not depicted in FIG. **34**A.

[0431] Valve **3400** may be actuated by differential pressure and by electromagnetic force. In other embodiments, the valve **3400** may be actuated by differential pressure, electromagnetic forces, and mechanical forces in various states of operation. The valve **3400** may include a mechanical or pneumatic spring (not shown) to bias the valve towards closing, cushion opening forces, and/or replace or supplement the closing actuation mechanism. The valve **3400** shown in FIG. **34**A is a high-side valve, as defined above, and in various embodiments may be substituted for the poppet-style high-side valves depicted in previously described figures, thereby realizing various additional advantages.

[0432] Valve **3400** features a seat **3405** that may pass through or be integral with the end-cap **3410** of a cylinder assembly. The opening **3415** in the seat **3405** constitutes the gated port **3415** of the valve **3400**. Valve **3400** also includes a valve member **3420**, a permanent magnet **3425** attached to or integral with the valve member **3420**, and an actuation mechanism **3430**. The actuation mechanism **3430** may include or consist essentially of a ferromagnetic core **3435** and a winding **3440** through which an electric current may be made to flow. In FIG. **34**A, the seat **3405** and winding **3440** are depicted as aligned tubular or ring-shaped structures viewed in cross-section. As depicted in FIG. **34**A, current is flowing around the winding **3440** clockwise as viewed from the gated port **3415**. In the portion of the winding **3440** depicted in cross-section to the left of the ferromagnetic core **3435**, current is moving directly out of the page, while in the portion of the winding **3440** depicted in cross-section to the right of the ferromagnetic core **3435**, current is moving directly into of the page, as indicated by conventional symbols **3445**. This direction of flow is herein termed the "clockwise" direction. When current flows clockwise in the winding **3440**, the ferromagnetic core **3435** is magnetized so that the end of the core **3435** distal to the gated port **3415** is a north magnetic pole and the end of the core **3435** proximal to the gated port **3415** is a south magnetic pole. The permanent magnet **3425** is fixed so that its north pole is distal to the gated port **3415** and its south pole is proximal to the gated port **3415**.

[0433] In FIG. **34**A, the valve **3400** is depicted in a partly open state. The sense of magnetization of the core **3435** causes a south magnetic pole to face the north pole of the permanent magnet **3425** across a gap **3450**. When valve **3400** is in a fully open state, the gap **3450** is minimal or absent (e.g., the permanent magnet **3425** and core **3435** may be in contact). An attractive magnetic force proportional to the inverse square of the effective distance x between the north pole of the permanent magnet **3425** and the south pole of the core **3435**, where x is approximately proportional to the width of the gap **3450** plus a nonzero constant, acts on both the actuation mechanism **3430** and the valve member **3420**. The actuation mechanism **3430** is connected to the body (not shown) of the valve **3400** and is not free to move towards or away from the gated port **3415**, whereas the valve member **3420** is free to move towards or away from the gated port **3415**. The magnetic upward force (magnetic opening force) F_{om} tends to cause the valve member **3420** to move toward the actuation mechanism **3430**. The magnetic opening force F_{om} may vary as a function of time, depending on the width of the gap **3450** and the direction and magnitude of the current in the winding **3440**. In FIG. **34**A, the core **3435** is depicted as a solid cylinder of ferromagnetic material; in other embodiments, the core **3435** has other shapes, and may be constructed so that its reluctance may be altered by an operator or control system, in which case the reluctance of the core **3435** may vary with time. Where the reluctance of the core **3435** may vary with time, the magnetic force may depend in a time-varying way on the reluctance of the core **3435** as well as on the width of gap **3450** and the current in the winding **3440**.

[0434] The direction and magnitude of the current in the winding **3440** may be deliberately varied during operation of the valve **3400** to achieve various operational advantages. For example, when the valve member **3420** is in contact with the seat **3405** (i.e., when the valve **3400** is closed), and opening of the valve **3400** is initiated, a relatively large current in the winding **3440** may be employed to accelerate the valve member **3420** away from the seat **3405**. This current may be decreased (or even removed entirely) as the valve member **3420** moves toward the actuation mechanism **3430**.

[0435] Differential pressure in the cylinder chamber **3455** and flow chamber **3460** may, in some states of operation, provide an additional hydraulic opening force F_{oh} that acts in the same sense as the magnetic opening force F_{om}. For example, the pressure in the cylinder chamber **3455** may be greater than the pressure in the flow chamber **3460**. In this case, after opening of the valve **3400** has been initiated and the valve member **3420** is no longer in contact with the seat **3405**, fluid flow **3465** will generally occur through the gated port **3415**. The fluid flow **3465** will tend to continue to exert a hydraulic opening force F_{oh} on the valve member **3420** throughout the opening process, though F_{oh} may diminish as the valve member **3420** moves away from the gated port **3415**. The hydraulic opening force F_{oh} may suffice to initiate, assist, and/or complete the opening of valve **3400**.

[0436] As the valve member **3420** moves toward the actuation mechanism **3430**, the current in the winding **3440** (and thus the magnetic opening force F_{om}) may be varied in such a manner as to increase, maintain, decrease, or reverse the acceleration of the valve member **3420** toward the actuation mechanism **3430**. Through suitable variation of the winding current and thus of F_{om}, rapid opening of the valve **3400** and reduction of collision forces may be achieved with minimal expenditure of energy.

[0437] In another state of operation, not depicted in FIG. **34**A, the direction of the current in the winding **3440** is counterclockwise. In this state of operation, the north and south poles of the ferromagnetic element **3435** will be reversed, and a north magnetic pole will be presented by the actuation mechanism **3430** to the north magnetic pole of the permanent magnet **3425**. The proximity of the two north magnetic poles will cause a repulsive force (magnetic closing force F_{cm}) to act upon the valve member **3420**. Depending on the differential pressure in the flow chamber **3460** and the cylinder chamber **3455**, a hydraulic force may act on the valve member **3420** either in concordance with or in opposition to the magnetic closing force F_{cm}. The valve member **3420** will accelerate in the direction of the net or sum force upon it. Thus, regardless of differential pressure in the flow chamber **3460** and cylinder chamber **3455**, the valve member **3420** will move toward the gated port **3415** if a sufficiently large magnetic closing force F_{cm} is exerted upon the valve member **3420** by the actuation mechanism **3430**. In short, the valve **3400** may be closed, regardless of differential pressure across the valve **3400**, by passage of a sufficiently large counterclockwise current through the winding **3440**.

[0438] During closing of valve **3400**, as the valve member **3420** moves away from the actuation mechanism **3430**, the current in the winding **3440** (and thus the magnetic closing force F_{cm}) may be varied in such a manner as to increase, maintain, decrease, or reverse the acceleration of the valve member **3420** away from the actuation mechanism **3430**. Through suitable variation of the current and thus of F_{cm}, rapid closing of the valve **3400** and reduction of collision forces may be achieved with minimal expenditure of energy. Additionally, a ferromagnetic material (e.g. flux intensifier, not shown) may be positioned in the return flux path (at least in part above or surrounding actuation mechanism **3430**) to maximize or otherwise optimize the flux density at the valve seat.

[0439] FIG. **34**B is a schematic cross-sectional drawing of the valve of FIG. **34**A in a different state of operation. As depicted in FIG. **34**B, the valve **3400** is closed, i.e., the valve member **3420** is in contact with the seat **3405**, occluding the gated port **3415**. Differential pressure in the flow chamber **3460** and cylinder chamber **3455** may provide a force that either tends to hold the valve member **3420** in contact with the seat **3405** (i.e., to the hold the valve **3400** closed) or that tends to move the valve member **3420** away from the seat **3405** (i.e., to open the valve **3400**).

[0440] In FIG. **34**B, the current in the winding **3440** is depicted as moving in a counterclockwise direction. Consequently, the ferromagnetic element **3430** presents a north magnetic pole to the north magnetic pole of the permanent magnet **3425**, and a magnetic closing force F_{cm} tending to hold the valve **3400** closed will act upon the valve member **3420**. If the differential pressure in the flow chamber **3460** and cylinder chamber **3455** is such as to hold the valve **3400** closed, the valve **3400** will remain closed even if the current in the winding **3440** is zero; if the differential pressure in the flow chamber **3460** and cylinder chamber **3455** is such as to open the valve **3400**, the current in the winding **3435** may be set to a value that produces a countervailing magnetic closing force F_{cm} sufficient to keep the valve closed (i.e., an F_{cm} larger than the hydraulic opening force exerted by the differential pressure).

[0441] FIG. **35**A is a schematic cross-sectional drawing of major components of an illustrative electromagnetic valve **3500** in accordance with embodiments of the present invention. For clarity, components of the valve **3500**, including an outside port and the walls of the flow chamber, are not depicted in FIG. **35**A. Valve **3500** is actuated by differential pressure and by electromagnetic force. In other embodiments, the valve **3500** may be actuated by differential pressure, electromagnetic forces, and/or mechanical forces in various states of operation. The valve **3500** may include a mechanical or pneumatic spring (not shown) to bias the valve towards closing, cushion opening forces, and/or replace or supplement the closing actuation mechanism. The valve **3500** shown in FIG. **35**A is a low-side valve, as defined above, and in various embodiments may be substituted for the poppet-style low-side valves depicted in previously described figures, thereby realizing additional advantages.

[0442] Valve **3500** features a seat **3505** that may pass through or be integral with the end cap **3510** of a cylinder assembly (not otherwise shown). The opening **3515** in the seat constitutes the gated port **3515** of the valve **3500**. Valve **3500** also includes a valve member **3520**, a rod **3570** attached to the valve member **3520**, a permanent magnet **3525** attached to or integral with the rod **3570**, and an actuation mechanism **3530**. The actuation mechanism **3530** may include or consist essentially of a ferromagnetic core **3535** and a winding **3540** through which an electric current may be made to flow. In FIG. **35**A, the seat **3505**, core **3535**, and winding **3540** are depicted as aligned, tubular or ring-shaped structures viewed in cross-section. As depicted in FIG. **35**A, the current in the winding **3535** is moving in a clockwise direction as defined above. When current flows clockwise in the winding **3540**, the ferromagnetic core **3535** is magnetized so that the end of the core **3535** distal to the gated port **3515** is a north magnetic pole and the end of the core **3535** proximal to the gated port **3515** is a south magnetic pole. The permanent magnet **3525** is fixed so that its south pole is proximal to the gated port **3515** and its north pole is distal to the gated port **3515**.

[0443] In FIG. **35**A, the valve **3500** is depicted in an fully open state. The magnetization of the core **3535** causes a north magnetic pole to face the south pole of the permanent magnet **3525** across a gap **3550**. When valve **3500** is in a fully open state, gap **3550** is minimal or absent (e.g., the permanent magnet **3525** and core **3535** may be in contact). An attractive magnetic force proportional to the inverse square of the effective distance x between the south pole of the permanent magnet **3525** and the north pole of the core **3535**, where x is approximately proportional to the width of the gap **3550** plus a nonzero constant, acts on both the actuation mechanism **3530** and the permanent magnet **3525**. The magnetic force acting on the permanent magnet **3525** is communicated to the rod **3570** and valve member **3520**. The actuation mechanism **3530** is connected to the body (not shown) of the valve **3500** and is not free to move towards or away from the gated port **3515**, whereas the permanent magnet **3525**, rod **3570**, and valve member **3520** are free to move towards or away from the gated port **3515**. The magnetic downward force (magnetic opening force) F_{om} acting on the permanent magnet **3525** tends to cause the valve member **3520** to move downward (i.e., away from the gated port **3515**). The magnetic opening force F_{om} may vary as a function of time, depending on the width of the gap **3550** and the direction and magnitude of the current in the winding **3540**. In FIG. **35**A, the core **3535** is depicted as a tube of solid ferromagnetic material; in other embodiments, the core **3535** is constructed so that its reluctance may be altered by an operator or control system, in which case the reluctance of the core **3535** may vary with time. If the reluctance of the core **3535** may vary with time, the magnetic force may depend in a time-varying way on the reluctance of the core **3535** as well as on the width of the gap **3550** and the current in the winding **3540**.

[0444] The direction and magnitude of the current in the winding **3540** may be deliberately varied during operation of the valve **3500** to achieve various operational advantages. For example, when the valve member **3520** is in contact with the seat **3505** (i.e., when the valve **3500** is closed), and opening of the valve **3500** is initiated, a relatively large current in the winding **3540** may be employed to accelerate the valve member **3520** away from the seat **3505**, speeding opening of the valve **3500**. This current may be decreased as the permanent magnet **3525** moves toward the actuation mechanism **3530**.

[0445] Differential pressure in the cylinder chamber **3555** and flow chamber **3560** may, in some states of operation, provide an additional hydraulic opening force F_{oh} that acts in the same sense as the magnetic opening force F_{om}. For example, the pressure in the cylinder chamber **3555** may be less than the pressure in the flow chamber **3560**. In this case, after opening of the valve **3500** has been initiated and the valve member **3520** is no longer in complete contact with the seat **3505**, fluid flow **3565** will generally occur through the gated port **3515**. The fluid flow **3565** will tend to continue to exert a hydraulic opening (downward) force F_{oh} on the valve member **3520** throughout the opening process. F_{oh} may diminish as the valve member **3520** moves away from the gated port **3515**. The hydraulic opening force F_{oh} may suffice to initiate, assist, and/or complete the opening of valve **3500**.

[0446] During opening of valve **3500**, as the permanent magnet **3525** moves downward (toward the actuation mechanism **3530**), the current in the winding **3540** (and thus the magnetic opening force F_{on}) may be varied in such a manner as to increase, maintain, decrease, or reverse the acceleration of the valve member **3520** away from the seat **3505**. Through suitable variation of the current and thus of F_{om}, rapid opening of the valve **3500** and reduction of collision forces may be achieved with minimal expenditure of energy.

[0447] In another state of operation, not depicted in FIG. **35**A, the direction of the current in the winding **3540** may be made counterclockwise. In this state of operation, the north and south poles of the ferromagnetic element **3535** will be reversed, and a south magnetic pole will be presented by the actuation mechanism **3530** to the south magnetic pole of the permanent magnet **3525**. The proximity of the two south magnetic poles will cause an upward (closing) force F_{cm} to act upon the permanent magnet **3525**, rod **3570**, and valve member **3520**. The valve member **3520** will thus tend to move upward, i.e., in the direction of the gated port **3515**. Depending on the differential pressure in the flow chamber **3560** and the cylinder chamber **3555**, a hydraulic force may act in concordance with, or in opposition to, the magnetic closing force F_{cm}. The valve member **3520** will accelerate in the direction of the net or sum force upon it. Thus, regardless of differential pressure in the flow chamber **3560** and cylinder chamber **3555**, the valve member **3520** will tend to move toward the gated port **3515** if a sufficiently large magnetic closing force F_{cm} is exerted upon the valve member **3520** by the actuation mechanism **3530**. In short, the valve **3500** may be closed, regardless of differential pressure across the valve **3500**, by passage of a sufficiently large counterclockwise current through the winding **3540**.

[0448] During closing of valve **3500**, as the valve member **3520** moves toward the actuation mechanism **3530**, the current in the winding **3540** (and thus the magnetic closing force F_{cm}) may be varied in such a manner as to increase, maintain, decrease, or reverse the acceleration of the valve member **3520** toward the actuation mechanism **3530**. Through suitable variation of the current and thus of F_{cm}, rapid closing of the valve **3500** and reduction of collision forces may be achieved with minimal expenditure of energy. Additionally, a ferromagnetic material (e.g., a flux intensifier, not shown) may be positioned in the return flux path (at least in part below or surrounding actuation mechanism **3530**) to maximize or otherwise optimize the flux density at the valve seat.

[0449] FIG. **35**B is a schematic cross-sectional drawing of the valve of FIG. **35**A in a different state of operation. As depicted in FIG. **35**B, the valve **3500** is closed, i.e., the valve member **3520** is in contact with the seat **3505**, occluding the gated port **3515**. Differential pressure in the flow chamber **3560** and cylinder chamber **3555** may provide a force that either tends to hold the valve member **3520** in contact with the seat **3505** (i.e., to the hold the valve **3500** closed) or that tends to move the valve member **3520** away from the seat **3505** (i.e., to open the valve **3500**).

[0450] In FIG. **35**B, the current in the winding **3540** is depicted as moving in a counterclockwise direction. Consequently, the ferromagnetic element **3535** presents a south magnetic pole to the south magnetic pole of the permanent magnet **3525**, and a magnetic force tending to hold the valve **3500** closed will be produced. If the differential pressure in the flow chamber **3560** and cylinder chamber **3555** is such as to hold the valve **3500** closed, the valve **3500** will remain closed even if the current in the winding **3540** is zero; if the differential pressure in the flow chamber **3560** and cylinder chamber **3555** is such as to open the valve **3500**, the current in the winding **3540** may be set to a value that produces a magnetic closing force F_{cm} sufficient to keep the valve closed (i.e., F_{cm} larger than the hydraulic opening force exerted by the differential pressure). Generally, the systems described herein may be operated in both an expansion mode and in the reverse compression mode as part of a full-cycle energy storage system with high efficiency. For example, the systems may be operated as both compressor and expander, storing electricity in the form of the potential energy of compressed gas and producing electricity from the potential energy of compressed gas. Alternatively, the systems may be operated independently as compressors or expanders.

[0451] Embodiments of the invention may, during operation, convert energy stored in the form of compressed gas and/or recovered from the expansion of compressed gas into gravitational potential energy, e.g., of a raised mass, as described in U.S. patent application Ser. No. 13/221,563, filed Aug. 30, 2011, the entire disclosure of which is incorporated herein by reference.

[0452] The terms and expressions employed herein are used as terms of description and not of limitation, and there is no intention, in the use of such terms and expressions, of excluding any equivalents of the features shown and described or portions thereof, but it is recognized that various modifications are possible within the scope of the invention claimed.

What is claimed is:

1. A method for at least one of storing energy in or recovering energy with an energy-storage system comprising (i) a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port, the valve comprising a valve member for occluding the gated port, and (ii) an actuation system for actuating the valve, the actuation system comprising (a) an actuation cylinder and (b) a piston disposed within and dividing the actuation cylinder into first and second chambers, the method comprising:

within the cylinder assembly, at least one of (i) compressing gas to store energy or (ii) expanding gas to recover energy; and

at least one of prior to, during, or after the at least one of compression or expansion, at least one of admitting fluid into or exhausting fluid from the cylinder assembly at least in part by actuating the valve from a closed state to an open state by admitting fluid into the first chamber of the actuation cylinder to increase fluid pressure therein, thereby moving the piston toward the second chamber,

wherein, during the actuation, (i) fluid exits the second chamber of the actuation cylinder at a first rate to maximize speed of the piston motion, and (ii) thereafter, fluid exits the second chamber at a second rate slower than the first rate to decelerate the piston before the piston reaches an end surface of the actuation cylinder.

2. The method of claim **1**, wherein the second rate of fluid flow decreases as the piston moves toward the end surface of the actuation cylinder.

3. The method of claim **1**, wherein, during the actuation, the piston occludes at least a portion of an orifice in the second chamber as the piston moves toward an end surface of the actuation cylinder, thereby slowing the flow of fluid from the second chamber from the first rate to the second rate.

4. The method of claim **3**, wherein, when the piston is disposed proximate the end surface, the orifice is completely occluded by the piston.

5. The method of claim **3**, wherein a lateral dimension of at least a portion of the orifice varies as a function of distance from the end surface of the actuation cylinder.

6. The method of claim **3**, wherein (i) a lateral dimension of a first portion of the orifice does not vary as a function of distance from the end surface of the actuation cylinder and (ii) a lateral dimension of a second portion of the orifice varies as a function of distance from the end surface of the actuation cylinder.

7. The method of claim **3**, wherein a lateral boundary of at least a portion of the orifice has a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the orifice is partially occluded.

8. The method of claim **1**, wherein fluid is admitted into the first chamber through both (i) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

9. The method of claim **1**, wherein, during at least a portion of the actuation, fluid exits the second chamber through both (i) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

10. A method for at least one of storing energy in or recovering energy with an energy-storage system comprising (i) a cylinder assembly having a valve for controlling fluid flow into and out of the cylinder assembly through a gated port, the valve comprising a valve member for occluding the gated port, and (ii) an actuation system for actuating the valve, the actuation system comprising (a) an actuation cylinder, (b) a piston disposed within and dividing the actuation cylinder into first and second chambers, and (c) an occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, the method comprising:

within the cylinder assembly, at least one of (i) compressing gas to store energy or (ii) expanding gas to recover energy; and

at least one of prior to, during, or after the at least one of compression or expansion, at least one of admitting fluid into or exhausting fluid from the cylinder assembly at least in part by actuating the valve from a closed state to an open state by admitting fluid into the first chamber of the actuation cylinder to increase fluid pressure therein, thereby moving the piston toward the second chamber,

wherein, during the actuation, (i) fluid flows out of the second chamber through the occludable orifice unoccluded by the piston, thereby maximizing speed of the piston motion, and (ii) thereafter, the piston occludes at least a portion of the occludable orifice, whereby fluid flow from the second chamber is decreased to decelerate the piston before the piston reaches an end surface of the actuation cylinder.

11. The method of claim **10**, wherein the occludable orifice is completely occluded by the piston by the end of the actuation.

12. The method of claim **10**, wherein a lateral dimension of at least a portion of the occludable orifice varies as a function of distance from an end surface of the actuation cylinder.

13. The method of claim **10**, wherein (i) a lateral dimension of a first portion of the occludable orifice does not vary as a function of distance from an end surface of the actuation cylinder and (ii) a lateral dimension of a second portion of the occludable orifice varies as a function of distance from the end surface of the actuation cylinder.

14. The method of claim **10**, wherein a lateral boundary of at least a portion of the occludable orifice has a shape defined by a function $y(x)=C(V_{max}^2-2Ax)^{1/2}$, where C is a constant, V_{max} is a velocity of the piston in the actuation cylinder when the occludable orifice is not occluded, and A is a magnitude of deceleration of the piston in the actuation cylinder when the occludable orifice is partially occluded.

15. The method of claim **10**, wherein fluid is admitted into the first chamber through both (i) a second occludable orifice configured to be at least partially occluded by the piston during movement of the piston within the actuation cylinder, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

16. The method of claim **10**, wherein, during at least a portion of the actuation, fluid exits the second chamber through both (i) the occludable orifice, and (ii) a fixed orifice configured to not be occluded by the piston during movement of the piston within the actuation cylinder.

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