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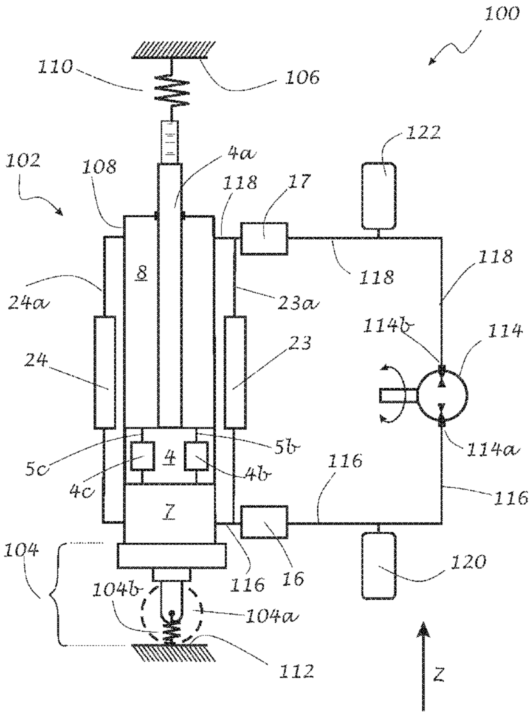

Fig. 1

(57) Abstract: Active suspension actuator systems including an actuator with a compression volume and an extension volume are described. In some embodiments, the system includes one or more flow control devices in fluid communication with the compression volume and/or the extension volume of the actuator. In some instances, a flow control device may include a pressure balanced blow-off valve (PBOV). In some embodiments, the system includes a high capacity bidirectional base valve. In some embodiments, two or more flow control devices cooperate to, for example, damp low amplitude oscillations in the extension and/or compression volumes, and to allow the build-up of pump generated differential pressures while discharging rapid road induced differential pressure spikes between the extension and compression volumes.

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Pressure compensated active suspension actuator system

RELATED APPLICATIONS

[0001] This application claims the benefit under 35 U.S.C. § 119(e) of U.S. application number 62/863,202, filed June 18, 2019, U.S. application number 62/842,088, filed May 2, 2019, U.S. application number 62/821,834, filed March 21, 2019, and U.S. application number 62/816,666, filed March 11, 2019, the disclosures of which are incorporated herein by reference in their entirety.

FIELD

[0002] Disclosed embodiments are related to vehicular active suspension systems that include hydraulic actuators.

BACKGROUND

[0003] Vehicular hydraulic active suspension systems, which take advantage of fluids to store, convert, and/or transmit power, may generally include one or more hydraulic actuators. The flow of hydraulic fluid within an active actuator system may be controlled by a hydraulic machine that functions as a pump, in some operating conditions, and/or hydraulic motor, in other operating conditions, in order to control a force applied between a sprung mass and an unsprung mass of a vehicle.

SUMMARY

[0004] In some embodiments, an active suspension actuator system may include an actuator that has a pressure tube. The pressure tube may have an internal volume that slidably receives a piston which divides at least a portion of the internal volume into a compression volume and an extension volume. In at least one mode of operation the pressure in the compression volume may oscillate at multiple frequencies that may include a first frequency. The system may also include hydraulic machine with a first port and a second port, where the first port may be connected to the compression volume by a first flow path that does not include the second port and where the second port is connected to the extension volume by a second flow path that does not include the first port. The hydraulic machine may be a hydraulic pump or a hydraulic motor. The system may also include a controller that operates the hydraulic machine as a hydraulic pump in at least a first operating mode and as a hydraulic motor in a second operating mode. As used herein, the term “compression volume” refers to the portion of the internal volume of an actuator which is compressed by the piston when the actuator is compressed, and the term “extension volume” refers to the portion of the internal volume of the actuator which is compressed by the piston when the actuator is extended during rebound.

[0005] The system may also include a damping flow control device, with a total impedance that resists the flow in a manner that damps flow oscillations in at least a portion of the first flow path. The flow control device may be structured such that its impedance (i.e. its damping effect) decreases when the pressure drop across the flow control device exceeds a first preset threshold pressure drop.

[0006] The system may also include a first bypass flow control device that is located between the compression volume and extension volume. The first bypass flow control device may include a first pressure balanced blow-off valve (PBOV) with a pressure offset. The pressure offset may prevent the PBOV from opening unless the pressure in the compression chamber or the pressure differential across the piston exceeds a preselected value. Once the pressure threshold (i.e. either the pressure differential threshold or the absolute pressure threshold) is exceed, the PBOV responds or effectively responds to pressure changes above a certain frequency but not to pressure changes that are below such a frequency. A PBOV may also operate in a similar fashion to discharge fluid from the extension volume into the compression volume. In some embodiments, a second PBOV may be used in conjunction with the first PBOV such that one discharges from the compression volume to the extension volume while the second PBOV discharges in the opposite direction.

[0007] In some embodiments the pressure offset may be an absolute offset or a relative offset. The damping coefficient of at least a portion of the first flow path may be greater than 10 but less than 400 newton-meters per second. Damping coefficients outside this range may also be used as the disclosure is not limited to this range.

[0008] In some embodiments, the damping coefficient of the damping flow control device may be different when the flow is towards the compression volume compared to when the flow is away from the control volume.

[0009] The in some embodiments, the first bypass flow control device discharges may include a hydraulic filter that blocks or effectively blocks frequencies between 10 Hz and 100 Hz. In some embodiments hydraulic filters may block or effectively block frequencies in other ranges may also be used as the disclosure is not limited to this frequency range.

[0010] In some embodiments the one or more bypass flow control devices may be integrated with the piston. In some embodiments one or more PBOVs may be located in other flow paths that may be external to the pressure tube.

[0011] In some embodiments, an active suspension actuator system of a vehicle may include a piston with a first face (or side) that includes a first inlet port and a second face (or side) that includes a first outlet port. The first inlet port may be fluidly connected to the first outlet port by a first fluid flow passage internal to the piston. The actuator may also include a pressure tube with an internal volume (e.g. a cylindrical volume) divided into a first chamber (e.g. a compression volume or an extension volume) and a second chamber (e.g. an extension volume or a compression volume) by the piston. The piston may be slidably received in the internal volume, and the first inlet port may be fluidly connected to the first chamber. A piston rod may be included that is attached to a sprung mass of the vehicle (e.g. a vehicle body) at a first end and the piston at a second end. The system may also include a first flow control device attached to the piston (located on the compression volume side of the piston or the extension volume side of the piston) that selectively, fluidly connects the first outlet port to the second chamber based at least in part on the pressure differential between the first and second chambers and/or one or more of the following properties: (i) a rate of pressure rise in the first chamber, (ii) a rate of change of the pressure differential between the first and second chambers, (iii) a rate of pressure drop in the second chamber, and (iv) a frequency of pressure fluctuation in the first chamber. As used herein, the term “compression volume” in an actuator refers to a volume in the actuator that is compressed by the piston during a compression stroke of the actuator while the term “extension volume” refers to a volume in the actuator that is compressed by the piston during an extension stroke of the actuator.

[0012] In some embodiments, an active suspension actuator system of a vehicle may also include a second flow control device attached to the piston. Additionally, the first face of the piston may also include a second outlet port and the second face may also include a second inlet port, where the second inlet port may be fluidly connected to the second outlet port by a second fluid flow passage internal to the piston. Additionally, second inlet port may be fluidly connected to the second chamber, and the second flow control device may selectively, fluidly connect the second outlet port to the first chamber based at least in part on the pressure differential between the first and second chambers and/or one or more of the following properties: (i) a rate of pressure rise in the second chamber, (ii) rate of change of the pressure differential between the first and second chambers, (iii) a rate of pressure drop in the first chamber, and (iv) a frequency of pressure fluctuation in the first chamber, and (iv) a frequency of pressure fluctuation in the second chamber. As used herein, the term “selectively connect” refers to using a flow control device to allow the passage of fluid under some operating conditions and to block or effectively block fluid flow under other operating conditions.

[0013] In some embodiments of an active suspension actuator system, the flow control device may be a pressure balanced blow-off valve. In some embodiments, the pressure balanced blow-off device may include a hydraulic low pass filter that includes a dissipative element (e.g. an orifice, a laminar flow element, a viscous damping element) and a compliance element (e.g. a gas filled chamber or a spring loaded piston). In some embodiments. The low pass filter may be used to delay the rise pressure in one volume in the valve relative a second volume in the valve. The pressure balanced blow-off valve may be a passive valve (e.g. is not energized by electricity).

[0014] In some embodiments, an active suspension actuator system may include a sealing element (e.g. a sealing washer, a shim stack), that selectively controls fluid flow from an outlet port in a piston, is biased in a closed position by at least two spring elements (e.g. coil springs). Additionally, one of the at least two bias springs in the pressure balanced blow-off valve may be operatively interposed between a spring perch, that is fixed relative to the piston rod, and the sealing element. A second of the at least two bias springs may be operatively interposed between a spring perch, that moves relative piston rod, and the sealing element.

[0015] In some embodiments, a method of operating an active suspension actuator system of a vehicle may include operating the vehicle over a road surface with an active suspension actuator operatively interposed between a wheel assembly and the vehicle body. The actuator may include a compression volume and an extension volume that are fluidly connected to a first and second port of a hydraulic machine (e.g. a hydraulic pump operating as a hydraulic motor and/or a pump, or a hydraulic motor operating as a pump and/or a hydraulic motor) respectively. During operation a wheel associated with the wheel assembly may strike or encounter a discontinuity in the road (e.g. pothole, bump, speed bump, a crack, an expansion joint, a curb, debris). Striking or encountering the discontinuity, may result in an increase in pressure and/or an increase in the pressure differential between the compression volume and extension volume increase to a first value greater than a pre-selected threshold value. This increase may be at a first rate of increase. In response, a pressure balanced blow-off valve may open to discharge at least a portion of the increased pressure or pressure differential by establishing flow between the compression and extension volumes. Under certain operating conditions, the hydraulic machine may be used to increase the pressure in the compression volume or the pressure differential between the compression and extension volumes to a value greater than the threshold but at a rate lower than the first rate without opening the pressure balanced blow-off valve. Under certain operating conditions, the hydraulic machine may be used to increase the pressure in the compression volume or the pressure differential between the compression and extension volumes to a value less than the threshold but at a rate higher than the first rate without opening the pressure balanced blow-off valve. In some embodiments, the pressure balanced blow-off valve may be a passive valve.

[0016] In some embodiments, a method of operating an active suspension actuator system of a vehicle may include operating the vehicle over a road surface with an active suspension actuator operatively interposed between a wheel assembly and the vehicle body. The actuator may include an internal cylindrical volume that is separated into a first chamber and a second chamber by a piston that is slidably received in the internal volume, and where the first chamber and second chamber are fluidly connected to a first and second port of a hydraulic machine respectively. During operation a wheel associated with the wheel assembly may strike or encounter a discontinuity in the road (e.g. pothole, bump, speed bump, a crack, an expansion joint, a curb, debris). Striking or encountering the discontinuity, may result in an increase in pressure in the first chamber and/or the first chamber relative to the second chamber to a first value greater than a pre-selected threshold value at a first rate of increase. The method may further include opening a pressure balanced blow-off valve to discharge at least a portion of the increased pressure or the differential pressure by allowing fluid to flow from the first chamber to the second chamber. The method may further include operating the hydraulic machine to increase the pressure in the first chamber or the pressure differential between the chambers to a value greater than the threshold at a rate lower than the first rate without opening the pressure balanced blow-off valve. The method may further include operating the hydraulic machine to increase the pressure in the first chamber and/or the pressure differential between the chambers to a value less than the threshold value at a rate greater than the first rate without opening the pressure balanced blow-off valve. In some embodiments the pressure balanced blow-off valve may be a passive valve. In some embodiments the first chamber may be a compression volume and the second chamber may be an extension volume.

[0017] In one embodiment, an active suspension actuator, includes: an actuator that includes a pressure tube and a second tube at least partially surrounding the pressure tube, wherein the pressure tube includes a compression volume and an extension volume separated by a piston, wherein the pressure tube and the second tube have an extension volume end and a compression volume end, wherein the pressure tube and the extension tube form an intervening volume; a base valve assembly, located at a compression volume end of the pressure tube and the second tube, that includes: a base valve body with that includes internal compression flow conduits and extension flow conduits that fluidly connect the compression volume and at least a portion of the intervening volume; an extension shim stack securely attached to the valve body that includes an extension sealing shim that controls the flow from the intervening volume via the extension flow conduits to the compression volume and prevents flow in the opposite direction in the extension flow conduits; and a compression shim stack securely attached to the valve body that includes a compression sealing shim that controls the flow from the compression volume via the compression flow conduits to the intervening volume and prevents flow in the opposite direction in the compression flow conduits.

[0018] In another embodiment, a bidirectional base valve of an active suspension actuator includes: a base valve body with a first set of internal flow channels and a second set of internal flow channels; a first shim stack that is configured to regulate flow, through the first set of flow channels, from a compression volume of the actuator to a second volume in the actuator and blocks flow from the second volume from entering the first set of flow channels; and a second shim stack that is configured to regulate flow, through the second set of flow channels, that flows from the second volume to the compression volume and blocks flow from the compression volume from entering the second set of flow channels.

[0019] It should be appreciated that the foregoing concepts, and additional concepts discussed below, may be arranged in any suitable combination, as the present disclosure is not limited in this respect. Further, other advantages and novel features of the present disclosure will become apparent from the following detailed description of various non-limiting embodiments when considered in conjunction with the accompanying figures. In cases where the present specification and a document incorporated by reference include conflicting and/or inconsistent disclosure, the present specification shall control.

BRIEF DESCRIPTION OF DRAWINGS

[0020] The accompanying drawings are not intended to be drawn to scale. In the drawings, each identical or nearly identical component that is illustrated in various figures may be represented by a like numeral. For purposes of clarity, not every component may be labeled in every drawing. In the drawings:

[0021] Fig. 1 illustrates an embodiment of a vehicular active suspension actuator system with multiple flow control devices;

[0022] Fig. 2 illustrates an embodiment of a vehicular active suspension actuator system with multiple gas pressurized accumulators, and a flow passage fluidly connected to the compression volume;

[0023] Fig.3 shows the relationship between pressure drop and flow velocity between two positions in the flow passage of Fig. 2;

[0024] Fig. 4 illustrates the embodiment of a vehicular active suspension actuator system of Fig. 2 with an embodiment of a flow control device located in the flow passage fluidly connected to the compression volume;

[0025] Fig. 5 shows the relationship between pressure drop and flow velocity between the two positions in the flow passage of Fig. 4;

[0026] Fig. 6 illustrates the embodiment of a vehicular active suspension actuator system that includes another embodiment of a flow control device located in the flow passage fluidly connected to the compression volume;

[0027] Fig. 7 shows the relationship between pressure drop and flow velocity between the two points in the flow passage of Fig. 6;

[0028] Fig. 8 illustrates an embodiment of a vehicular active suspension actuator system that includes another embodiment of a flow control device located in the flow passage fluidly connected to the compression volume;

[0029] Fig. 9 shows the relationship between pressure drop and flow velocity between the two positions in the flow passage of Fig. 8;

[0030] Fig. 10 illustrates an embodiment of a vehicular active suspension actuator system that includes an embodiment of a flow control device fluidly interposed in a flow passage of Fig. 2 that is fluidly connected to the extension volume;

[0031] Fig. 11 illustrates an embodiment of a pressure balanced blow off valve with a preset absolute pressure offset;

[0032] Fig. 12 a graph showing the relationship between the degree of opening of a pressure balanced blow-off valve illustrated in Fig. 11 as a function of the frequency of pressure oscillations to which the inlet of the pressure balanced blow-off valve is exposed.;

[0033] Fig. 13 illustrates another embodiment of a pressure balanced blow off valve with a preset relative pressure offset; and

[0034] Fig. 14 illustrates a further embodiment of a vehicular active suspension actuator system with multiple flow control devices.

[0035] Fig. 15 illustrates an embodiment of a vehicular active suspension actuator system with multiple flow control devices;

[0036] Fig. 16 illustrates a cross section view of a piston of an active suspension actuator with dual pressure balanced blow-off valves incorporated into the piston;

[0037] Fig. 17 illustrates a cross section view of a piston of an active suspension actuator with a single pressure balanced blow-off valve incorporated into the piston;

[0038] Fig. 18 illustrates the piston, piston/rod interface, and dual pressure balanced blow-off valves of Fig. 1 in greater detail;

[0039] Fig. 19 illustrates a cross section elevation view of a sealing washer and the forces acting on it in the longitudinal direction;

[0040] Fig. 20 illustrates a piston face of the piston shown in Fig. 15;

[0041] Fig. 21 illustrates a second piston face of the piston shown in Fig. 15;

[0042] Fig. 22 a perspective exploded cross section view of the piston assembly of Fig. 18;

[0043] Fig. 23 illustrates of the apparatus of Fig. 18 with a spring loaded sealing washer displaced from its seated position;

[0044] Fig. 24 illustrates the apparatus of Fig. 18 with a floating spring perch displaced from its resting position.

[0045] Fig. 25 shows a perspective cross-section of an embodiment of an actuator piston with two shim-stack PBOVs each of which includes an expandable pressure chamber;

[0046] Fig. 26 shows a planar cross-section of the embodiment in Fig. 25;

[0047] Fig. 27 shows a planar cross-section view of the embodiment in Fig. 25 with an enlarged pressure chamber in the compression-side PBOV;

[0048] Fig. 28 illustrates a perspective cross section view of an embodiment of a dual poppet PBOV piston assembly with a variable bleed;

[0049] Fig. 29 illustrates a planar cross section view of the embodiment of Fig. 28;

[0050] Fig. 30 illustrates a plana cross section view of the embodiment of Fig. 29 with a blocked bleed passage.

[0051] Fig. 31 illustrates an embodiment of an actuator with a base valve incorporated at the compression end of a twin tube actuator;

[0052] Fig. 32 illustrates an embodiment of an actuator with a flow control device configured to regulate flow into and out of the extension volume;

[0053] Fig. 33 shows a graph of the relationship of pressure drop across a base valve and the flow rate through the base valve;

[0054] Fig. 34 shows an isometric drawing illustrating an embodiment of the body of a bidirectional base valve including a bleed hole;

[0055] Fig. 35 illustrates an isometric cross section view of the embodiment of Fig. 34

[0056] Fig. 36 illustrates a first cross section of a base-valve with each shim-stack in a closed position;

[0057] Fig. 38 illustrates the first cross section of Fig. 36 where the compression shim-stack in an open position;

[0058] Fig. 38 illustrates a second cross section of the base-valve with the extension shim stack in an open position;

[0059] Fig. 39 illustrates a first cross section of another embodiment of a bidirectional base-valve;

[0060] Fig. 40 illustrates the embodiment of Fig. 39 with the extension shim stack in an open position;

[0061] Fig. 41 illustrates a second cross section of the embodiment of Fig. 39;

[0062] Fig. 42 illustrates the second cross section of the base-valve of Fig. 39 with the extension shim-stack is in an open position;

[0063] Fig. 43 shows a section of isometric view of yet another embodiment of a bidirectional base valve;

[0064] Fig. 44 illustrates a first cross section of the high capacity base-valve of Fig. 43 including a bleed hole, with each shim-stack in a closed position; and

[0065] Fig. 45 illustrates a second cross section of the base-valve of Fig. 43 with each shim-stack in a closed position.

DETAILED DESCRIPTION

[0066] Fig. 1 illustrates an active suspension actuator system **100** of a vehicle. Actuator **102**, of the system, may be operatively interposed between an unsprung mass **104** of the vehicle (e.g. wheel assembly) and a sprung mass **106** of the vehicle (e.g. vehicle body). Actuator **102** includes a pressure tube **108** that slidably receives piston **4** in an internal cylindrical volume. In some embodiments, such as the embodiment of Fig. 1, the pressure tube **108** may also serve as the housing of the actuator **102**. In other embodiments, the housing may include an outer tube that, at least partially, surrounds the pressure tube **108**. In some embodiments, an annular fluid filled volume between the pressure tube **108** and the housing may serve as a flow passage and/or a gas filled accumulator. U.S. Patent No. 9,689,382 entitled “INTEGRATED ENERGY GENERATING DAMPER”, filed April 8, 2015, the disclosure of which is incorporated herein by reference in its entirety, discloses integrated multi-tube actuators, for example, in Figs. 6 and 7 and associated description.

[0067] The piston **4** divides at least a portion of the internal cylindrical volume of the pressure tube **108** of actuator **102** into a compression volume **7** and an extension volume **8**. In the embodiment of Fig. 1, the piston **4** is attached to the sprung mass **106** by an intervening piston rod **4a** and top mount **110**. In some embodiments, the top mount **110** may effectively be a spring element and is illustrated as a spring element in Fig. 1. In some embodiments the top mount **110** may include a damping element in parallel or in series with the spring element.

[0068] In the embodiment of Fig. 1, the pressure tube **108** is directly or indirectly affixed to the unsprung mass **104** (e.g. wheel assembly), which includes a tire **104a** shown in phantom. Tire **104a** that may travel along a road surface **112** effectively performs as a spring and is represented as spring element **104b**. It should be noted that, alternatively, in some embodiments, the actuator **102** may be inverted such that the pressure tube **108** is directly or indirectly affixed to the top mount **110** while the piston rod **4a** is attached to the unsprung mass **104** and the disclosure is not so limited.

[0069] The embodiment illustrated in Fig. 1 includes a hydraulic machine **114** that includes a first port **114a** and a second port **114b**. Flow passage **116** fluidly connects port **114a** to the compression volume **7** and flow passage **118** fluidly connects port **114b** to the extension volume **8**. The actuator system **100** may include one or more accumulators, such as for example, accumulator **120** that is fluidly connected to flow passage **116** by a branch flow passage, and/or accumulator **122** that is fluidly connected to flow passage **118** by a another branch flow passage. The accumulators may function as fluid storage elements and/or compliance or spring elements. Such accumulators may include a gas spring and/or mechanical spring, e.g. coil spring.

[0070] In the embodiment in Fig. 1, the force applied on the sprung mass **106** in the z direction by the actuator system **100** is determined by the pressure in the compression volume **7** multiplied by the cross sectional area of the piston (i.e. cross section that is transverse to the longitudinal axis of piston rod) minus the pressure in the extension volume multiplied by the difference in the cross sectional area of the piston **4** and the cross sectional area of the piston rod **4a**.

[0071] In some embodiments, the force applied on the sprung mass by the actuator system **100** may be regulated by controlling the pressures in the compression volume **7** and/or extension volume **8**. The pressure in these volumes may be influenced by road-induced disturbances that may cause the piston **4** to move relative to the pressure tube **108**.

Alternatively or additionally, in some embodiments, the pressures in the compression volume **7** and/or extension volume **8** may be influenced by fluid flow into and/or out of those volumes and/or the pressure differential produced by the hydraulic device **114**. In some embodiments, flow into and out of the compression volume **7** may be controlled by, for example, flow control device **16** operatively positioned in flow passage **116,** flow control device **4b** operatively positioned in flow passage **5b** in piston **4,** and/or flow control device **23** operatively positioned in flow passage **23a**.

[0072] In some embodiments, additionally or alternatively, flow into and out of the extension volume **8** may be controlled by flow control device **17** operatively positioned in flow passage **118,** flow control device **4c** operatively positioned in flow passage **5c** in piston **4**, and/or flow control device **24** operatively positioned in flow passage **24a**. In some embodiments, one or more such flow control devices may be omitted, additional flow control devices may be incorporated in various flow channels, and/or multiple flow control devices may be consolidated into a single flow control device.

[0073] In some embodiments, a first set of flow control devices **14** and **15,** and/or a second set of bypass flow control devices **23** and **24** may be used to exchange a quantity of hydraulic fluid between the compression volume **7** and the extension volume **8** without passing through the hydraulic device **114**. Some embodiments may include only the first set of bypass flow control devices. Some embodiments may include only the second set of bypass flow control devices. Some embodiments may include both sets of bypass flow control devices. Some embodiments may include only one bypass flow control device from the first set and/or one bypass from the second set. It is noted that in some embodiments one or more of the flow control devices, such as **4b**, **4c**, **23**, **24**, **16**, and **17**, in Fig. 1, may be bi-directional or unidirectional and the disclosure is not limited in this respect.

[0074] The present disclosure is not limited to a particular number, positioning, or combination of flow control devices such as those shown in Fig. 1. Each of the flow control devices may include one or more damping elements that dissipate at least a portion of the kinetic and/or potential energy of the fluid that passes through the damping element. Damping elements may include, without limitation, orifices, capillary tubes, and/or other flow elements that are configured to viscously damp flow. Alternatively or additionally, each of the flow control devices may include, without limitation, one or more passive or active valves such as, for example, check valves, blow-off valves, poppet valves, pressure balanced blow-off valves, shim stacks, electrically actuated valves, and pressure actuated valves. In some embodiments of the actuator system **100** of Fig. 1, the actuator **102** may be inverted such that the pressure tube **108** maybe attached to the sprung mas **106** by means, for example, of the top mount **110** and the piston rod **4a** may be attached to the unsprung mass **11**.

[0075] In some embodiments, the hydraulic machine **114** of the active suspension actuator system **100** may be operated as a pump to apply an active force, i.e. a force in the direction of motion, on the sprung mass **106**, relative to the unsprung mass **104,** or a passive or resistive force in a direction that opposes the motion, of the sprung mass **106**, relative to the unsprung mass **04**. Alternatively or additionally, the hydraulic machine **114** may be operated as a hydraulic motor to produce a passive or resistive force, i.e. a force opposed to the direction of relative motion between the unsprung mass and sprung mass. The hydraulic machine **114** may be a hydraulic pump that is operated as a hydraulic pump and/or a hydraulic motor. Alternatively, the hydraulic machine **114** may be a hydraulic motor that is operated as a hydraulic motor and/or a hydraulic pump.

[0076] In some embodiments of Fig. 1, flow control device **16** and/or flow control device **17** may be configured to provide a first level of fluid damping [e.g. 0.8-1.2 bar/(L/min) or 1.0-1.1 bar/(L/min)] when the fluid flow rate through the device in a given direction is at a first flow rate [e.g. 2-10 L/min or 6-7 L/min], but a second lower level of damping [e.g. 0.05-0.1 Bar/(L/min)] when the flow rate through the flow control device is a higher second flow rate [e.g. 10-200 L/min or 7-100 L/min]. Additionally, the flow control device **16** and/or flow control device **17** may be configured to include a sufficiently large effective flow area(s) to accept substantially all or all [e.g. greater than 90%, 95%, 98% or 100%] of the flow volume displaced in the extension volume **8** by the piston during extension and/or displaced in the compression volume **7** during compression without exceeding a predetermined maximum pressure or rate of pressure rise in either the compression volume **7** or the extension volume **8** respectively. Additionally, the flow control device(s) **16** and/or **17** may be configured to include a sufficiently large effective flow area(s) to accept pump flow during operation without exceeding a maximum pressure drop [e.g. 3-5 Bar].

[0077] Additionally, in some embodiments, a first set of flow control devices **4b** and **4c**, and/or a second set of bypass flow control devices **23** and **24** may be used to exchange a quantity of hydraulic fluid between the compression volume **7** and the extension volume **8** without passing through (i.e. bypassing) the hydraulic machine **114**.

[0078] These flow control devices may act to equilibrate the pressures in the two chambers when: the differential pressure between them increases rapidly [e.g. 200-300 Bar/sec or 300-500 Bar/sec] and the differential pressure is greater than a threshold value [e.g. 2-10 Bar or 2-30 Bar].

[0079] Fig. 2 illustrates an embodiment of a hydraulic, active suspension actuator system **200** of a vehicle. The active suspension actuator system **200** of Fig. 2 includes an actuator **202,** with a pressure tube **204** that may include a predominantly cylindrical internal volume, a hydraulic machine **114** (e.g. a hydraulic pump, a hydraulic motor) which may be operatively coupled to an electric machine (electric generator, electric motor) (not shown), and a multiplicity of hydraulic fluid flow passages. At least a portion of the internal volume of pressure tube **204** may be separated into a compression volume **7** and an extension volume **8** by a piston **206** that is slidably received in the pressure tube **204**. Piston **206** is operatively coupled to a sprung mass **208** by intervening piston rod **4a**. In some embodiments, the piston rod **4a** may be securely attached to the sprung mass **208** by means of an intervening top mount **210**.

[0080] As in Fig. 1, the embodiment of active suspension actuator system **200** may include a hydraulic machine **114** with a first port **114a** and a second port **114b**. Any of the active suspension actuator systems described herein may include one or more accumulators operatively interposed between the hydraulic machine and the compression volume, and/or between the hydraulic machine and the extension volume. For example, the active suspension actuator system **200** includes one flow-through accumulator **214** operatively interposed between the hydraulic machine **114** and the compression volume **216** as well as a branched accumulator **218** and branched accumulator **220** operatively interposed between the extension volume **222** and the hydraulic machine. It is noted that any one or more of these accumulators, or none of them, may be flow-through accumulators and the disclosure is not so limited.

[0081] The embodiment illustrated in Fig. 2 includes flow passage **224** that fluidly connects compression volume **216**, of the actuator **202,** to volume **214a** in accumulator **214**. In some embodiments, accumulator **214** may include volume **214b** that contains a compressible material such as, for example, nitrogen, air or other appropriate gas. In some embodiments, the gas in volume **214b** may be separated from volume **214b** by floating piston **214c** or contained in a sealed compressible bladder (not shown) in which case the piston **214c** may be unnecessary. In other embodiments, volume **214a** may be separated from volume **214b** by a flexible diaphragm (not shown). The gas or compressible material in volume **20b** may act as a spring element to add compliance to the hydraulic circuit. In some embodiments, the gas or compressible material may be replaced or augmented by a metal spring such as a coil spring.

[0082] Flow passage **226** fluidly connects port **114a** to volume **214a** and flow passages **228** and **230** in combination fluidly connect port **114b** to the extension volume **222**. One or more of the accumulators may function as fluid storage elements and/or compliance or spring elements. Such accumulators may include a gas spring and/or mechanical spring, e.g. coil spring.

[0083] Fig. 3 illustrates the relationship between the pressure drop, ΔPxy, between positions **232** and **234** in flow passage **224** that carries fluid to and from the compression volume **216**. If the flow passage is a largely unobstructed flow passage configured to carry expected maximum flow rates in the passage, i.e. low impedance, the pressure drop at low flow rates may be too small to produce sufficient damping. As used herein, “low flow rate” in a given flow passage in the active suspension actuator system refers to the flow rate, in the given passage, when the piston is moving at a speed of less than 0.2 meters/second in compression and/or extension. As used herein, “high flow rate” in a given flow passage in an active suspension actuator system refers to the flow that occurs when the piston is moving at a speed greater than 0.6 meters/second in compression and/or extension but less than 1 meter/second. As used herein, “very high flow rate” in a given flow passage in an active suspension actuator system refers to the flow that occurs when the piston is moving at a speed greater than or equal to 1 meter per second and less than 2 meters per second. For example, in the embodiment of Fig. 2, the flow rate in passage **224** is low when the piston is moving at less than 0.2 meters per second in compression or extension.

[0084] In the embodiment illustrated in Fig. 2, pressure perturbations, which may be induced in the compression volume **216** by, for example, road disturbances or the hydraulic machine, may induce flow oscillations between positions **232** and **234.** The characteristics of the flow oscillation may be a function of various parameters including, for example, one or more of: the geometry of flow passage **224**, density of the fluid in the flow passage **224**, the compliance of the accumulator **214**, the mass of fluid in compression volume **216**, the mass of the pressure tube **204**, the mass of the unsprung mass **104**, and the spring constant of the tire **104a**. The Inventors have recognized that in some embodiments, the interaction of these components with pressure perturbations in the compression volume **216** may resemble the harmonic behavior of a spring mass oscillator. Under such circumstances, if there is limited fluid damping (i.e. under-damped flow) in certain passages where flow oscillations may be present, such as flow passage **224** and/or **230**, any oscillations that are generated by, for example, a road disturbance or the operation of the hydraulic machine, may persist for an extended period. This may result in objectionable levels and/or duration of vibrations transferred to the vehicle body over an extended period.

[0085] The Inventors have recognized that flow passages that are configured to carry flow at high flow rates with acceptable pressure drops may not function properly at low flow rates. They may have insufficient damping at low flow rates to dissipate flow oscillations. In Fig. 3, the slope angle **300** of the curve **302** is indicative of the impedance of the flow passage **224** of the embodiment in Fig. 2. The flow of fluid in passage **224** may be under-damped under certain operating conditions, such as, when oscillations are induced during low flow conditions. Such oscillations may be induced by interaction between tire **104a** and road surface imperfections and/or discontinuities. Such oscillations may produce fluctuating forces in actuator **202** which may be transmitted to the vehicle body by piston rod **4a** and may be sensed by occupants of the vehicle.

[0086] As discussed above, the Inventors have recognized that damping levels in one or more flow passages in an active suspension actuator system may be insufficient to dissipate certain oscillations at a sufficiently rapid rate. The Inventors have recognized that, in some embodiments, it may be necessary to add damping elements to certain flow passages in order to dissipate oscillations. Such damping elements may be discrete elements such as, for example, orifices or other restrictions or distributed elements that are distributed along a length of pipe or flow passage.

[0087] Fig. 4 illustrates an active suspension actuator system **400**, similar to the embodiment of Fig. 2 but including an added flow control device **402** in flow passage **224**. The flow control device **402** in the embodiment of Fig. 4 includes a damping element **402a** which may include, for example, an orifice or a conduit that induces viscous damping. It should be noted that the damping effect of flow control device **402** is cumulative with any inherent damping present in flow passage **224**. Fig. 5 illustrates the relationship between the pressure drop, ΔPxy, between position **232** and **234** in flow passage **224**, as a function of flow velocity in passage **224**. The increased slope angle **500** of curve **502**, as compared to slope of curve **300**, reflects the increased impedance and damping due to damping element **402a**. It is noted that in the embodiment of Fig. 4, the flow velocity in flow passage **224**, may be directly proportional or effectively directly proportional to the velocity of piston **206** under at least some operating conditions. However, the relation between flow and pressure drop in some embodiment may be nonlinear and the disclosure is not so limited.

[0088] However, the Inventors have recognized that damping levels sufficient to effectively dissipate oscillations may excessively restrict flow under other operating conditions such as high flow rates. This may result in, for example, an undesirable increase in pressure in the compression volume and/or the extension volume. Furthermore, such damping may lead to inefficiencies during active operation. For example, when the active suspension actuator system is operating to apply an active force to the sprung mass, it may be desirable to rapidly pump fluid from one side of the piston to the other (e.g., from the extension chamber to the compression chamber, or vice versa). The damping force provided by the damping element **402a** may resist this rapid transport of fluid. In turn, additional work may be required to overcome the damping force provided by the damping element, leading to a loss in efficiency of the active suspension actuator system during active operation.

[0089] Fig. 6 illustrates an active suspension actuator system **600** similar to the embodiment of Fig. 2 but including a flow control device **602** in flow passage **224**. This flow control device, in addition to damping element **402a**, may include a biased blow-off valve **602b** which may be, for example, a spring-loaded check valve. In some embodiments, when the ΔPxy between positions **232** and **234** reaches a threshold value, valve **602b** may open to permit some of the flow in flow passage **224** to bypass the damping element **402a**.

[0090] Fig. 7 illustrates the relationship between the pressure drop ΔPxy, between position **232** and **234** in flow passage **224**, as a function of flow velocity in passage **224**. Before the valve **602b** opens at point **702** in graph **700**, the relationship between ΔPxy and flow velocity may be the same as the relationship illustrated in Fig. 5 (i.e. slope **500**). But when the pressure drop on compression exceeds a threshold value of ΔPxy_b, valve **602b** opens and the impedance between positions **232** and **234** drops. Once the valve **602b** opens, the overall impedance of the flow control device **602** is represented by slope angle **706**. In the embodiment in Fig. 6, the impedance of the flow control device **602** may be determined as the parallel combination of the branch that includes the damping element **402a** and the impedance (represented by the impedance **602a**) of the branch that includes the blow-off valve **602b**. It should be noted that in some embodiments, elements **602a** and **602b** may be combined into a single device. Alternatively, in some embodiments, elements **602a**, **602b**, and **402a** may be combined into a single device. It should be noted that the impedance of damping element **402a** in Fig. 5 and Fig. 7 is shown to be independent of the direction of the flow, i.e. the slope of line **502** is the same in compression and extension. In some embodiments, the impedance of the flow in one direction (e.g. during compression) may be greater or less than the flow in the opposite direction (e.g. during extension).

[0091] As illustrated in Fig. 7, in the embodiment of Fig. 6, the pressure drop between positions **232** and **234** continues to grow with increasing flow into the compression volume **216**.

In some embodiments, this may be undesirable during, for example, active operation. For example, when the hydraulic machine **114** is operated as a pump to force fluid into the compression volume **216**, for example, to actively extend actuator **14**, all of the flow entering the compression volume **216** may flow through the damping element **402a**. This may be inefficient because excess energy would need to be expended to overcome the impedance of damping element **402a** under high flow conditions.

[0092] Fig. 8 illustrates an active suspension actuator system **800** that includes flow control device **802**. Flow control device **802** includes damping element **402a**, blow-off valve **602b** and impedance **602a,** and blow-off valve **802b** and impedance **802a**. This arrangement may allow damped flow through damping element **402a** in both directions, but also the ability to bypass the damping element **402a** in either direction, with at least a portion of the flow, when the pressure drop across the damping element **402a** exceeds a preset threshold level in a given direction. In some embodiments, the cracking pressure of valve **802b** may be different than the cracking pressure of valve **602b**. In other embodiments, these cracking pressures may be equal or effectively equal. Similarly, in some embodiments, impedance elements **602a** and **802**, that are fluidly in series with the blow-off valves, may provide the same or different resistances to flow.

[0093] Fig. 9 illustrates graph **60** which illustrates the relationship between ΔPxy and flow velocity into and out of the compression volume during compression or extension of the actuator. Both operating conditions when flow is leaving the compression volume (i.e. the actuator is undergoing compression), and when flow is entering the compression volume (i.e the actuator is in extension) are shown. During compression, the slope of segment **902** represents the impedance of the damping element **402a** when both valves **602b** and **802b** are closed. The slope of segment **904** represents the impedance of the flow control device **802** during compression when fluid is flowing out of the compression volume, the pressure differential has surpassed threshold ΔPxy_b, and valve **602b** has opened.

[0094] The slope of segment **906**, which in some embodiments may be different than the slope of segment **902**, represents the impedance of damping element **402a** when fluid is flowing into the compression volume **216**. The slope of segment **908** represents the overall impedance of flow control device **802** when fluid is flowing into the compression volume **216**, the pressure differential has exceeded threshold ΔPxy_c, valve **802b** is open, and valve **602b** is closed. In Fig. 9, the relationships between ΔPxy and velocity in four operating modes are represented by linear or effectively linear segments **902**, **904**, **906**, and **908**. In some embodiments, one or more of these relationships may be non-linear and the disclosure is not so limited.

[0095] It should be noted that in some embodiments the damping element **402a** may be a single component such as, for example, an orifice or other restriction that occupies a limited portion of flow passage **224**. In certain other embodiments the damping element **402a** may include one or more components that are localized in a small section of flow passage **224** and/or distributed along some or the entire length of the flow passage **224**.

[0096] Fig. 10 illustrates an active suspension actuator system **1000** that includes flow control device **1002** operatively interposed in flow passage **1004** between the compliance element **220** and the extension volume **222**. Flow control device **1002** may be incorporated in addition to or instead of flow control device **802**. This flow control device may include damping element **1006**, blow-off valve **1008** and impedance **1010,** and blow-off valve **1012** and impedance **1014**. This arrangement may allow damped flow in both directions, but also the ability to bypass the damping element **1006** in either direction when the pressure drop across the damping element **1006** in a given direction exceeds a preset threshold level in that direction. In some embodiments, the cracking pressure of valve **1008** may be different than the cracking pressure of valve **1012**. In other embodiments, these cracking pressures may be equal or effectively equal. Similarly, in some embodiments, the impedance elements **1006** and **1014** may provide effectively the same or different resistance to flow.

[0097] The flow control device **1002** damps fluid flowing in both directions with damping element **1006**. The damping rate may be reduced when the flow rate leaving the extension volume **222** increases to a point where the pressure drop across flow control device **1002** exceeds a preset threshold value. Under such operating conditions, valve **1012** may open and allow at least some of the flow leaving the extension volume **222** to bypass damping element **1006** thus reducing the overall impedance of the flow control device **1002**. In some embodiments, the flow control device **1002** may also include a second bypass valve **1008**. This arrangement may allow damped flow in both directions, but also the ability to bypass the damping element **1006** in either direction when the pressure drop across the damping element **1006** in a given direction exceeds a preset threshold level in that direction.

[0098] It is noted that, in some embodiments, the impedance of one or more damping elements may be the same in both flow directions while in other embodiments, the impedance of one or more damping elements may depend on the direction of flow. In some embodiments, one or more flow control devices may be used to limit the maximum pressure in the compression volume and/or the extension volume by providing a flow path between these volumes that bypasses the hydraulic machine in the system.

[0099] Fig 11 illustrates an exemplary embodiment of actuator piston **1099** attached to piston rod **4a** that includes a flow control device **1104b** and a flow control device **1104c.** Flow control device **1104b** may include an inlet port **1080** and outlet port **1081** and pressure balance blow-off valve (PBOV) **1083**. The inlet port **1080** of PBOV **1083** is exposed to the pressure of the fluid in the extension volume (not shown) and the outlet port **1081** is exposed to the pressure of the fluid in the compression volume (not shown).

[00100] The PBOV **1083** includes a two-position valve **1114** that is actuated by the force differential between the force applied by piston **1085a** that is slidably received in cylinder **1086a** and the force applied by piston **1085b** that is slidably received in cylinder **1086b**.

[00101] In the embodiment of Fig. 11, piston **1085b** has a larger diameter than piston **1085a**. When the pressure in volume **1087b** is equal to the pressure in volume **1087a**, the resulting force imbalance on the two-position valve **1084** keeps the valve **1104** in the closed position. In some embodiments, PBOV **1083** may also include a travel stop **1085c** that limits the travel of piston **1085a** in cylinder **1086a**.

[00102] Flow passages **1118a** and **1088b** fluidly connect the volume **1087a** to the inlet port **1080**. Flow passages **1088c** and **1088b** fluidly connect volume **1087b** to inlet port **1080**. The pressure in gas volume **1089** biases the PBOV **1083** in the closed position. The gas volume **1109** is separated from the liquid in volume **1087b** by floating piston **1087c**. In some embodiments, additionally or alternatively, the bias force provided by the gas volume **1109** may be provided by a coil spring (not shown) or other spring element.

[00103] In some embodiments, pistons **1085a** and **1085b** may be of equal diameter. Flow passage **1088c** may also include a hydraulic an impedance element **1090** that in conjunction with the compliance of volume **1087b** may form a low pass hydraulic filter that blocks or effectively block high frequency changes in pressure at the inlet port **1080** from reaching volume **1087b**. In some embodiments, the hydraulic impedance **1090** may be tuned to block, or effectively block, frequencies above a threshold value. In some embodiments, the threshold frequency may be preselected to be in the range of 12-15 Hz. Alternatively, in some embodiments, the threshold frequency may be preselected to be in the range of 10-20 Hz. The threshold frequency may be selected to be in other frequency ranges as the disclosure is not limited in this respect. During operation, in some embodiments, if the rate of pressure increase in the extension volume and/or at inlet port **1080** is above a predetermined threshold value, the pressure in volume **1087a** may increase more rapidly than the pressure in volume **1087b** due to hydraulic filter formed by impedance **1090**. If the rate of pressure rise continues for a sufficient period, then the force bias on the PBOV **1083** due to the gas pressure in volume **1089** and/or the piston size differential of the two-position valve may be overcome and the two-position valve may move to its open position. As a result, fluid flow may then be established between the inlet port **1080** and the outlet port **1081**. The operation of the PBOV **1083** then acts to mitigate the effect of the rapid increase in the pressure in the extension volume (above a certain rate of increase) while allowing the more gradual rate of pressure build-up that may result from, for example, the operation of the hydraulic machine to be effectively applied to the piston **1100**. When the rate of pressure increase is slow, whether caused by the hydraulic machine or road disturbance, the pressures in volume **1087a** and **1087b** effectively track each other. Because piston **1085b** is larger, the PBOV **1083** may remain in a closed position under such an operating condition.

[00104] In some embodiments, piston **1100** may also include an additional flow control device **1104c**. Flow control device **1104c** may include an inlet port **1100** and outlet port **1101** and pressure balanced blow-off valve (PBOV) **1103**. The inlet port **1100** is exposed to the pressure of the fluid in the compression volume (not shown) and the outlet port **1101** is exposed to the pressure of the fluid in the extension volume (not shown). The PBOV **1103** may include a two-position valve **1104** that is actuated by the force differential between the force applied by piston **1105a** and the force applied by piston **1105b**. As in the case of PBOV **1083**, the low pass filter formed by using **1110** may be tuned to block, or effectively block, frequencies above a threshold value. In some embodiments, the threshold frequency may be preselected to be in the range of 12-15 Hz. Alternatively, in some embodiments, the threshold frequency may be preselected to be in the range of 10-20 Hz. The threshold frequency may be selected to be in other frequency ranges as the disclosure is not so limited. During operation, in some embodiments, if the rate of change of pressure increase in the compression volume is above a predetermined threshold value, the pressure in volume **1107a** may increase more rapidly than the pressure in volume **1107b**. If the rate of pressure rise continues for a sufficient period, then the force bias on the PBOV **1103** may be overcome and the two-position valve may move, fully or partially, to its open position and flow may be established between the inlet port **1100** and the outlet port **1101**. The operation of the PBOV **1103** dissipates the effect of the rapid increase in the pressure in the compression volume (above a certain rate of increase) while allowing the more gradual buildup of pressure that may result from, for example, the operation of the hydraulic machine.

[00105] In some embodiments, the flow control device **1104c** may also include a biased valve, such as for example, a spring biased check valve **1120**. Check valve **1120** may effectively prevent flow between the inlet port **1100** and outlet port **1101** even when PBOV **1103** is in a partially or fully open position. Accordingly, flow between the inlet port **1100** and the outlet port **1101** may be prevented or effectively prevented unless both biased check valve **1120** and the two-position valve **1104** are at least partially open.

[00106] Accordingly, flow control device **1104c** may be used to prevent or mitigate rapid rise in pressure of the compression volume when the pressure of the compression volume is above a preset pressure threshold established by the pressure/force bias of valve **1120** while also allowing slow and/or rapid increase in pressure when compression volume pressure is below that threshold. Flow control device **1104c** may be used to, for example, prevent or limit the rapid rise of pressure in the compression volume when the pressure is above a threshold value while allowing rapid pressure rise, for example during wheel control, at frequencies in the range of 10-15 Hz, when the pressure is below the threshold value. Accordingly, the pressure in the compression volume may increase at a given rate without triggering the PBOV when the total pressure is below a threshold value. Alternatively, when the pressure in the compression volume is above the threshold pressure, valve **1220** may open, in which case the PBOV would be trigger if there is a rapid pressure increase.

[00107] Fig. 12 shows graph **1115** that illustrates an example of the PBOV **1103**, of the embodiment of the flow control device in Fig. 11, may operate when valve **1120** is open or effectively open. At low frequencies, PBOV **1103** may remain closed or effectively closed, as shown by line **1115a**, regardless of the pressure at inlet port **1100**. The PBOV **1103** may be tuned by, for example, selecting the appropriate low pass hydraulic filter formed by using **1110.** In some embodiments, a PBOV may open when inlet port **1100** is exposed to a pressure oscillation of approximately 12 Hz for a sufficient duration (line **1115b**). The transition (curve **1115c**) from curve **1115a** to **1115b** depends on, for example, the design parameters of the hydraulic filter and the size of pistons **1105a** and **1105b**.

[00108] In some embodiments, the bias on check valve **1120** may be an absolute bias so that it may be activated by, for example, the total pressure at inlet port **1100**. For example, if the force bias on valve **1120** is tuned so that the valve **1120** opens at a total pressure of 50 psi, then the valve **1120** would open regardless of how that pressure is reached. In the above example, if the pressure of the compression volume is 40 psi, valve **1120** would open if there is an additional pressure increase of 10 psi due to, for example, a road disturbance. However, if the same road disturbance were to occur when the pressure in the compression chamber was, for example, 20 psi, the valve **1120** would remain closed. The PBOV **1103** in the flow control device **1104c** of the embodiment in Fig. 13 is an absolute bias PBOV because flow between ports **1100** and **1101** may occur when the pressure at port **1100** is sufficient to overcome the bias of valve **1120**.

[00109] Fig. 13 illustrates a piston **1300** with a flow control device **1304d**with a relative bias PBOV **1303**. In the embodiment of Fig. 13, cylinder **1304a** includes a travel stop **1305** that prevents piston **1304b** from traveling beyond a preset point in cylinder **1304a**. Spring element **1310** (e.g. a coil spring), that is anchored to cylinder **1311a** applies a bias closing force on the two-position valve **1312**. The two-position valve **1312** is configured to remain in the closed position until the net force on pistons **1311b** and **1304b** exceeds the bias force applied by spring element **1310**. In the configuration of Fig. 13, piston **1304b** and **1311b** effectively have the same diameter and the PBOV **1303** is therefore insensitive to slowly changing pressures at inlet port **1310**. This is because pressures in both volumes **1311c** and **1304c** track the pressure at inlet port **1320** when the pressure at port **1320** changes slowly. The two-position valve **1312** in Fig. 13 may be triggered when a rapidly changing pressure rise (e.g. greater than 10 Hz, 12 Hz, or greater than 12 Hz but less than 1000 Hz), is blocked or effectively blocked by the filter, formed by using impedance **1330** in conjunction with the compliance of volume **1311c,** from reaching volume **1311c,** but reaches cylinder

1304c. The increased pressure in **1304c** may create a force imbalance that overcomes the bias imposed by spring element**1310**. The two-position valve **1312** of the embodiment in Fig. 13 is remains effectively closed when the same pressure reaches both cylinders in the PBOV **1303**. PBOV **1303** is a relative biased device because the flow between port inlet port **1320** and outlet port **1321** depends on the differential pressure between the cylinders **1304c** and **1311c** and independent of the absolute pressures at the ports.

[00110] The valves in flow control devices illustrated in Figs. 6, 8, 10, 11 and 13 are shown as passive or pressure actuated valves. Alternatively or additionally, in some embodiments electrically activated valves may be used in one or more such bypass flow control devices. Such electrically controlled valves may be activated based on data from various sensors, such as for example, pressure sensors that sense the pressure in the compression and/or extension volumes. Alternatively, such electrically controlled valves may be activated based on information from, for example, accelerometers that are configured to measure the acceleration of the unsprung mass. Such acceleration may be the relative acceleration between the sprung mass and the unsprung mass.

[00111] In the embodiment of Fig. 1, one or more of the flow control devices **4b, 4c, 16, 17, 23** and **24** may be tuned by selecting appropriate parameters. Fig. 14 illustrates an example of a special case of the embodiment of Fig. 1. Active suspension actuator system **1400** includes actuator **1401** with pressure tube **1401a**. The pressure tube includes an internal volume that is divided into a compression volume **1401b** and extension volume **1401c** by piston **1410**. Piston **1410** includes flow control devices **1410a** and **1410b**. In the exemplary embodiment illustrated in Fig. 14, the overall damping coefficient of flow control device **1411** may be selected to appropriately damp oscillation that may result at any given operating condition. The low flow velocity damping coefficient of outward flowing (from the compression volume) damping element **1402b** may be the same as or different than the damping coefficient of the inward flowing (to the compression volume **1401b**) damping element **1402a**.

[00112] The cracking pressure of valves **1402c** and **1402d** and their associated impedances represented as **1402e** and **1402f**, respectively, may be selected to trim the amount of pressure drop and damping at higher flow velocities passing through flow control device

1411. The check valves **1402g** and **1402h** allow different damping coefficients to be used for the flow to and from the compression volume at low flow rates. The hydraulic low pass filter and the force bias of the PBOVs (i.e. flow control devices **1410a** and **1410b**) may be selected to determine at what frequencies and pressure levels the PBOVs **1410a** and **1410b** may activate.

[00113] By appropriately selecting such parameters, the active suspension actuator systems of, for example, the embodiments illustrated Fig. 1, Fig. 6, Fig. 8, and Fig. 14, may operate to:

(1) control flow between the actuator (i.e. the compression volume and/or the extension volume) and a compliance element with a first flow control device,

(2) damping fluid flow velocity oscillations in the first flow control device, induced, for example, by low velocity wheel events, at a first level for flow velocities below a threshold magnitude,

(3) reduce the level of damping for flow velocities that are greater than the threshold magnitude,

(4) produce a pressure differential across the piston of the actuator by using the hydraulic machine as a pump, and

(5) at least partially discharge the pressure differential across the piston by using a second flow control device that bypasses the hydraulic machine and that is activated when the rate of increase pressure differential across the piston is greater that a preset threshold value.

[00114] Fig. 15 illustrates an active suspension actuator system **1501** of a vehicle. Actuator **1502**, of the system, may be operatively interposed between an unsprung mass **1511** of the vehicle (e.g., wheel assembly) and a sprung mass **1509** of the vehicle (e.g., vehicle body). Actuator **1502** includes a pressure tube **1503** that slidably receives piston **1504** in an internal cylindrical volume. In some embodiments, such as the embodiment of Fig. 15, the pressure tube **1503** may also serve as a housing of the actuator **1502**. In other embodiments, the housing may include an outer tube that, at least partially, surrounds the pressure tube.

[00115] In the embodiment illustrated in Fig. 15, piston **1504**, which includes two pressure balanced blow-off valves (PBOVs), divides the internal cylindrical volume of the pressure tube **1503** of actuator **1502** into a compression volume **1507** and an extension volume **1508**. In the embodiment of Fig. 15, the piston **1504** is attached to the sprung mass **1509** by an intervening piston rod **1504a** and top mount **1510**. In some embodiments, the top mount may effectively be a spring element and is illustrated as a spring element in Fig. 15. In some embodiments the top mount may include damping element in parallel or in series with the spring element.

[00116] In the embodiment of Fig. 15, the pressure tube **1503** is attached to the unsprung mass **1511** (e.g. wheel assembly), which includes a tire **1512** (shown in dashed line). Tire **1512** that travels along a road surface **1513** effectively performs as a spring and is represented as spring element **1512a**. The embodiment illustrated in Fig. 15 includes a hydraulic machine **1522** that includes a first port **1522a** and a second port **1522b**. Flow passage **1521a** fluidly connects port **1522a** to the compression volume **1507** and flow passage **1521b** fluidly connects port **1522b** to the extension volume **1508**. The actuator system **1501** may include one or more accumulators, such as for example, accumulator **1518a** that is fluidly connected to flow passage **1521a** by a branch flow passage, and/or accumulator **1518b** that is fluidly connected to flow passage **1521b** by another branch flow passage. Other types of accumulators, such as flow-through accumulators may be used, as the disclosure is not so limited. The accumulators may function as fluid (e.g. hydraulic fluid or other liquid) storage elements and/or compliance or spring elements.

[00117] In the embodiment illustrated in Fig. 15, the net force applied on the sprung mass **1509** by the actuator system **1501** is equal to the net force applied to the piston **1504** by the fluid (e.g. hydraulic fluid or other liquid) in the compression volume , in a direction that parallel to the longitudinal axis of the piston rod, minus the net force applied to the piston **1504** by the fluid (e.g. hydraulic fluid or other liquid) in the extension volume.

[00118] In some embodiments, the force applied on the sprung mass **1509** by the actuator system **1501** may be regulated by controlling the pressures in the compression volume **1507** and/or extension volume **1508**. The pressure in these volumes may be influenced by road-induced disturbances that may cause the piston **1504** to move relative to the pressure tube **1503**. Alternatively or additionally, in some embodiments, the pressures in the compression and/or extension volumes may be influenced by fluid flow into and/or out of those volumes. In some embodiments, flow into and out of the compression volume **1507** may be controlled by, for example, flow control device **1516** operatively positioned in flow passage **1521a** and/or PBOV **1505a** or **1505b**.

[00119] In some embodiments, flow into and/or out of the extension volume **1508** may be controlled by flow control device **1517** operatively positioned in flow passage **1521b** and/or PBOV **1505b**. In some embodiments one or more such flow control devices may be omitted, additional flow control devices may be incorporated in various flow channels, and/or multiple flow control devices may be consolidated into a single flow control device. For example, in some embodiments, only a single PBOV may be attached to piston **1504**. In some embodiments, the operation of one or more PBOVs, **1505a** and **1505b**, may be replaced by PBOVs that are located external to the pressure tube. For example, such PBOVs may be incorporated in or replace, for example, the flow control devices **1523** and **1524**. The present disclosure is not limited to a particular number, positioning, or combination of flow control devices (e.g., PBOVs) such as those shown in Fig. 15.

[00120] In some embodiments, the hydraulic machine **1522** of the active suspension actuator system **1501** may be operated as a pump to apply an active force, i.e. a force in the direction of motion, on the sprung mass **1509**, relative to the unsprung mass **1511,** or a passive or resistive force in a direction that opposes the motion, of the sprung mass **1509**, relative to the unsprung mass **1511.** Alternatively or additionally, the hydraulic machine **1522** may be operated as a hydraulic motor to produce a passive or resistive force, i.e. a force opposed to the direction of motion, on the sprung mass, relative to the unsprung mass. The hydraulic machine **1522** may be a hydraulic pump that is operated as a hydraulic pump and/or a hydraulic motor. Alternatively, the hydraulic machine **1522** may be a hydraulic motor that is operated as a hydraulic motor and/or a hydraulic pump.

[00121] Fig. 16 shows an embodiment of actuator piston **1526** attached to piston rod **1504a** that includes a flow control device **1514** (shown in dashed line) and flow control device **1515** (shown in dashed line). Flow control device **1514** may include an inlet port **1530** and outlet port **1531** and pressure balanced blow-off valve (PBOV) **1533**. The inlet port **1530** of PBOV **1533** is exposed to the pressure of the fluid in the extension volume (not shown) and the outlet port **1531** is exposed to the pressure of the fluid in the compression volume (not shown).

[00122] The PBOV **1533** includes a valve **1534** that is actuated by the force differential between the force applied by piston **1535a** that is slidably received in cylinder **1536a** and the force applied by piston **1535b** that is slidably received in cylinder **1536b**. In some embodiments valve **1534** may be a continuously variable valve as indicated in Fig. 16. Alternatively, valve **1534** may be a two position valve with only no-flow and full-flow positions.

[00123] In the embodiment of Fig. 16, piston **1535b** is larger than piston **1535a**. When the pressure in volume **1537b** is equal to the pressure in volume **1537a**, the resulting force imbalance on the two-position valve **1534** keeps the valve in the closed position. In some embodiments, PBOV **1533** may also include a travel stop **1535e** that limits the travel of piston **1535a** in cylinder **1536a**.

[00124] Flow passages **1538a** and **1538b** fluidly connect the volume **1537a** to the inlet port **1530.** Flow passages **1538c** and **1538b** fluidly connect volume **1537b** to inlet port **1530.** The pressure in gas volume **1539** biases the PBOV **1533** in the closed position. The gas volume **1539** is separated from the liquid in volume **1537b** by floating piston **1537c**. In some embodiments, additionally or alternatively, the bias force provided by the gas volume may be provided by a coil spring (not shown) or other spring element.

[00125] In some embodiments, pistons **1535a** and **1535b** may be of equal diameter. Flow passage **1538c** may also include a hydraulic low pass filter dissipative element **1540** that may block or effectively block high frequency changes in pressure at the inlet port **1530** from reaching volume **1537b**. The compliance of chamber **1537b** along with the inlet restive element **1540** create a hydraulic low-pass filter that limits the rate of pressure change in chamber **1537b** in reference to the pressure change experienced at inlet **1530**. In some embodiments, the hydraulic low pass filter which is made up of the compliance element **1539** and the dissipative element **1540** may be tuned to block, or effectively block, pressure fluctuations at frequencies above a threshold value. In some embodiments, the threshold frequency may be preselected to be in the range of, for example 12-15 Hz or 5-20 Hz. The threshold frequency may be selected to be in other appropriate frequency ranges as the disclosure is not limited in this respect. In some embodiments, the low pass filter may be tuned to provide sufficient backpressure on valve **1534** to prevent the valve **1534** from opening for pressure fluctuations rates that are associated with, for example, body-control operating pressure rates. In some embodiments, these rates may be, for example, below 5 Hz in frequency and full force amplitude (e.g. up to 6000 N). Inventors have recognized that in some embodiments, the performance and reliable operation of a PBOV may be improved when the piston area exposed to the filtered volume (e.g. **1537b**) is greater than the piston area exposed to the unfiltered volume (e.g. **1537a**). In some embodiments this ratio may be in the range of 1.05 to 1.3. Other area ratios may be selected and the disclosure is not limited in this respect.

[00126] During operation, in some embodiments, if the rate of pressure increase in the extension volume (not shown) and/or at inlet port **1530** is above a predetermined threshold value, the pressure in volume **1537a** will increase more rapidly than the pressure in volume **1537b** due to the hydraulic filter that is formed by the dissipative element **1540** and the compliance of volume **1537b**. If the rate of pressure rise continues for a sufficient period, then the force bias on the PBOV **1533** will be overcome and the two position valve will move to its open potion and flow will be established between the inlet port **1530** and the outlet port **1531**. The operation of the PBOV **1533** then acts to mitigate the effect of the rapid increase in the pressure in the extension volume (above a certain rate of increase) while allowing the more gradual rate of pressure increase that may result from, for example, the operation of the hydraulic machine, to be effectively applied to the piston **1504**. The dissipative element **1540** may be, for example, a fluid restriction, orifice, or other element that provides resistance to flow in order to act as a low pass filter dissipative element in response to pressure fluctuations. In some embodiments an orifice restriction may be in the range of 0.1 to 2 mm hydraulic equivalent but may be dependent on the volumetric stiffness of chamber **1537b**.

[00127] In some embodiments, piston **1504** may also include an additional flow control device **1515.** Flow control device **1515** may include an inlet port **1550** and outlet port **1551** and balance pressure blow-off valve (PBOV) **1553**. The inlet port **1550** is exposed to the pressure of the fluid in the compression volume (not shown) and the outlet port **1551** is exposed to the pressure of the fluid in the extension volume (not shown). The PBOV **1553** may include a two-position valve **1554** that is actuated by the force differential between the force applied by piston **1555a** and the force applied by piston **1555b**.

[00128] As in the case of PBOV **1533,** a low pass filter, which may include dissipative element **1560** and compliance element **1567c** may be tuned to block, or effectively block, frequencies above a threshold value. In some embodiments, the threshold frequency may be preselected to be in the range of, for example, 12-15 Hz, or 5-20 Hz. The threshold frequency may be selected to be in other frequency ranges as the disclosure is not so limited. During operation, in some embodiments, if the rate of change of pressure increase in the compression volume is above a predetermined threshold value, the pressure in volume **1557a** (i.e. unfiltered volume) may increase more rapidly than the pressure in volume **1557b** (i.e. filtered volume). If the rate of pressure rise continues for a sufficient period, then the force bias on the PBOV **1553** will be overcome and the two position valve may open, fully or partially, and flow may be established between the inlet port **1550** and the outlet port **1551**. The operation of the PBOV **1553** may act to mitigate shock of rapid pressure increase in the compression volume (by discharging the pressure build-up when rate of pressure increase is above a certain threshold value) while allowing the more gradual pressure build-up that may result from, for example, the operation of the hydraulic machine as a pump.

[00129] The flow control device **1515** may also include, for example, a biased valve, such as for example, a spring biased check valve **1560a**. Spring biased check valve **1560a** may effectively prevent flow between the inlet port **1550** and outlet port **1551** even when PBOV **1553** is in an open position due to a rapid pressure event. Accordingly, flow between the inlet port **1550** and the outlet port **1551** may be prevented or effectively prevented unless both of the spring biased check valve **1560a** and the two-position valve **1553** are at least partially open.

[00130] Accordingly, flow control device **1515** may be used to prevent or curtail rapid pressure increase in the compression volume when the pressure is above a preset pressure threshold by the pressure/force bias of valve **1560a** while also allowing rapid increase in pressure at compression volume pressures below that threshold. Flow control device **1515**

may be used to, for example, prevent or limit the rapid rise of pressure in the compression volume when the pressure is above a threshold value while allowing rapid pressure rise, for example during active wheel control, at frequencies, in the range of 10-15 Hz , when the pressure is below the threshold value. Accordingly, an increased pressure in the compression volume may occur without triggering valve **1560a**, regardless of the rate of increase of pressure when the total pressure is below the bias pressure of the valve. Alternatively, in some embodiments when the pressure differential between the compression and extension volumes is above the threshold pressure, a valve may open, in which case the PBOV may react to a rapid differential pressure rise.

[00131] In some embodiments, the piston may include only an extension PBOV (i.e. an EPBOV) that, when open or effectively open, may discharge fluid from the extension volume to the compression volume or a compression PBOV (i.e. a CPBOV) that, when open or effectively open, may discharge fluid from the compression volume to the extension volume.

[00132] Fig. 17 shows an embodiment **1527** of actuator piston **1527a** attached to piston rod **1504a** that includes a flow control device **1515** (shown in dashed line). Flow control device **1515** may include an inlet port **1563a** and outlet port **1563b** and pressure balanced blow-off valve (PBOV) **1563**. The inlet port **1563a** of PBOV **1563** is exposed to the pressure of the fluid in the compression volume (not shown) and the outlet port **1563b** is exposed to the pressure of the fluid in the extension volume (not shown).

[00133] The PBOV **1563** includes a valve **1564** that is actuated by the net force applied by piston **1565a**, piston **1565b**, and spring **1565d.** Piston **1565a** is slidably received in cylinder **1562a** and the force applied by piston **1565a** is determined by the product of pressure in volume **1567a** and the cross sectional area of the cylinder **1562a** that is transverse to the longitudinal axis of the cylinder **1562a**. Piston **1565b** is slidably received in cylinder **1562b** and the force applied by piston **1565b** is determined by the product of pressure in volume **1567b** and the cross sectional area of the cylinder **1562b** that is transverse to its longitudinal axis of the cylinder **1562b**. The motion of the assembly made up of piston **1565b,** valve **1564,** and piston **1565a** is limited by mechanical stop **1565e**. In some embodiments valve **1564** may be a continuously variable valve as indicated in Fig. 17.

Alternatively, valve **1564** may be a two position valve with only no-flow and full-flow positions. In some embodiments, if the diameter of **1565b** is larger than the diameter of **1565a**, when the pressure in volume **1567b** is equal to the pressure in volume **1567a**, the resulting force imbalance on valve **1564** helps keep the valve **1564** in the closed position and piston **1565b** seated against mechanical stop **1565e**.

[00134] Flow passages **1561a** and **1561b** fluidly connect the volume **1567a** to the inlet port **1563a**. Flow passages **1561c** and **1561b** fluidly connect volume **1567b** to inlet port **1563a**. Volume **1567c** is fluidly connected to the outlet port **1563b** by flow passages **1561d** and **1561e** and separated from volume **1567b** by piston **1567c**. In some embodiments, the position of piston **1565c** is determined the force resulting from the pressures in volumes **1567b** and **1567c** and the force applied by spring **1565d**.

[00135] As in the case of PBOV **1533,** shown in Fig. 16, the low pass filter dissipative element **1560** in flow passage **1561c** in conjunction with the compliance of chamber **1567b** achieved by spring **1565f** may be sized or tuned to block, or effectively block, frequencies and/or rates of pressure rise above a threshold value. For example, in some embodiments, the threshold frequency may be preselected to be in the range of 12-15 Hz or 5-20 Hz. The threshold frequency may be selected to be in other frequency ranges as the disclosure is not so limited.

[00136] In the embodiment of Fig. 17, if the pressure at the inlet port **1563a** (i.e. pressure in the compression volume) increases slowly relative to the pressure at the outlet port **1563b** (i.e. pressure in the extension volume), the pressure in volume **1567a** will track or effectively track the pressure at port **1563a**. Concurrently, fluid will flow through flow passages **1561b** and **1561c** so that the pressure in volume **1567b** will also equilibrate with the pressure at port **1563a**. If the rate of pressure change at port **1563b** relative to pressure at port **1563b** is lower than hydraulic filter dissipative element **1560** is configured to block or effectively block, the pressure in volume **1567b** will also track the pressure in volume **1567a**. As a result, the force balance on valve **1564** would remain unchanged or effectively unchanged and the valve would remain in the closed position.

[00137] Similarly, if the pressure at port **1563b** were to drop slowly, piston **1565b** would move upwards but fluid would flow from the inlet port through flow passages **1561b**

and **1561c**. As a result, the force balance on valve **1564** would remain unchanged or effectively unchanged and the valve would remain in the closed position.

[00138] During operation, in some embodiments, if the rate of change of pressure increase at port **1563a** relative to port **1563b** is above a predetermined threshold value, the pressure in volume **1567a** may increase more rapidly than the pressure in volume **1567b** due to the blocking effect of the filter that includes dissipative element **1560.** In some embodiments, the effective flow restriction of element **1560** may be 10 times or more restrictive than the inherent hydraulic restriction of flow path **1561a**. If the rate of pressure rise continues for a sufficient period, then the force bias on the PBOV **1563** that keeps the valve **1564** in the closed position may be overcome and the valve **1564** may move fully or partially to its open position and flow may be established between the inlet port **1563a** and the outlet port **1563b**. The operation of the PBOV **1563** may act to mitigate the effect of the rapid increase in the pressure in the compression volume (above a certain rate of increase).

[00139] Similarly if the pressure at port **1563b** were to drop rapidly relative to the pressure at port **1563a**, fluid would flow out of volume **1567c** causing the piston **1565c** to move in the positive y direction. But because of the blocking effect of filter dissipative element **1560** at high rates of flow increase in flow passage **1561c**, the pressure in volume **1567b** may fall. As a result, the force balance on valve **1564** to shift such that it may move in the positive y direction to its fully open or partially open position.

[00140] The piston, piston/rod interface, the EPBOV (extension side pressure balanced blow off valve) and CPBOV (compression side pressure balanced blow off valve) of Fig. 15 are shown in greater detail in Fig. 18. An embodiment of a system with two flow control devices external to the piston **1504** is illustrated in Fig. 18. Piston **1504** is fixedly attached to extension stud **1504b** at approximately the midpoint of the stud. The piston **1504** separates the internal volume of the pressure tube **1503** (shown in dashed line) The extension stud **1504b** has a proximate end, which is fixedly attached to piston rod **1504a**, and a distal end. Piston **1504** includes a first face **1504c** that is at least partially exposed to the fluid in the compression volume **1507** and a second face **1504d** that is at least partially exposed to the fluid in the extension volume **1508**. In some embodiments, piston **1504** includes at least one flow passage **1504e** that fluidly connects an outlet port **1504f** in the first face of piston **1504** with an inlet port **1504g** in the second face **1504d** of the piston **1504**.

[00141] In some embodiments, piston **1504** may include at least one flow passage **1504h** that fluidly connects an inlet port **1504i** in the first face of piston **1504** with an outlet port **1504j** in the second face **1504d**. In some embodiments, a sealing washer **1504k** may be used to selectively seal or selectively effectively seal one or more outlet ports (e.g. outlet port **1504f**) in the first face **1504c** of piston **1504**. In some embodiments, a sealing washer **1504l** may be used to selectively seal or selectively effectively seal the one or more outlet ports (e.g. outlet port **1504j**) in the second face **1504d**. The interface between the sealing washers **1504k** and **1504l** and the piston faces **1504c** and **1504d**, respectively, may be radially extending flat surfaces. However, other appropriately shaped mating surfaces may be used as the disclosure is not so limited. In some embodiments, the sealing washers may be annular disks as illustrated in Fig. 18. Alternatively, appropriately shaped shim-stacks or other sealing elements may be used as the disclosure is not so limited.

[00142] In the embodiment of Fig. 18, the sealing washers **1504k** and/or **1504l** may be biased against the sealing surfaces of faces **1504c** and/or **1504d** by one or more hydraulic and/or mechanical forces. Fig. 19 illustrates a force balance on sealing washer **1504l**, when it is in its seated position, as illustrated in Fig. 18. The illustrated embodiment of the sealing washer **1504l** is an annular disk with a central axis **1581** that coincides with the longitudinal axis of piston rod **1504a** and/or extension stud **1504b**. The central opening **1582** of the sealing washer **1504l** may include a central cylindrical portion **1583** that may be configured to receive the extension stud **1504b** and a surrounding bearing. The balance of the forces acting on sealing washer **1504l** determine whether the washer will remain in a seated position against piston face **1504d** or lift off that seated position. The washer may maintain a hydraulic seal when the sealing contact force is positive and non-zero. In the embodiment of Fig. 19, if the net force in the negative y direction is greater or equal to the net force in the positive y direction, the sealing washer **1504l** may remain seated against the mating surface of the piston. If there is a net force in the positive y direction, the sealing washer **1504l** may lift off the mating surface of the washer, allowing fluid to flow from the compression volume **7**, through flow passage **1504h,** to the extension volume **1508**.

[00143] The embodiment of the dual PBOV system, shown in Fig. 18, may include a fixed proximate spring perch **1571** affixed or secured to rod **1504a** and distal spring perch **1573** affixed or secured to the distal end of extension stud **1504b**. Fixed spring **1571a** is operatively interposed between proximate fixed spring perch **1571** and sealing washer **1504l**. Fixed spring **1571a** is configured to apply a mechanical force on sealing washer **1504l**. Similarly, in some embodiments, fixed spring **1572a** is operatively interposed between distal fixed spring perch **1572** and sealing washer **1504k**. In some embodiments, fixed spring **1572a** is configured to apply a mechanical bias force on sealing washer **1504k**.

[00144] The embodiment illustrated in Fig. 18 may also include proximate floating perch **1573** and distal floating perch **1574.** Floating springs **1573a** and **1574a** are operatively disposed between sealing washers **1504k** and **1504l** and floating perches **1574** and **1575,** respectively. The combination of bias forces applied by floating spring **1573a** and fixed spring **1571a**, unless overcome by an opposing net hydraulic force, may maintain the sealing washer **1504l** seated against the second face **1504d** or piston **1504**. Similarly, in some embodiments, floating spring **1574a** and fixed spring **1572a**, unless overcome by an opposing net hydraulic force, may maintain sealing washer **1504k** seated against the first face **1504e** of piston **1504b**.

[00145] In the embodiment illustrated in Fig. 18 the compression volume **1507** is fluidly connected to manifold **1575** by means of recessed pocket **1504m**, inlet port **1504i**, and flow passages **1504h**, **1576a**, and **1576b** and flow restriction **1576c**. Flow restriction **1576c** may be, for example, an orifice, a laminar flow element (e.g. a tube where the flow is laminar during operation) or other appropriate component that induces a hydraulic pressure drop. Slow changes in pressure of the compression volume **1507** may reach the manifold **1575** without or effectively without mitigation. However, because of the resistance of the flow passages and especially flow restriction **1576c**, rapid increases in pressure of compression volume **1507** may be attenuated when they reach the manifold **1575**. The pressure differential between the pressure in manifold **1575** and the pressure in the extension volume **1508** results in a net hydraulic force being applied to the floating perch **1573** in the longitudinal direction. Therefore, for slowly changing compression volume **1507** pressures, the pressure in the manifold **1575** may track or effectively track the pressure in the compression volume **1507**. Therefore, for slowly changing pressures in the compression volume **1507**, the net force on the floating perch **1573** may compress floating spring **1573a** thus increasing the mechanical force on the sealing washer **1504l**. The increased mechanical force counteracts the increased hydraulic force on the sealing washer **1504l** that may lift the sealing washer **1504l** from its seated position. Therefore, PBOV **1576,** may resist opening and at least partially discharging the pressure differential due to increased pressure in the compression volume if those changes are slow, such as for example between 0 Hz and 4 Hz.

[00146] However, for rapidly changing compression volume pressures, the intervening resistance between the compression volume **1507** and the manifold **1575** may block or effectively block the manifold pressure from tracking the compression volume pressure. Without sufficient increase in the pressure of manifold **1575** there may not be sufficient pressure differential applied to the floating spring perch **1573** to cause the floating spring to compress or compress sufficiently. Without the additional mechanical force applied by the floating spring **1573a**, the sealing washer **1504l** may lift off its seated position and allow the pressure differential between the compression volume **1507** and the extension volume **1508** to at least partially discharge.

[00147] In a similar fashion, the PBOV **1577** may be used to discharge the differential pressure between the extension volume **1508** and the compression volume **1507** when the pressure increase in the extension volume **1508** is rapid. However, the PBOV **1577** may allow for the pressure in the extension volume **1508** to be increased slowly, for example by the operation of the hydraulic machine **1522** of Fig. 15, to the maximum pressure capacity of the hydraulic machine **1522**, without causing the sealing washer **1504k** to lift off its sealing position. Therefore, flow from the extension volume **1508** to the compression volume **1507**, through flow passage **1504e**, would be effectively blocked.

[00148] It should be noted, in some embodiments, PBOV **1576** and/or PBOV **1577** may be configured to be insensitive to rapid pressure increases so long as the differential pressure between the compression volume **1507** and extension volume **1508** remains below a preset threshold. For example, the spring constants and initial compression of spring **1571a** and/or spring **1573a**, may be selected such that the sealing washer **1504l** remains in its sealing position, regardless of the rate of pressure rise in the compression volume **1507**, as long as the net longitudinal hydraulic force on the sealing washer **1504l** is less than the mechanical force applied on that washer by the combination of the springs **1571a** and/or **1573a**.

[00149] Fig. 20 and Fig. 21 illustrate the first face **1504c** and second face **1504d**, respectively, of piston **1504** of the embodiment shown in Fig. 18. These two faces are annular, planar, radially extending surfaces with openings for inlet and outlet ports. Some embodiments may include surfaces that are not planar but that include protrusions and recesses as the disclosure is not so limited. The piston **1504** of Figs. 5 and 6 includes a central opening **1590** that sealably receives extension stud **1504b** and spacer bushing **1578.**

[00150] In the embodiment shown in Fig. 20, piston face **1504c** includes three outlet ports **1504f** and three inlet ports **1504i**. When sealing washer **1504k** is in its seated position, the outlet ports **1504f** are sealed or effectively sealed so that fluid flow through the piston from the extension volume **1508** to the compression volume **1507** is blocked or effectively blocked. However, when the sealing washer **1504k** is seated against the piston face **1504c**, inlet ports **1504i** are not blocked because these openings include recessed pockets **1504m**. Fluid in the compression volume may enter the inlet ports **1504i** by means of recessed pockets **1504m** even when the sealing washer **1504k** is in its sealing position. Therefore, in some embodiments, sealing washer **1504k** may be used to seal one or more outlet ports **1504j**, while allowing flow from the extension volume **1508** to enter one or more inlet ports **1504i** through recessed pockets **1504m**.

[00151] Similarly, in the embodiment shown in Fig. 21, piston face **1504d** includes three outlet ports **1504j** and three inlet ports **1504g**. When sealing washer **1504l** is in its seated position, the outlet ports **1504j** are sealed or effectively sealed so that fluid flow through the piston form the compression volume **1507** to the extension volume **1508** is blocked or effectively blocked. However, when the sealing washer **1504l** is seated against the piston face **1504d**, inlet ports **1504g** are not blocked because these openings include recessed pockets **1504n**. Fluid in the compression volume **1507** may enter the inlet ports **1504j** by means of recessed pockets **1504n** even when the sealing washer **1504l** is in its sealing position. Therefore, in some embodiments, sealing washer **1504l** may be used to seal one or more outlet ports **1504j**, while allowing flow from the compression volume **1507** to enter one or more inlet ports **1504g** through recessed pockets **1504n**.

[00152] Fig. 22 illustrates a sectioned, exploded, perspective view of the dual PBOV system of Fig. 18. Sealing elements **1579a**, **1579b**, **1579c,** and **1579d**, that may be, for example, o-rings are shown. Also shown are flow resistance blocks **1576b** and **1576c** that may include an orifice or other component that induces a hydraulic pressure drop.

[00153] Fig. 23 illustrates the dual PBOV piston and piston rod combination **1590** where the sealing washer **1504l** (shown in cross hatched section) has lifted from its sealing position against face **1504d** of the piston **1504** by a displacement amount **1591.** As a result, fluid in the compression volume **1507** may flow to the extension volume **1508** along, for example, flow path **1592.**

[00154] Fig. 24 illustrates the dual PBOV piston and piston rod combination **1595** where the sealing washer **1504l** is in its sealing position against face **1504d** of the piston **1504**. Proximate floating perch **1573** (shown in cross hatched section) has been displaced by displacement amount δ **1596,** as a result of a pressure differential between the pressure in manifold **1575** and the pressure in the extension volume. In some embodiments, the displacement of the floating perch **1573** relative to the piston rod **1504a** may compress floating spring **1573a** by an equal amount. This compression of floating spring **1573a** may increase the mechanical force on the sealing washer **1504l** by an amount equal to the spring constant of floating spring **1573a** multiplied by the displacement δ.

[00155] Fig. 1 shows piston **4** that includes two flow control devices **4c** and **4b**. One or both of these flow control devices may be bidirectional flow control devices. Fig. 18 shows piston **1504** with dual unidirectional flow control devices, PBOVs **1576** and **1577**. PBOV **1576** is a unidirectional poppet valve that allows fluid flow only from the compression volume **1507** to the extension volume **1508**. PBOV **1577** is a unidirectional poppet valve that allows fluid flow only from the extension volume **1508** to the compression volume **1507**. One or both of these poppet valve PBOVs may be replaced by another appropriate type of flow control device and the disclosure is not limited in this respect.

[00156] Fig. 25 illustrates yet another embodiment of an active suspension actuator piston assembly **1600** with piston **1602**. In this embodiment, compression shim stack **1604**

and the extension shim stack **1606** effectively perform the function of PBOVs **1576** and **1577**, respectively.

[00157] Piston **1602** separates the compression volume from the extension volume of the actuator. In the embodiment in Fig. 25, shim stack **1604**, **in** its closed position, is configured and preloaded to prevent or effectively prevent fluid flow, through one or more flow paths, such as **1607,** from the compression volume to the extension volume. Similarly, in its closed position, shim stack **1606**, is configured and preloaded to prevent or effectively prevent fluid flow, through one or more flow paths such as **1608,** from the extension volume to the compression volume.

[00158] When these shim stacks are exposed to a hydraulic pressure differential above a certain threshold value, they may deflect sufficiently to at least partially unblock a flow path through the piston. The deflection may open a flow path by an amount that is a function of differential pressure and the spring constant of the shim stack. Fig. 26 shows a planar cross-section of the embodiment of Fig. 25. Piston **1602**, shim stack **1604**, and shim stack **1606** are operatively and securely sandwiched between radially extending shoulder **1601a** of piston **1601** and nut **1610** secured to the distal end of the piston rod.

[00159] The compression and extension shim stacks illustrated in Fig. 26 are in series with sealed hydraulic pressure chambers **1612** and **1614**, respectively. In some embodiments, these pressure chambers may be made up of two larger shims bonded to each other on the outer edge to form a flexible expansion chamber. When the pressure differential across the damper piston is increased, flow channel **1616** or **1618** may be used to convey hydraulic fluid from the high pressure side to increase the pressure in the flexible expansion chamber **1612** or **1614**, respectively. Fig. 27 illustrates a planar view of the embodiment of Fig. 25 where the pressure chamber **1612** has expanded to provide additional preload force on the shim stack **1604**. In some embodiments, the effective area of the flexible expansion chamber may be higher than the exposed pressure area under the shim stack, in which case the entire shim stack may simply deflect against the sealing face. This deflection may keep the shim stack seated, thus preventing the hydraulic fluid from flowing across the piston when exposed to higher pressure differentials. This design prevents hydraulic flow across the piston for slow changes in delta pressure. For fast changes in delta pressure, the shims deflect to open the flow path across the piston at a lower differential pressure across the piston. This behavior can be achieved by restricting the flow path connecting the compression or extension volumes to the pressure chamber of the associated shim stack. The combination of the volumetric compliance of each expansion chamber and the flow impedance of the associated flow channel (**1616** or **1618**), effectively acts as a low-pass hydraulic filter that limits the rate of increase in the pressure in the flexible expansion chamber. If the pressure change is faster than the ability for flow through the flow channel to fill the expansion chamber (i.e. increase the pressure), then the preload of the shim stack may remain unchanged or effectively unchanged and the shim stack may deflect away from the sealing face and allow flow across the main piston. As shown in Figs. 25-27, the restricted flow channels may include a small slot in the outer diameter of the piston rod. Alternatively or additionally, these flow channels may also include a notch or hole in the shims that creates a restricted flow path to the expansion chamber.

[00160] In some damper embodiments, a bleed hole may be located in, for example, the main piston which may act as a tuning element. As used herein, the term "tuning element" refers to a component with one or more parameters which may be altered, during calibration and/or operation, to modify the performance of a system that the tuning element is a part of. In some embodiments, if a bleed hole is large it may allow effective regulation of the pressure across the piston at low speeds or rates of pressure increase but may also lead to excessive pumping loss and inefficiency when, for example, the system is trying to build up pressure with a pump. To mitigate this tradeoff, the inventors have recognized that a significant bleed hole may be used at low delta pressures if the bleed element can be adjusted or fully closed as the delta pressure is increased, e.g. by operating the system pump. In some embodiments, this may be achieved by shutting off the bleed hole as the balancing piston attached to the expansion chamber moves to generate preload on the sealing plate. Inventors have further recognized that the displacement of movable internal components of flow control devices may be used to adjust and/or close off bleed holes in those flow control devices during certain operating conditions.

[00161] Fig. 28 illustrates a dual poppet valve PBOV piston assembly **1800** similar to the embodiment illustrated in Fig. 18. Piston assembly **1800** includes actuator piston **1802,** piston rod **1801**, compression PBOV **1804**, and extension PBOV **1806**. Fig. 29 illustrates a planar cross-section of the piston assembly **1800**. Fig. 29 also shows bleed hole **1902** that fluidly connects the compression volume**1904** to the extension volume **1903** via, for example, in-piston flow channel **1905**. Fig. 30 illustrates the embodiment shown in Fig. 29 where the spring perch **1906** is a displaced to position where it blocks bleed hole **1902.** In this embodiment the displacement of the spring perch **1906** may be used to dynamically tune the certain operating parameters of the PBOV and the active suspension actuator system by leaving the bleed passageway unobstructed, or partially or fully blocking the bleed hole depending on the operating condition.

[00162] Fig. 31 illustrates active suspension actuator system **2500** with actuator **2501** that includes pressure tube **2502** and second tube **2510** at least partially surrounding pressure tube **2502**. The pressure tube **2502** slidably receives piston **2504** that divides an internal volume in the pressure tube into a compression volume **2507** and an extension volume **2508**. The actuator system **2500** includes a hydraulic machine **2509** that may operate as a hydraulic pump and/or a hydraulic motor. The hydraulic machine **2509** may be operatively coupled to an electric machine (not shown) which may drive the hydraulic machine **2509** and/or be driven by the hydraulic machine **2509**.

[00163] In some embodiments, the hydraulic machine **2509** may include a first port **2509a** and a second port **2509b**. The longitudinal axis of the second tube **2510** and the pressure tube **2502** are parallel to or effectively parallel to the Z axis **2511**. The pressure tube **2502** and the second tube **2510** form an intervening volume **2513**, which may be annular in shape in whole or in part. The intervening volume **2513** may include a first conduit **2513a** that may be in fluid communication with the extension volume **2508** and a second flow conduit **2513b** that may be in fluid communication with the compression volume **2507**. The first and second flow conduits **2513a**, **2513b** of intervening volume **2513** may be fluidly separated by a barrier **2513c**. In the embodiment of Fig. 2, the first port **2509a** is in fluid communication with the extension volume **2508** via flow conduits **2514a** and **2513a**, while the second port **2509b** may be fluidly connected to accumulator **2515** via flow conduit **2514b**. Accumulator **2515** is fluidly connected to the compression volume **2507** via flow conduits **2514c** and **2513**.

[00164] The accumulator **2515** includes two volumes **2515a** and **2515b** where the volume **2515a** may be filled with hydraulic fluid and volume **2515b** may be filled with a compressible medium such as, for example, a gas such as air, nitrogen or argon. The material in volume **2515a** may be separated from the material in volume **2515b** by a piston **2515c** shown in Fig. 31 or a diaphragm or bladder (not shown).

[00165] The accumulator **2515** in Fig. 31 is a flow-through accumulator, but the disclosure is not limited to such an accumulator as any appropriate accumulator such as a branched or inline accumulator may be used instead of or in addition to the flow through accumulator **2515**. The flow between the accumulator **2515** and the compression volume **2507** may be controlled by a flow control device. The flow control device may be, for example, a base valve **2516** that is incorporated into the compression end of the actuator **2501** shown in Fig. 31. The base valve **2516** may also be incorporated into the extension end of the actuator in which case it would need to accommodate the piston rod **2004a**. The inventors have recognized that in some embodiments, the base valve **2516** may be configured to operate as a passive digressive valve in compression and/or extension.

[00166] Alternatively or additionally, as indicated in Fig. 31, and illustrated in Fig. 32, in some embodiments, active suspension actuator system **2560** may include an accumulator **2515** and/or a flow control device **2516** which may be operatively interposed between the hydraulic machine **2509** and the extension volume **2508**. Fig. 32 shows the flow control device **2561** located outside the second tube **2510**, however, it may be incorporated internally and the disclosure is not limited in this respect. In the case of an internal flow control device at the extension end of the actuator, the flow control device may include an opening configured to receive piston rod **2004a**.

[00167] Fig. 33 illustrates exemplary performance curves **2550** for a bi-directional digressive base valve. The pressure drop across the valve (ΔP) may increase linearly or effectively linearly as a function of flow rate Q in the low flow range (i.e. ranges A_E and A_C). It is noted that in some embodiments, pressure drop across the valve may also increase in a quadratic or other relationship if, for example, a simple orifice hole is used in the base valve to generate damping. In some embodiments this pressure drop may be due to flow through a leakage path through the valve.

[00168] In some embodiments, once the pressure drop reaches a level indicated by **2553a** during extension or **2553b** during compression, shim stacks or other pressure relief valves incorporated in the base valve may open to allow additional flow to pass through the base valve. In some embodiments, shim stacks may be configured to have a cracking pressure in the range of, for example, 10 psi to 300 psi. In some embodiments, the cracking pressure may be in the range between 5 psi and 500 psi. Cracking pressure in other ranges may also be used and the disclosure is not so limited.

[00169] For flow rates in ranges B_E and B_C, the damping may increase linearly or effectively linearly as a function of flow rate Q (curve **2551a** and **2551b**) or be constant or effectively constant as a function of flow rate Q (curve **2552a** and **2552b**). In some embodiments, the shim stacks may be configured to maintain a constant or effectively constant pressure drop over the ranges B_E and B_C. In some embodiments, ranges B_E and B_C represent flow rates from 0.25 to 5 GPM, or 0.1 to 4 GPM. In some embodiments for flow rates in the range C_E and/or C_C and above, the base valve may behave as an orifice restriction where the damping rate (or damping) may continue to increase.

[00170] Figs. 34-39 illustrate a structure of an exemplary bi-directional digressive (i.e., digressive during both the compression and extension flows) base valve. Fig. 34 illustrates an isometric drawing of a base valve body **2650**. This valve operates by controlling flow in both compression and extension by using two sets of shim stacks. These shim stacks are not shown in Fig. 34.

[00171] In the embodiment shown in Fig. 34, intake ports **2601**, are angularly distributed about a longitudinal axis of the base valve body **2600**. Each intake port leads into one of a first set of internal flow channels in the base valve body **2600** that convey the pressure in the compression volume to the compression shim stack (not shown). Outlet ports **2602** are connected to flow channels that convey the pressure in the extension volume (e.g. volume **2508** shown in Fig. 31) to an extension shim stack (not shown). Port **2603** connects to one internal flow channel in the base valve body **2600** that is configured to allow flow to bypasse both sets of shim stacks. Annular sealing surface **2604** is the surface that the sealing shim or shims of the extension shim stack seal against. All of the outlet ports **2602** are connected by annular cavity **2605**.

[00172] Fig. 35 shows an isometric cross-section of the base valve assembly **2650** that includes base valve body **2600** of Fig. 34, extension shim stack **2651**, and compression shim stack **2652**. The compression shim stack **2652** may open during a compression stroke of the actuator piston (e.g., piston **2504** shown in Fig. 31) while the extension shim stack **2651** may open when the piston moves in an extension stroke.

[00173] Bolt **2653** and nut **2654** may be used to secure the two shim stacks **2651** and **2652** to the base valve body **2600**. The large flange at the top of nut **2654** is used as a travel limiter to prevent excessive deflection of the shims under high hydraulic flow rates. A washer may be used under the head of bolt **2653**. The extension shim stack **2651** includes a sealing shim **2651a** which seats against a portion of sealing surface **2604** while compression shim stack **2652** includes a compression sealing shim **2652a** which seats against annular sealing surface **2655**. Inlet ports **2656**, which are angularly distributed about the longitudinal axis of the base valve body **2600**, convey the pressure in conduit **2513** via the internal extension passageways to the annular cavity **2605**. The pressure in the annular cavity **2605** acts on the compression sealing shim **2652a**.

[00174] Fig. 36 illustrates a front view of a cross-section of the base valve assembly **2650** located at the compression end of actuator **2501** shown in Fig. 31. Fig. 36 shows how the compression volume **2507** is fluidly connected to the annular cavity **2656.** The pressure in the annular cavity **2656** acts on the sealing compression shim **2652a** and under certain conditions causes the shim **2652a** to deflect and open.

[00175] Fig. 38 illustrates that once the compression sealing shim **2652a** deflects, fluid may flow from the compression volume **2507**, through the flow channels **2657**, which, in the embodiment of Fig. 38, are fluidly connected to conduit **2513**.

[00176] Fig. 38 illustrates another front view cross-section of the base valve **2650** where the pressure in conduit **2513** is conveyed to the extension sealing shim **2651a.** Under certain conditions, the sealing shim **2651a** may deflect and allow fluid from the conduit **2513** to flow into the compression volume **2507**.

[00177] The inventors have recognized that in an active suspension actuator operating at low piston velocities, significant damping may be preferred as discussed above with regard

to Fig. 33. However, the inventors have also recognized that when operating at higher flow rates, such as in flow ranges B_E (extension) and B_C (compression) in Fig. 33, much higher flow rates may need to be accommodated through the base valve than, for example, in the case of passive dampers. To allow such flow rates, the diameters of the compression and/or extension sealing shims **2651a**, **2652a** may need to be maximized.

[00178] For example, during rapid compression in passive dampers, fluid may flow, for example, to the extension volume through valves in the piston as well as the base valve. Under such circumstances, only the volume displaced by the intrusion of the piston rod into the extension volume may need to flow through the base valve.

[00179] In an active system, the inventors have recognized that the valves in the piston may, for example, need to be set at a high enough cracking pressure to allow the production of sufficient active force with the pump.

[00180] On the other hand, in some embodiments of active suspension systems, all or effectively all of the volume displaced by the piston may flow through the base valve. The inventors have recognized that the desired capacity of the base valve in either compression and extension and the pressure drop vs. flow relationship may be achieved by maximizing the diameter of the compression sealing shim **2652a** and/or the extension sealing shim **2651a**.

[00181] Fig. 39 illustrates a base valve assembly **2700** engaging the compression end of a twin tube damper/actuator. The diameter of the compression sealing shim and extension sealing shim are larger than the embodiment illustrated in Figs. 34-38. The embodiment shown in Fig. 39 includes an extension shim stack **2701** and a compression shim stack **2702** and base valve body **2703**. The shim stacks are secured to the valve body by bolt **2704** and nut **2705**. Shim stack **2701** includes sealing shim **2701a** that seals against annular surface **2706**. Shim stack **2702** includes sealing shim **2702a** that seals against annular surface **2708**.

[00182] The diameter of the sealing shim **2701a** may be increased relative to the embodiment shown in Figs. 34-38 because inlet ports **2710** of compression flow conduits **2715** are recessed into the base valve body **2703** and are not located in the same radial plane as the annular sealing surface **2706**. Before reaching the inlet ports **2710**, the fluid from the compression volume flows through an annular or partially annular volume **2720** between the valve body and the pressure tube **2502**.

[00183] The inventors have recognized that by using such a configuration, the diameter of the extension sealing shim **2701a**, may be greater than 70%, but less than 100% of the inner diameter of the pressure tube **2502**. Alternatively, the diameter of the extension sealing shim **2701a** may be greater than 80% but less than 100% of the inner diameter of the pressure tube **2502**. In some embodiments, the diameter of the extension sealing shim **2701a** may be greater than 90%, but less than 100%, of the inner diameter of the pressure tube **2502** or any other appropriate percentage of the inner diameter of the pressure tube **2502**.

[00184] Fig. 40 illustrates how the pressure in the compression volume **2722** may be conveyed to the compression sealing shim **2702a** through annular or partially annular region **2720**, compression flow channels **2715** in the base valve body **2703**, and the annular cavity **2721** to act on the compression sealing shim **2702a**. Under certain conditions, the pressure may cause the sealing shim **2702a** to deflect and create a flow path from the compression volume **2722** to the flow conduit **2513**.

[00185] Fig. 41 shows another cross-section of base valve assembly **2700**. Alternatively or additionally, as illustrated in Fig. 41, the inlet ports **2730** to the extension conduits **2731** may be recessed axially into the body **2703** of the base valve relative to the sealing surface. By using this configuration, the diameter of the compression sealing shim **2702a**, may be greater than 70%, but less than 100% of the inner diameter of the pressure tube **2502**. Alternatively, the diameter of the compression sealing shim **2702a** may be greater than 80% but less than 100% of the inner diameter of the pressure tube **2502**. In some embodiments, the diameter of the compression sealing shim **2702a** may be greater than 90%, but less than 100%, of the inner diameter of the pressure tube **2502**or any other appropriate percentage of the inner diameter of the pressure tube .

[00186] By using such a configuration, the diameter of the compression sealing shim **2702a** may be greater than the inner diameter of the pressure tube **2502**. In some embodiments, the diameter of the compression sealing shim **2702a** (not shown) may be greater than 70%, but less than 100% of the inner diameter of the second tube **2510**. Alternatively, the diameter of the compression sealing shim **2702a** may be greater than 80%

but less than 100% of the diameter of the second tube **2510**. In some embodiments, the diameter of the compression sealing shim **2702a**may be greater than 90%, but less than 100% of the diameter of the second tube **2510**or any other appropriate percentage of the diameter of the second tube **2510**.

[00187] Fig. 42 illustrates how the pressure in the flow conduit **2513** may be conveyed through inlet ports **2730**, extension flow channels **2731** in the body **2703** of the base valves, and annular cavity **2733**. Pressure in the flow conduit **2513** causes the sealing shim **2701a** to deflect and open a flow path between the flow conduit **2513** and the compression volume **2722** in the direction shown in Fig. 42.

[00188] The isometric cross-section shown in Fig. 43 illustrates another embodiment of a high capacity base valve assembly **2800** that includes a multipiece valve body **2801**, extension shim stack **2802**, and compression shim stack **2803** that are securely attached together by bolt **2804** and nut **2805**. Sealing extension shim **2802a** seals against sealing surface **2802b**, while sealing compression shim **2803a** seals against a portion of sealing surface **2803b**. Inlet port **2806** of compression flow channel **2807** is recessed axially further into the valve body **2801** relative to the sealing surface. Fig. 44 illustrates a front view cross-section view of the base valve assembly of Fig. 43 showing the compression flow channels **2807** with inlet ports **2806** that are axially recessed from the sealing surface **2802b** of the extension sealing shim **2802a**. In some embodiments, a bleed passage **2810** may allow for a flow across the valve without passing through either shim stack. In some embodiments this bleed hole may be in the rage between 0.1 and 3 mm, or 0.05 – 5 mm hydraulic diameter. The size, length, and number of bleed holes may be used to determine or tune the flow pressure drop relationship in the ranges A_E and A_C illustrated in Fig. 33.

[00189] Fig. 45 illustrates another front view cross-section of the base valve **2800** showing the extension inlet ports **2820** of extension flow conduits **2830** in the body of valve assembly **2800**. It should be noted that the inlet ports **2820** are recessed relative to the sealing surface **2803b** of the compression sealing shim **2803a**.

[00190] While the present teachings have been described in conjunction with various embodiments and examples, it is not intended that the present teachings be limited to such embodiments or examples. On the contrary, the present teachings encompass various alternatives, modifications, and equivalents, as will be appreciated by those of skill in the art. Accordingly, the foregoing description and drawings are by way of example only.

CLAIMS

1. An active suspension actuator system, comprising:

a pressure tube with an internal volume;

a piston, slidably received in the internal volume, that divides the internal volume into a compression volume and an extension volume, wherein in at least one mode of operation the pressure in the compression volume oscillates at a first frequency;

a hydraulic machine with a first port and a second port, wherein the first port is fluidly connected to the compression volume by a first flow path that does not include the second port and wherein the second port is fluidly connected to the extension volume by a second flow path that does not include the first port;

a controller configured to operate the hydraulic machine as a hydraulic pump in at least a first operating mode and as a hydraulic motor in a second operating mode;

a damping flow control device, with a total impedance, configured to damp flow oscillations in the flow in at least a portion of the first flow path, wherein the impedance of the flow control device decreases when the pressure drop across the flow control device exceeds a first preset threshold pressure drop;

a first bypass flow control device fluidly interposed between the compression volume and extension volume, wherein the first bypass flow control device includes a first pressure balanced blow-off valve with a pressure offset , wherein the pressure balanced blow-off valve allows fluid to flow from the compression volume to the extension volume when: the pressure differential across the first pressure balanced blow-off valve exceeds a first threshold pressure differential and a first frequency exceeds a first preset frequency threshold.

2. The active suspension actuator system of claim 1, wherein the pressure offset is one of an absolute offset and a relative offset.

3. The active suspension actuator system of claim 1, wherein the damping flow control device includes at least one damping element.

4. The active suspension actuator system of claim 3, further comprising an accumulator fluidly connected to the compression volume by means of at least a portion of the first flow passage that includes the damping flow control device.

5. The active suspension actuator system of claim 4, wherein the damping coefficient of the first flow path is in the range of 10 to 400 newton-meters per second.

6. The active suspension actuator system of claim 4, wherein the damping coefficient of the first flow path for fluid flow towards the compression volume is different from the damping coefficient of the first flow path for fluid flow away from the compression volume.

7. The active suspension actuator system of claim 1, wherein the damping flow control device includes a first flow path with a damping element and at least a second flow path with a biased check valve, wherein the first and second flow paths are fluidly parallel to each other.

8. The active suspension actuator system of claim 7, wherein in at least one operating condition the check valve in the second flow path opens when a flow through the first flow path induces a pressure drop that is greater than the preset threshold pressure drop.

9. The active suspension actuator system of claim 8, wherein an overall damping coefficient of the first flow path is lower in one flow direction when the check valve in the second flow path is open relative to when it is closed.

10. The active suspension actuator system of claim 7, wherein the blow off valve of the first bypass flow control device discharges fluid from the compression volume when the pressure in the compression volume is greater than the preset threshold pressure differential and pressure oscillation in the compression volume is in a range of frequencies between 10 Hz and 100 Hz.

11. The active suspension actuator system of claim 10, wherein the controller operates the hydraulic machine to apply an active force on an unsprung mass of the vehicle in a range of

frequencies between 10 Hz and 100 Hz when the pressure differential across the piston is less than preset threshold pressure differential.

12. The active suspension actuator system of claim 10, wherein the first bypass flow control device is located in the piston.

13. The active suspension actuator system of claim 12, further comprising a second bypass flow control device located in the piston that discharges fluid from the extension volume.

14. The active suspension actuator system of claim 10, wherein the first bypass flow control device is external to the pressure tube.

15. An active suspension actuator system of a vehicle, comprising:

a piston that includes a first inlet port and a first outlet port, wherein the first inlet port is fluidly connected to the first outlet port by a first fluid flow passage internal to the piston;

a pressure tube with an internal volume divided into a first chamber and a second chamber by the piston, wherein the piston is slidably received in the internal volume, and wherein the first inlet port is fluidly connected to the first chamber;

a piston rod attached to a sprung mass of the vehicle at a first end and the piston at a second end;

a first flow control device attached to the piston that selectively, fluidly connects the first outlet port to the second chamber based at least in part on a pressure differential between the first and second chambers and a property selected from the group consisting of: a rate of pressure rise in the first chamber, a rate of change of the pressure differential between the first and second chamber, a rate of pressure drop in the second chamber, and a frequency of pressure fluctuation in the first chamber.

16. The piston assembly of claim 15, further comprising a second flow control device attached to the piston, wherein the first face also includes a second outlet port and the second

face also includes a second inlet port, wherein the second inlet port is fluidly connected to the second outlet port by a second fluid flow passage internal to the piston and second inlet port is fluidly connected to the second chamber, and wherein the second flow control device selectively, fluidly connects the second outlet port to the first chamber based at least in part on the pressure differential between the first and second chambers and a property selected from the group consisting of: a rate of pressure drop in the first chamber, the rate of change of the pressure differential between the first and second chamber, a rate of pressure rise in the second chamber, and a frequency of pressure fluctuation in the second chamber.

17. The piston assembly of claim 15, wherein the first flow control device is a pressure balanced blow-off valve.

18. The piston assembly of claim 17, wherein the pressure balanced blow-off valve is a passive valve.

19. The piston assembly of claim 18, wherein the first pressure balanced blow-off valve includes a sealing washer, wherein the sealing washer, when in a sealing position, prevents flow from the first outlet port into the second chamber.

20. The piston assembly of claim 19, further comprising a first spring perch that is fixed relative to the piston rod, a first spring operatively interposed between the first spring perch and the sealing washer, a second spring perch that is configured to float relative to the piston rod, a second spring operatively interposed between the second spring perch and the sealing washer, wherein the first and the second springs combine to apply a net mechanical force on the sealing washer that urge the sealing washer towards a sealing position.

21. The piston assembly of claim 15 wherein the property is the rate of change of the pressure differential between the first and second chamber.

22. The piston assembly of claim 15 wherein the property is the rate of pressure rise in the first chamber.

23. A method of operating an active suspension actuator system of a vehicle; the method comprising;

over a road surface, operating the vehicle with the active suspension actuator operatively interposed between a wheel assembly and a vehicle body, wherein the actuator has a compression volume and an extension volume that are fluidly connected to a first and second port of a hydraulic machine respectively;

striking a discontinuity in the road surface with a wheel associated with the wheel assembly;

increasing a pressure in the compression volume to a first value greater than a pre-selected threshold value at a first rate of increase;

opening a pressure balanced blow-off valve to discharge at least a portion of the increased pressure from the compression volume to the extension volume;

operating the hydraulic machine to increase the pressure in the compression volume to a value greater than the threshold at a second rate lower than the first rate without opening the pressure balanced blow-off valve; and

operating the hydraulic machine to increase the pressure in the compression volume to a value less than the threshold value at a rate greater than the first rate without opening the pressure balanced blow-off valve.

24. The method of claim 23, wherein the pressure balanced blow-off valve is passive.

25. A method of operating an active suspension actuator system of a vehicle; the method comprising;

over a road surface, operating the vehicle with the active suspension actuator operatively interposed between a wheel assembly and a vehicle body, wherein the actuator has an internal cylindrical volume that is separated into a first chamber and a second chamber by a piston that is slidably received in the internal volume, and wherein the first chamber and second chamber are fluidly connected to a first and second port of a hydraulic machine respectively;

striking a discontinuity in the road surface with a wheel associated with the wheel assembly;

increasing a pressure in the first chamber to a first value greater than a pre-selected threshold value at a first rate of increase;

opening a pressure balanced blow-off valve to discharge at least a portion of the increased pressure from the first chamber to the second chamber;

operating the hydraulic machine to increase the pressure in the first chamber to a value greater than the threshold at a second rate lower than the first rate without opening the pressure balanced blow-off valve; and

operating the hydraulic machine to increase the pressure in the first chamber to a value less than the threshold value at a rate greater than the first rate without opening the pressure balanced blow-off valve.

26. The method of claim 25, wherein the pressure balanced blow-off valve is passive.

27. The method of claim 26, wherein the first chamber is a compression volume and the second chamber is an extension volume.

28. The method of claim 26, wherein the first chamber is an extension volume and the second chamber is a compression volume.

29. An active suspension actuator, comprising:

an actuator that includes a pressure tube and a second tube, the second tube at least partially surrounding the pressure tube, wherein the pressure tube includes a compression volume and an extension volume separated by a piston, wherein the pressure tube and the second tube have an extension volume end and a compression volume end, wherein the pressure tube and the extension tube form an intervening volume;

a base valve assembly, located at the compression volume end of the pressure tube and the second tube, that includes:

a base valve body that includes internal compression flow conduits and extension flow conduits that fluidly connect the compression volume and at least a portion of the intervening volume;

an extension shim stack securely attached to the base valve body that includes an extension sealing shim that controls the flow from the intervening volume via the extension flow conduits to the compression volume and prevents flow in the opposite direction in the extension flow conduits; and

a compression shim stack securely attached to the base valve body that includes a compression sealing shim that controls the flow from the compression volume via the compression flow conduits to the intervening volume and prevents flow in the opposite direction in the compression flow conduits.

30. The active suspension actuator of claim 29, wherein the base valve assembly includes a leakage path between the compression volume and the intervening volume.

31. The active suspension actuator of claim 29, wherein pressure drop as a function of flow rate through the base valve during compression is effectively equal to pressure drop as a function of flow rate through the base valve during extension.

32. A bidirectional base valve of an active suspension actuator, comprising:

a base valve body with a first set of internal flow channels and a second set of internal flow channels;

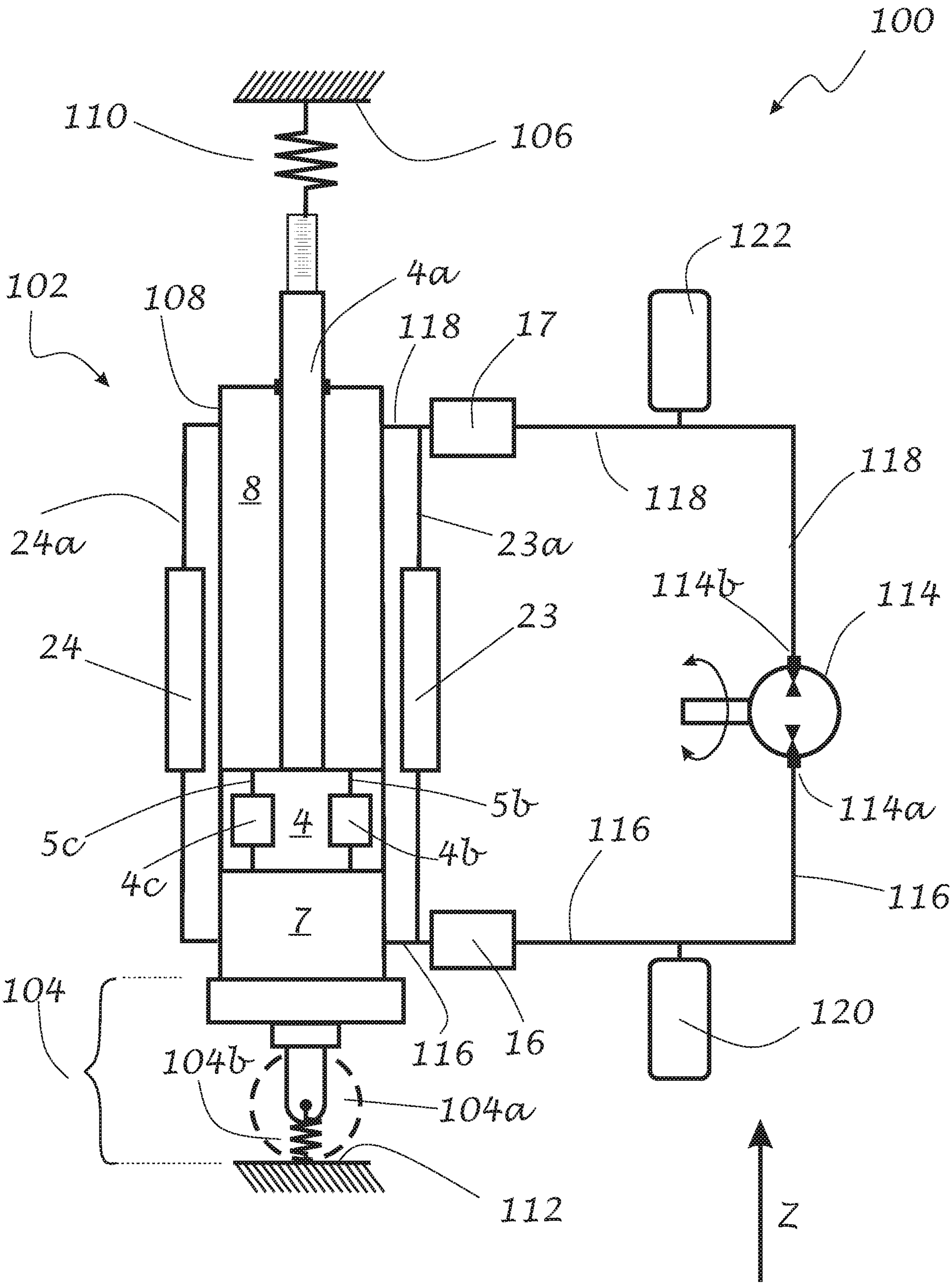
a first shim stack that is configured to regulate flow, through the first set of flow channels, from a compression volume of the actuator to a second volume in the actuator and blocks flow from the second volume from entering the first set of flow channels; and

a second shim stack that is configured to regulate flow, through the second set of flow channels, that flows from the second volume to the compression volume and blocks flow from the compression volume from entering the second set of flow channels.

33. The bidirectional base valve of claim 4, further comprising at least one bleed flow passage that fluidly connects the compression volume and the second volume that bypasses the first shim stack and the second shim stack.

34. The bidirectional base valve of claim 5, wherein the first shim stack includes at least one sealing shim that seals against a first surface of the base valve body and wherein the inlet ports of the first set internal flow channels is axially recessed from the first sealing surface.

35. The bidirectional base valve of claim 5, wherein the second shim stack includes at least one sealing shim that seals against a second surface of the base valve body and wherein the inlet ports of the second set internal flow channels is axially recessed from the second sealing surface.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17
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118
8
24a
23a
114b
114
24
23
5c
5b
4
4c
4b
114a
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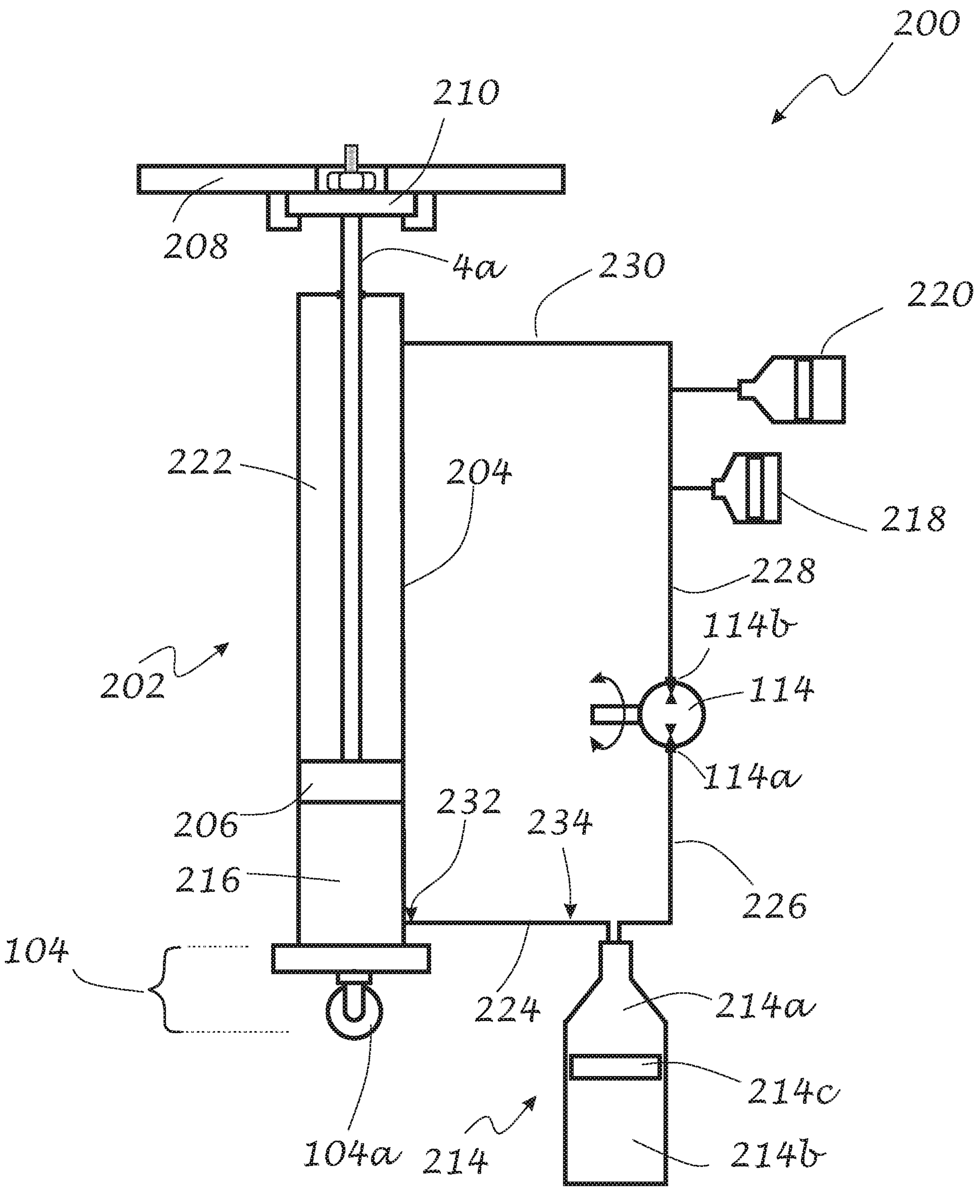
116
7
104
116
16
120
104b
104a
112
Z

Fig. 1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214
214b

Fig. 2

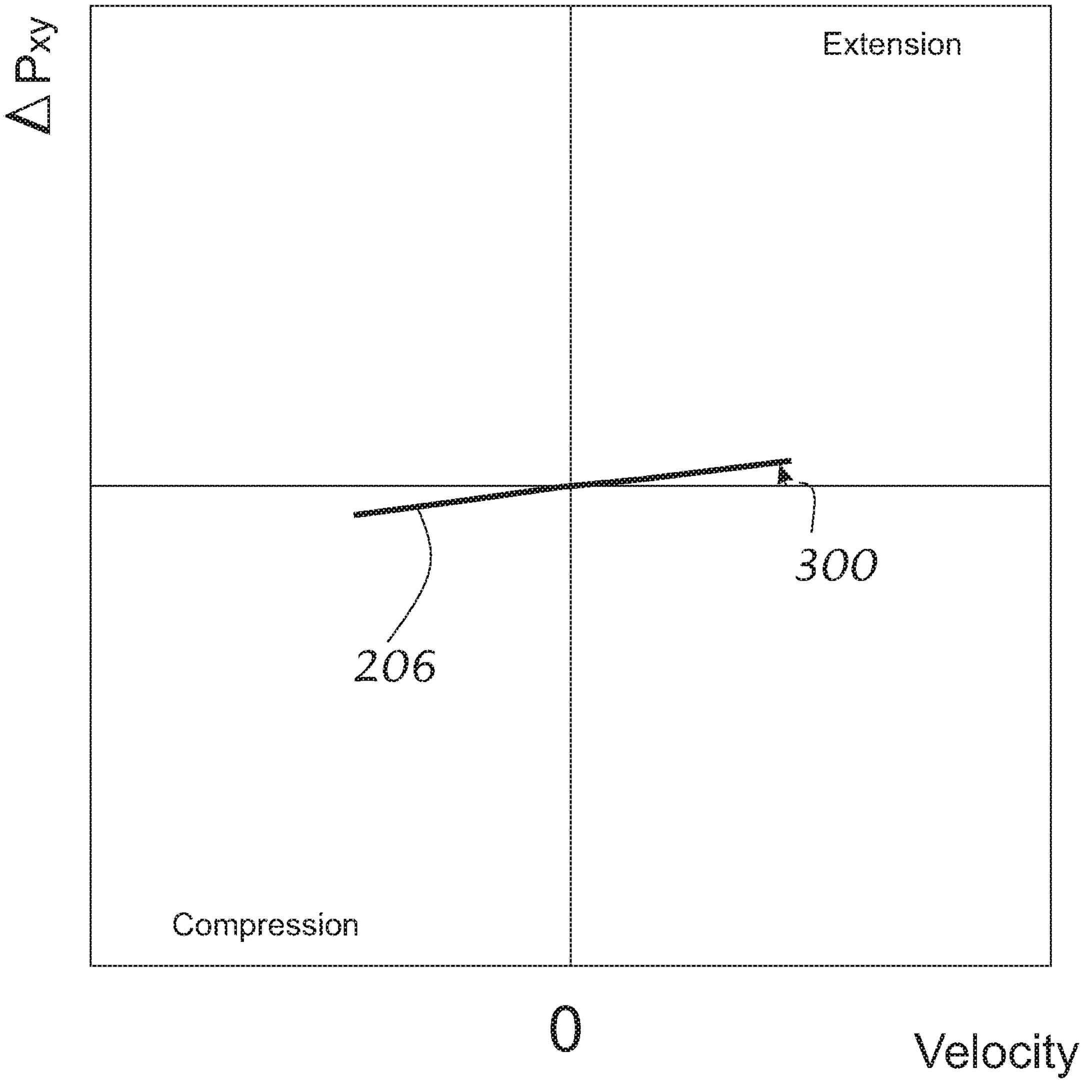


Fig. 3

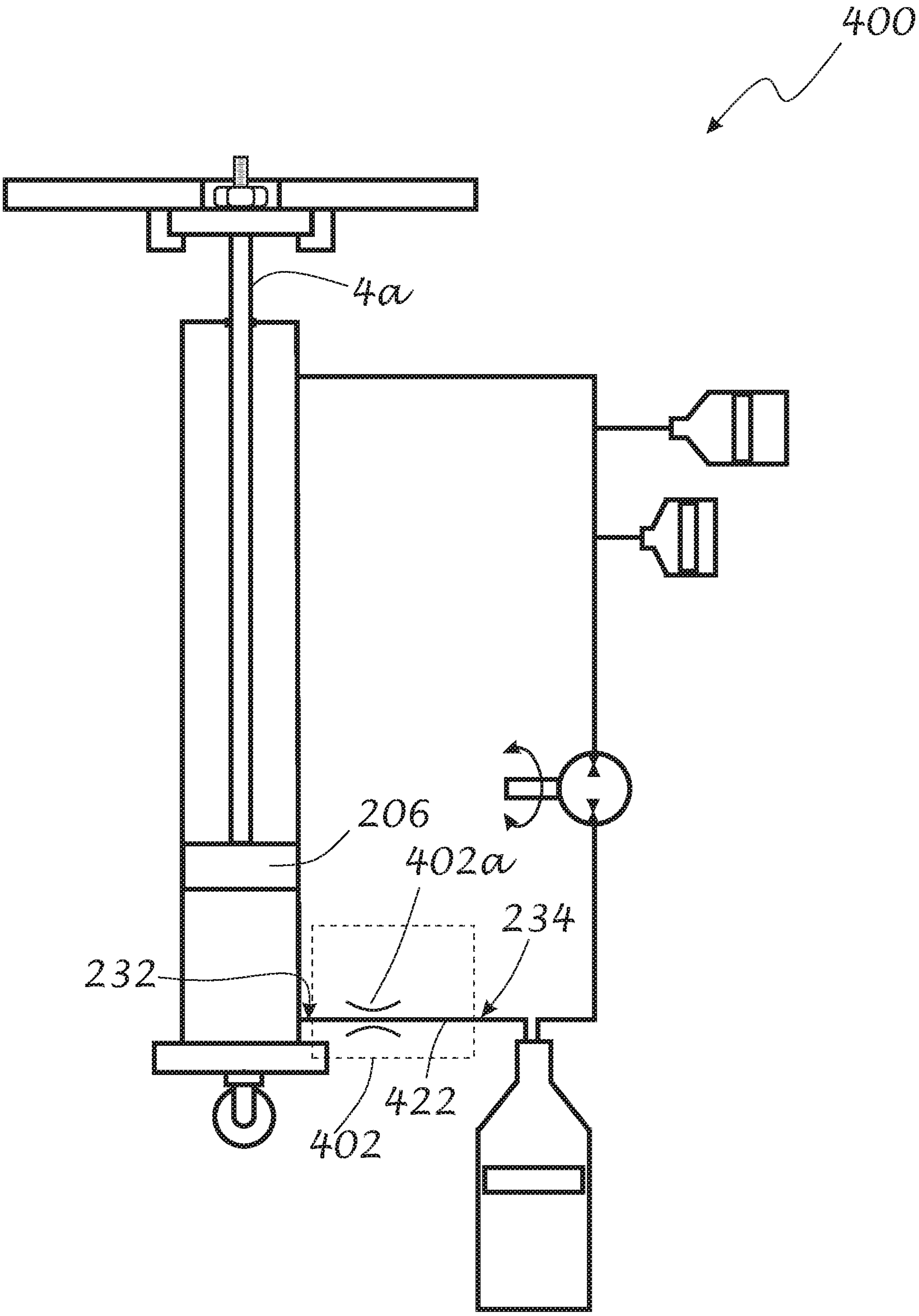


Fig. 4

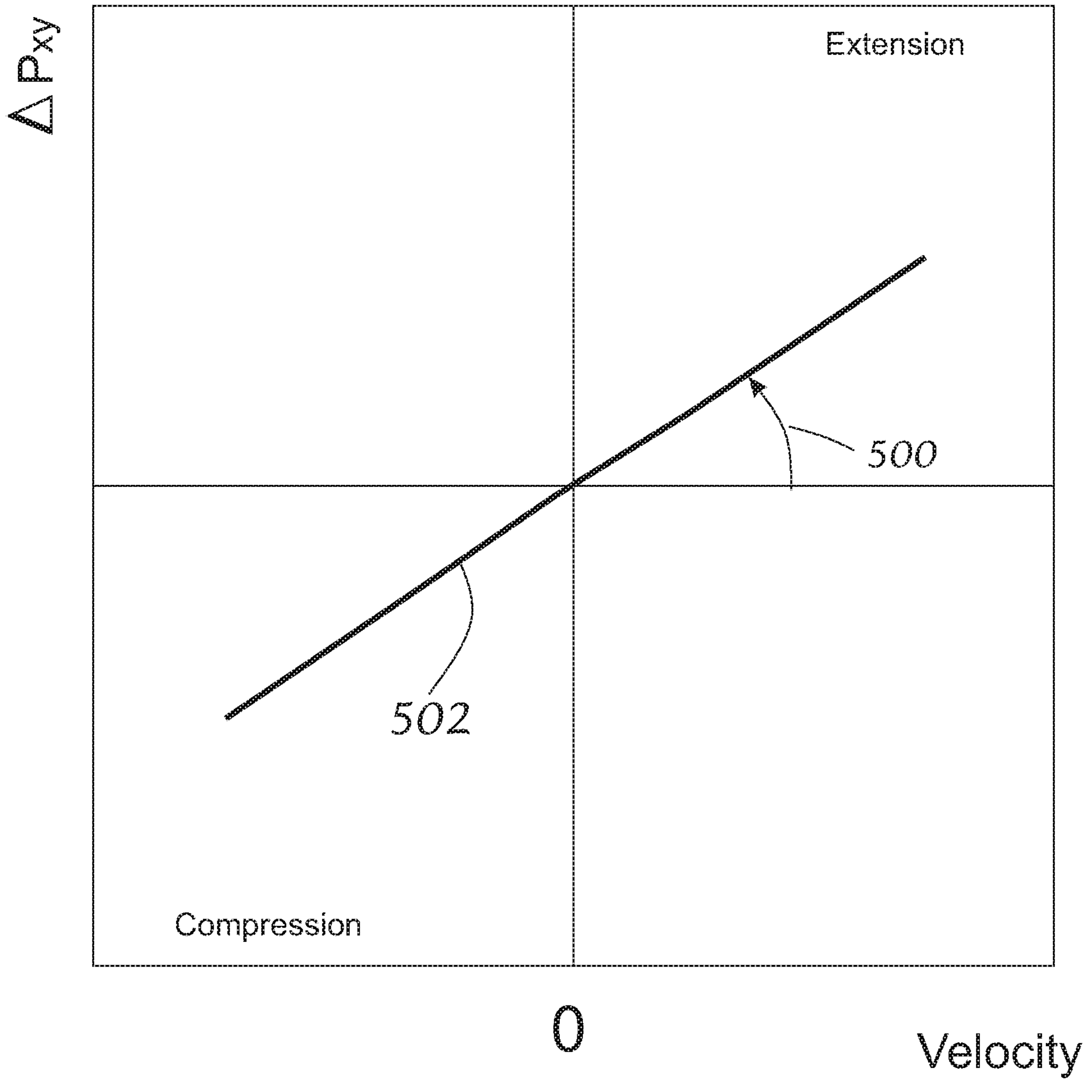


Fig. 5

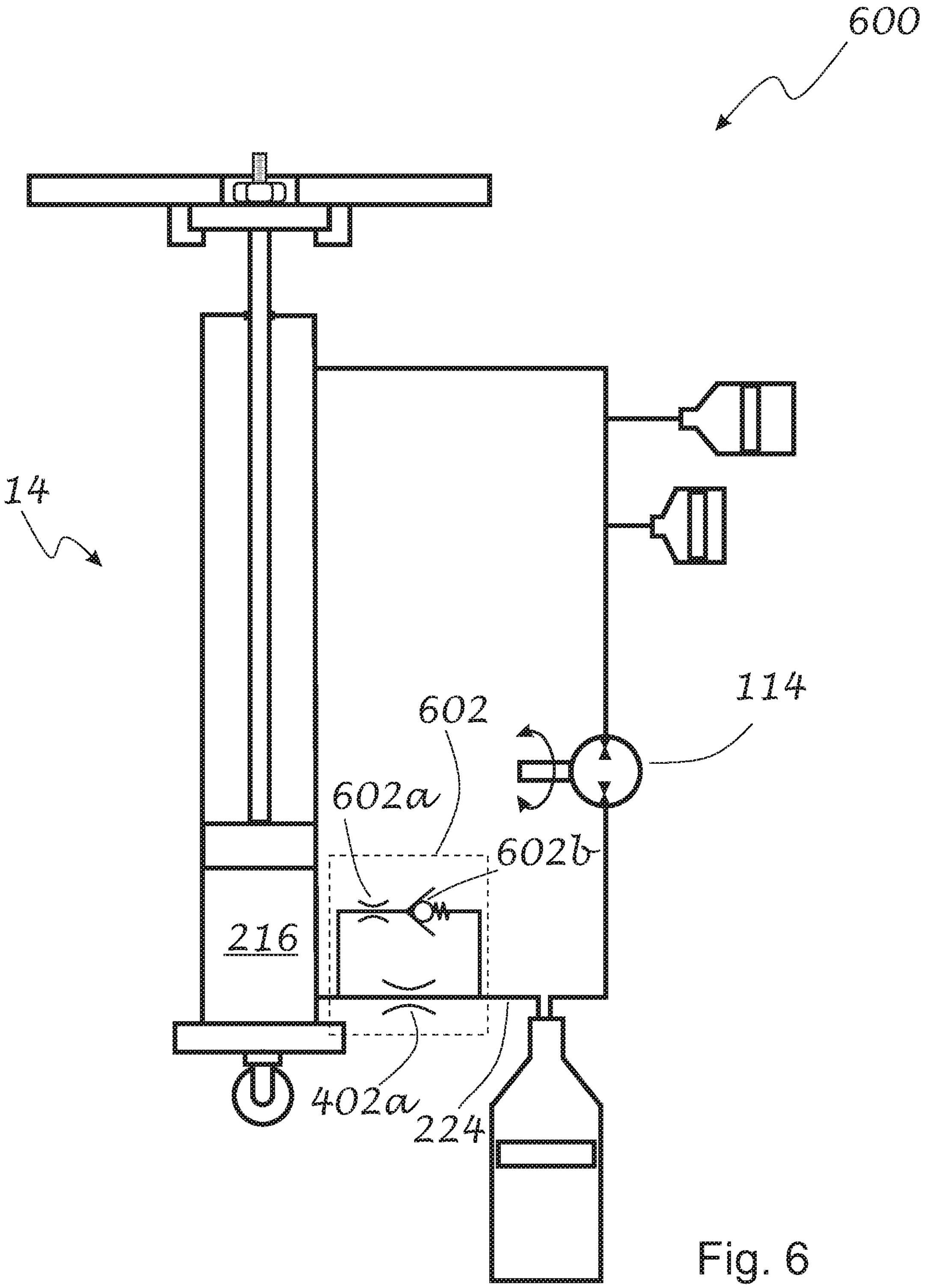


Fig. 6

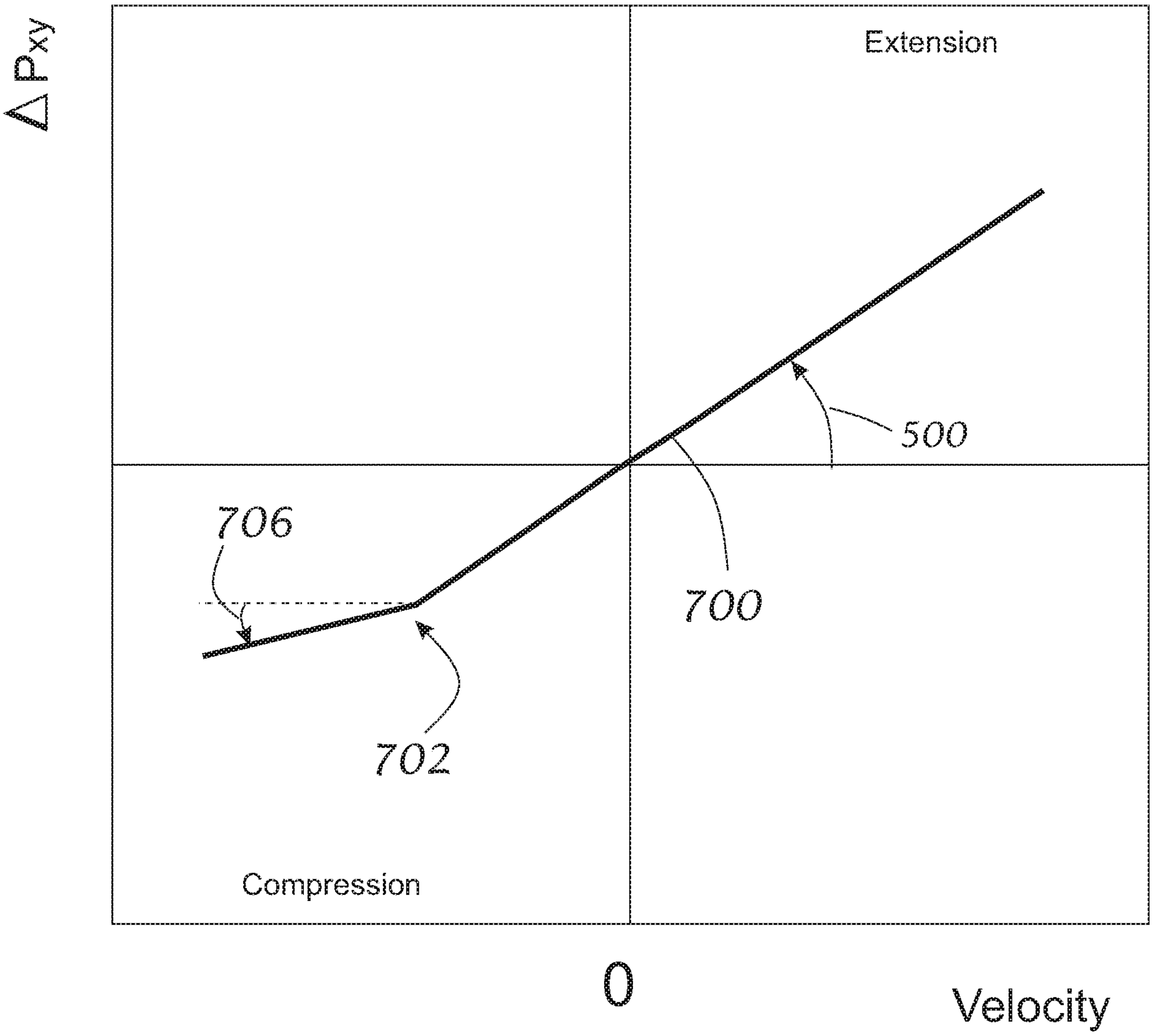


Fig. 7

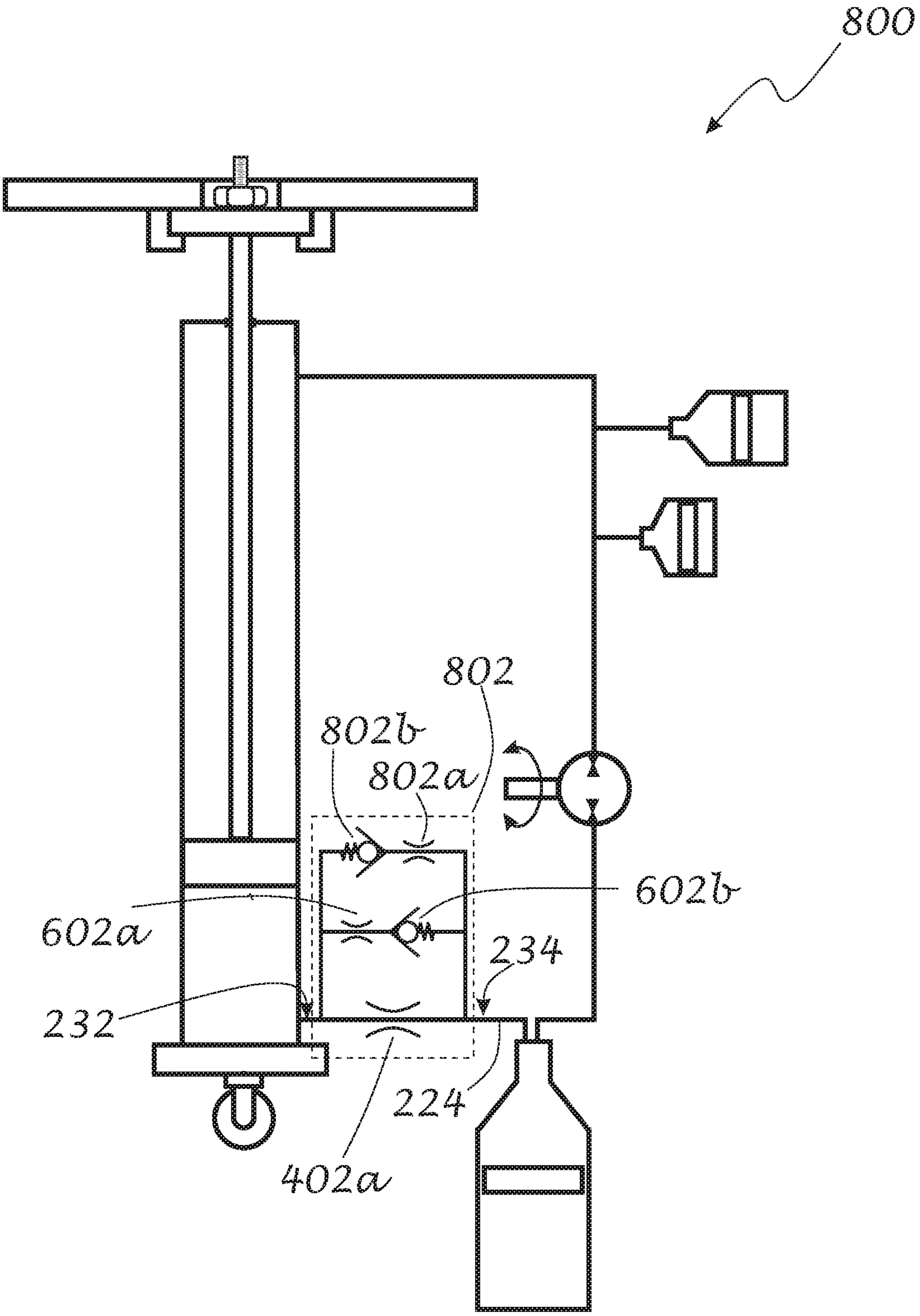
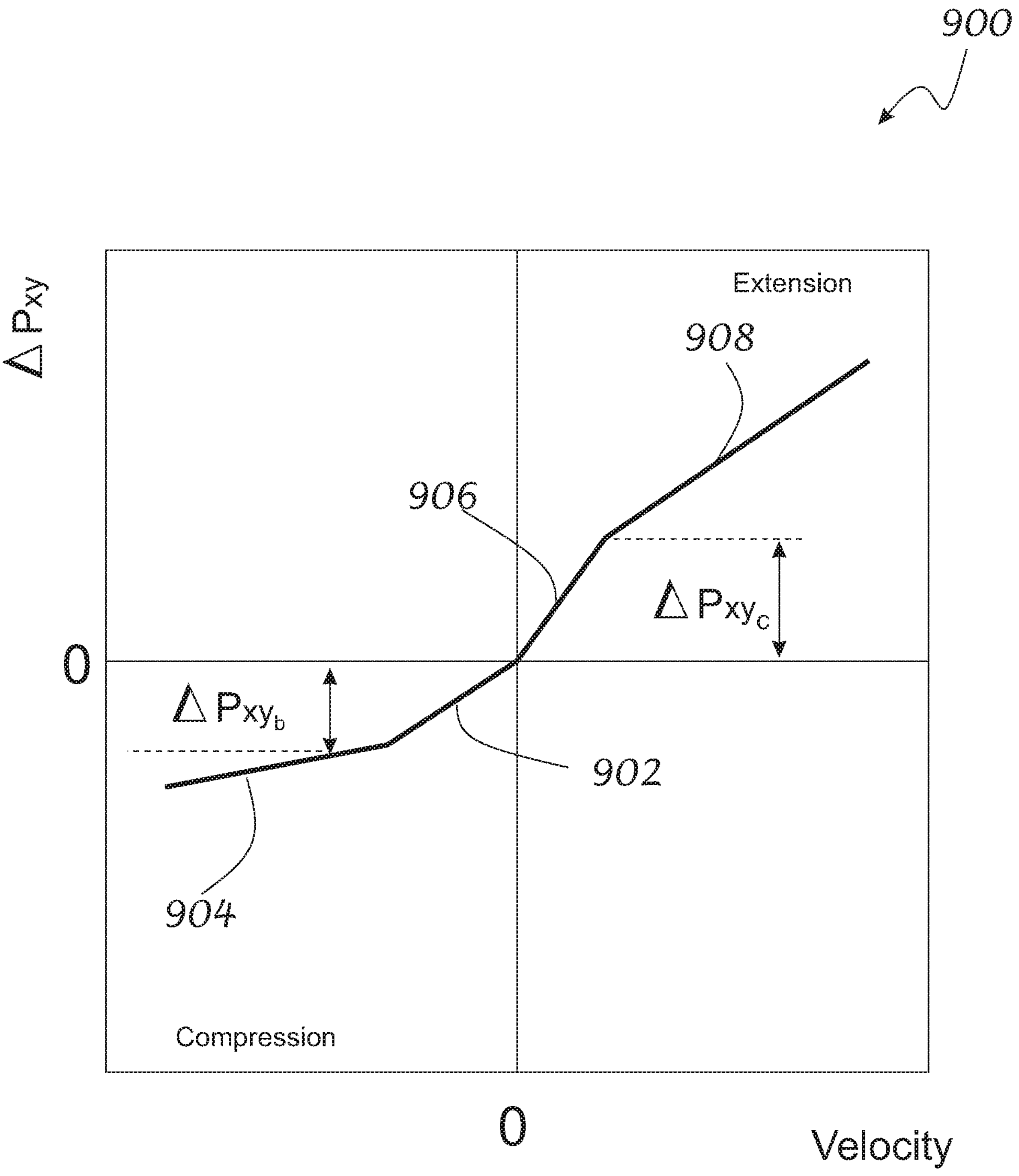


Fig. 8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ΔPxyb
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Compression
0
Velocity

Fig. 9

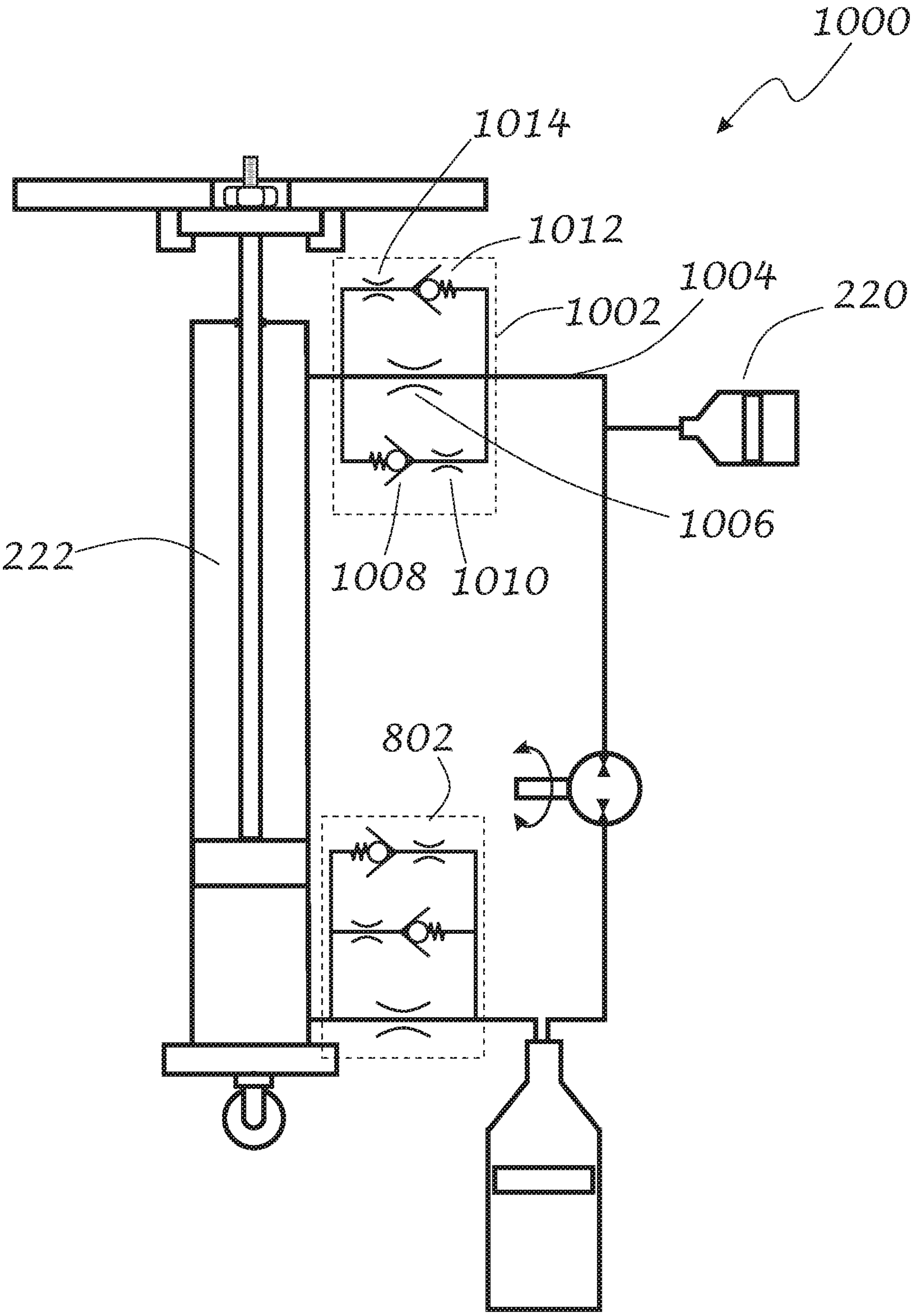
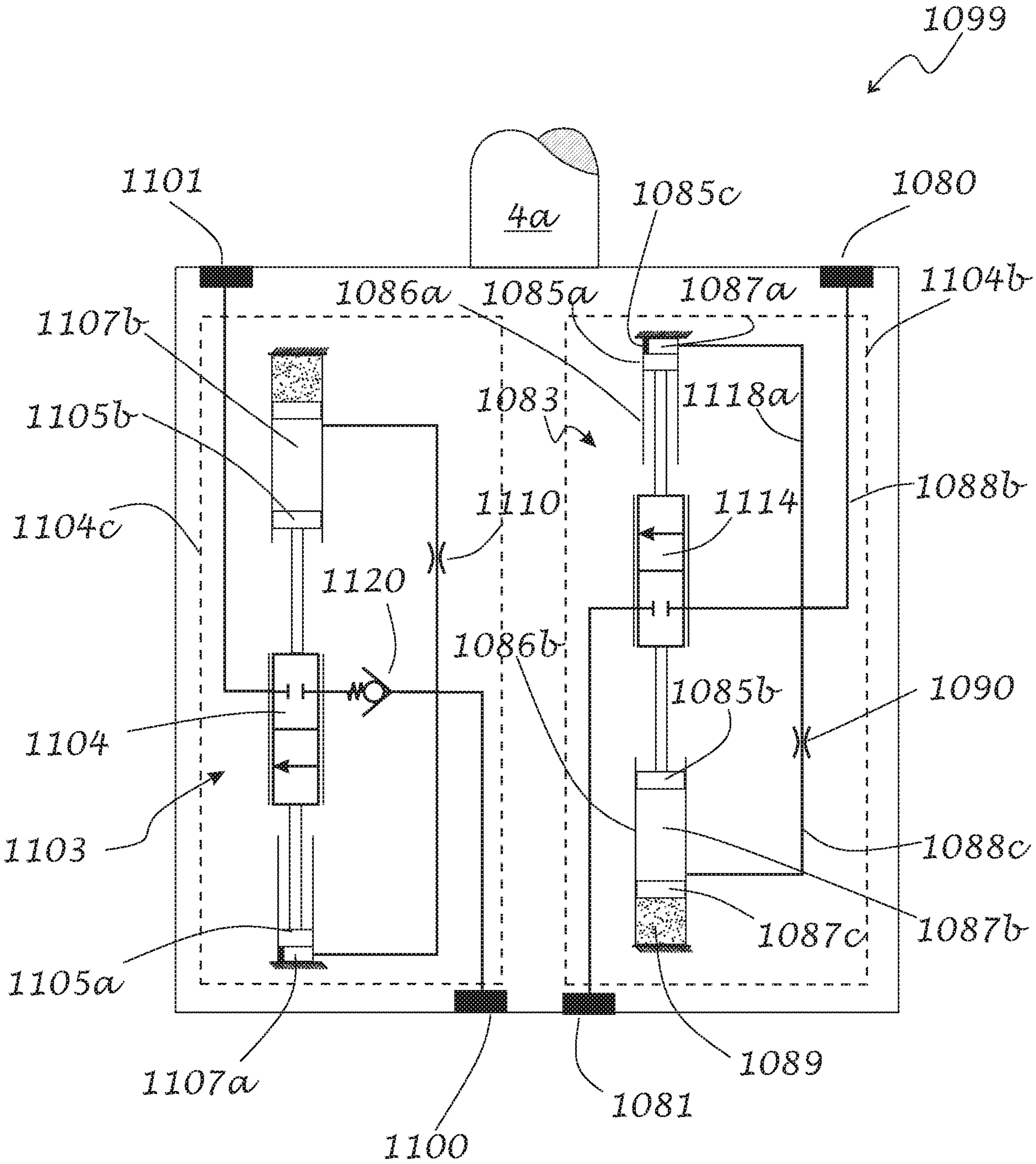


Fig. 10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1104c
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1107a
1081
1100

Fig. 11

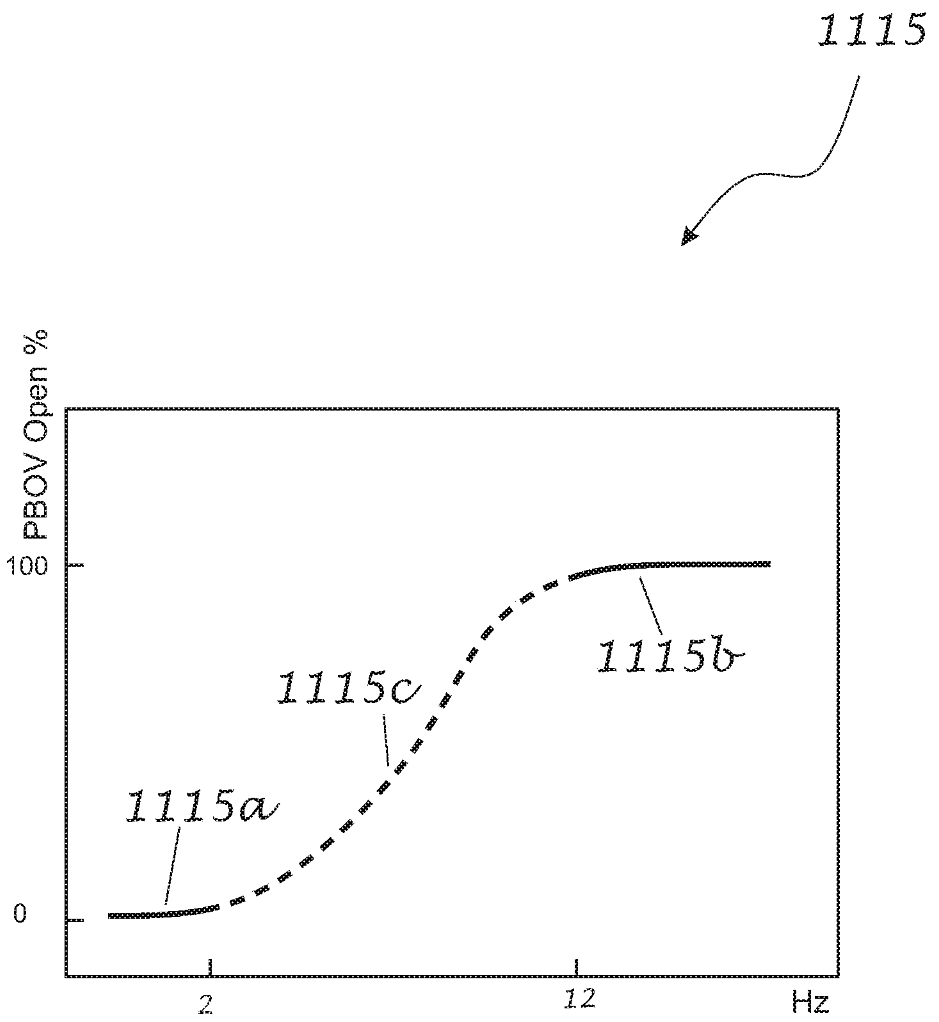

Fig. 12

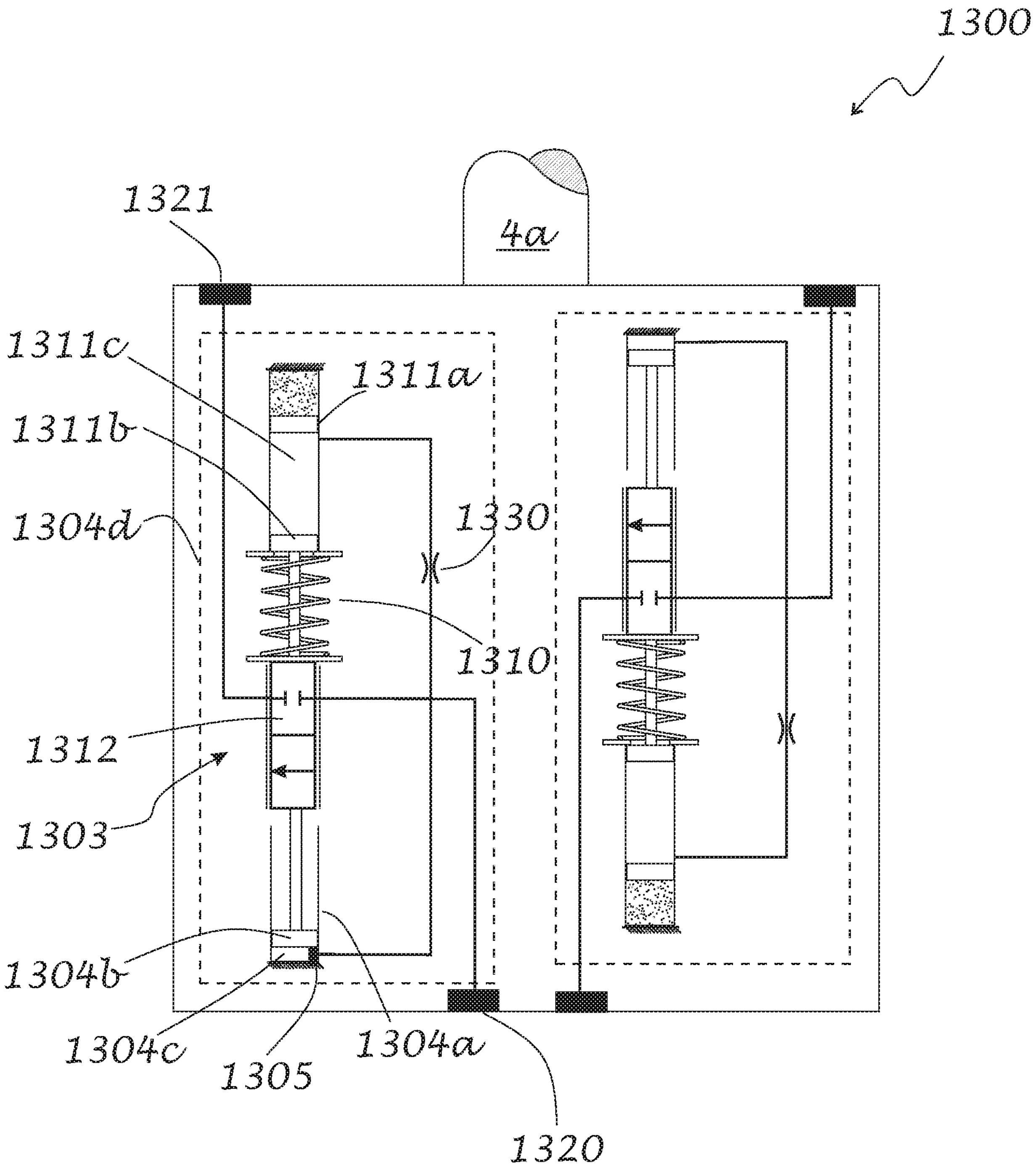
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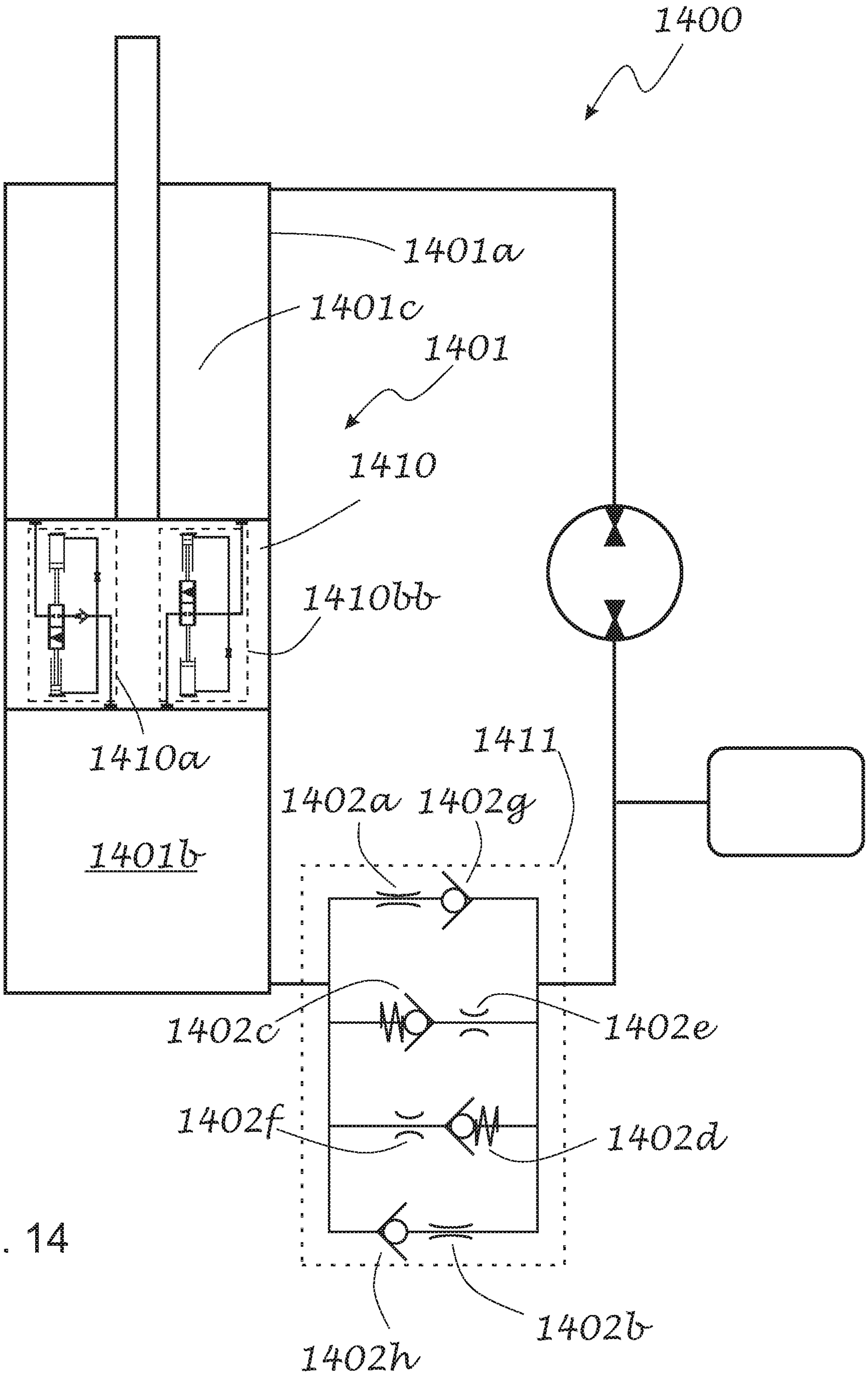
Fig. 13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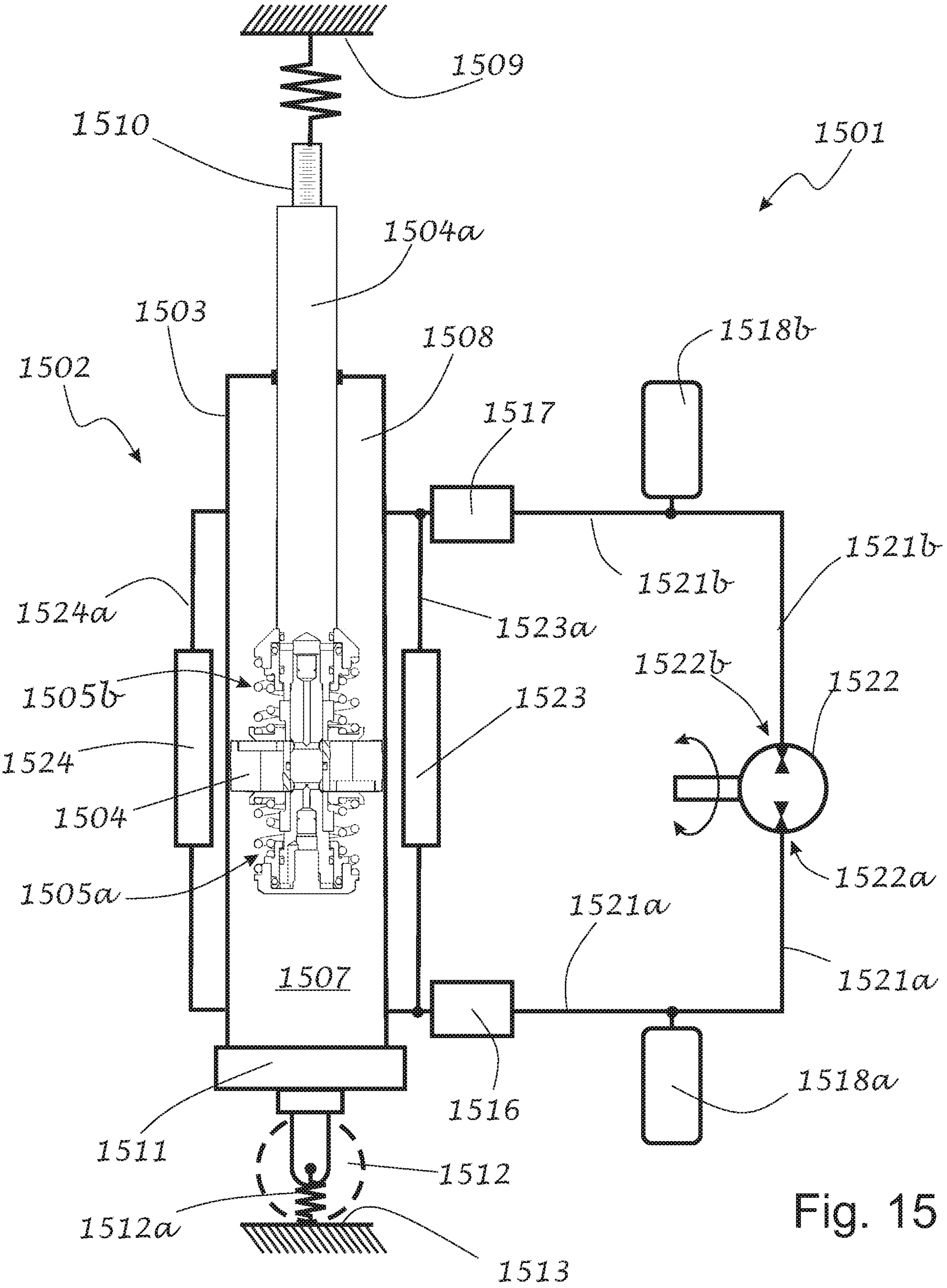
1402h
1402b

Fig. 14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1505b
1523
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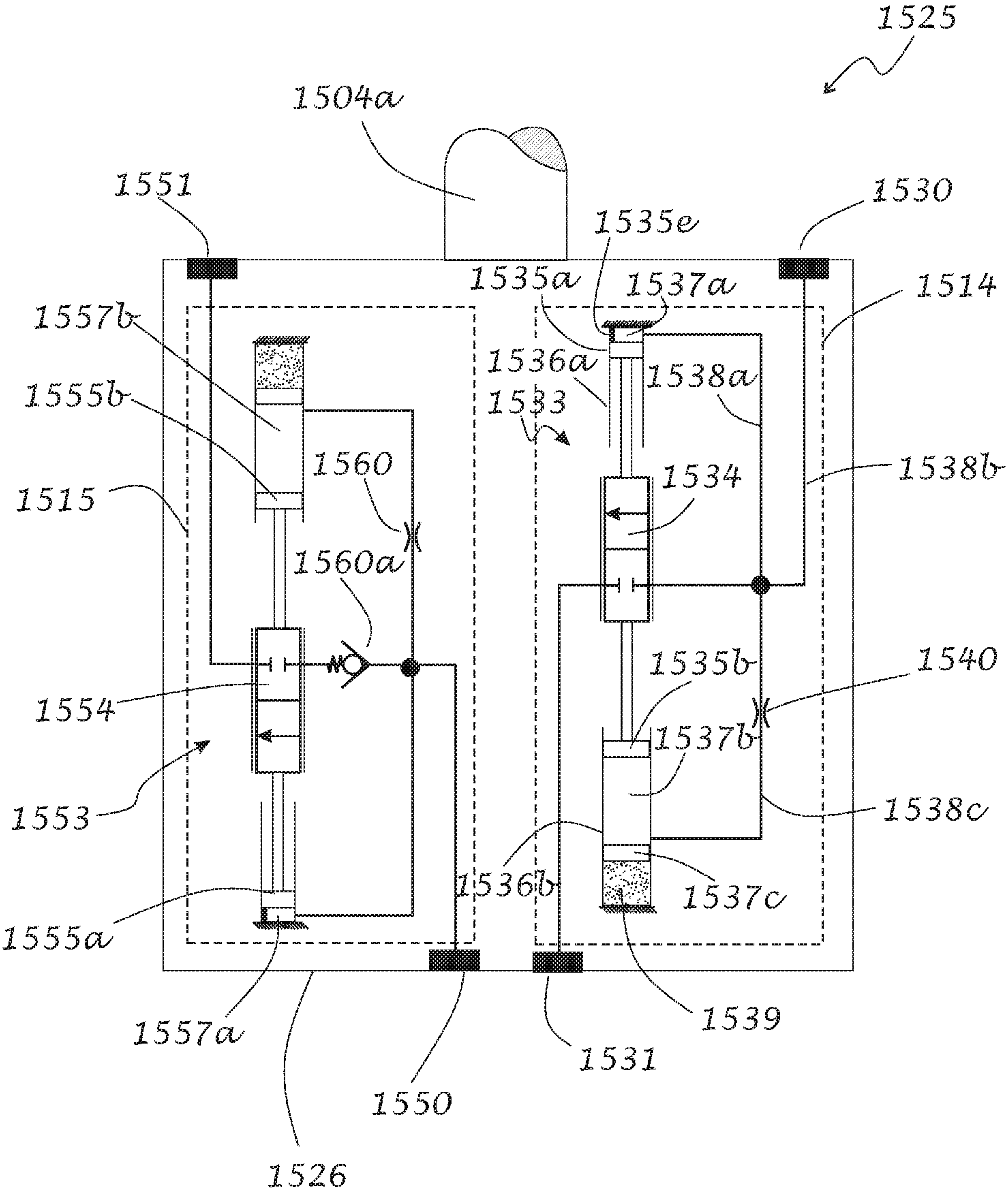
Fig. 15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1560a
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Fig. 16

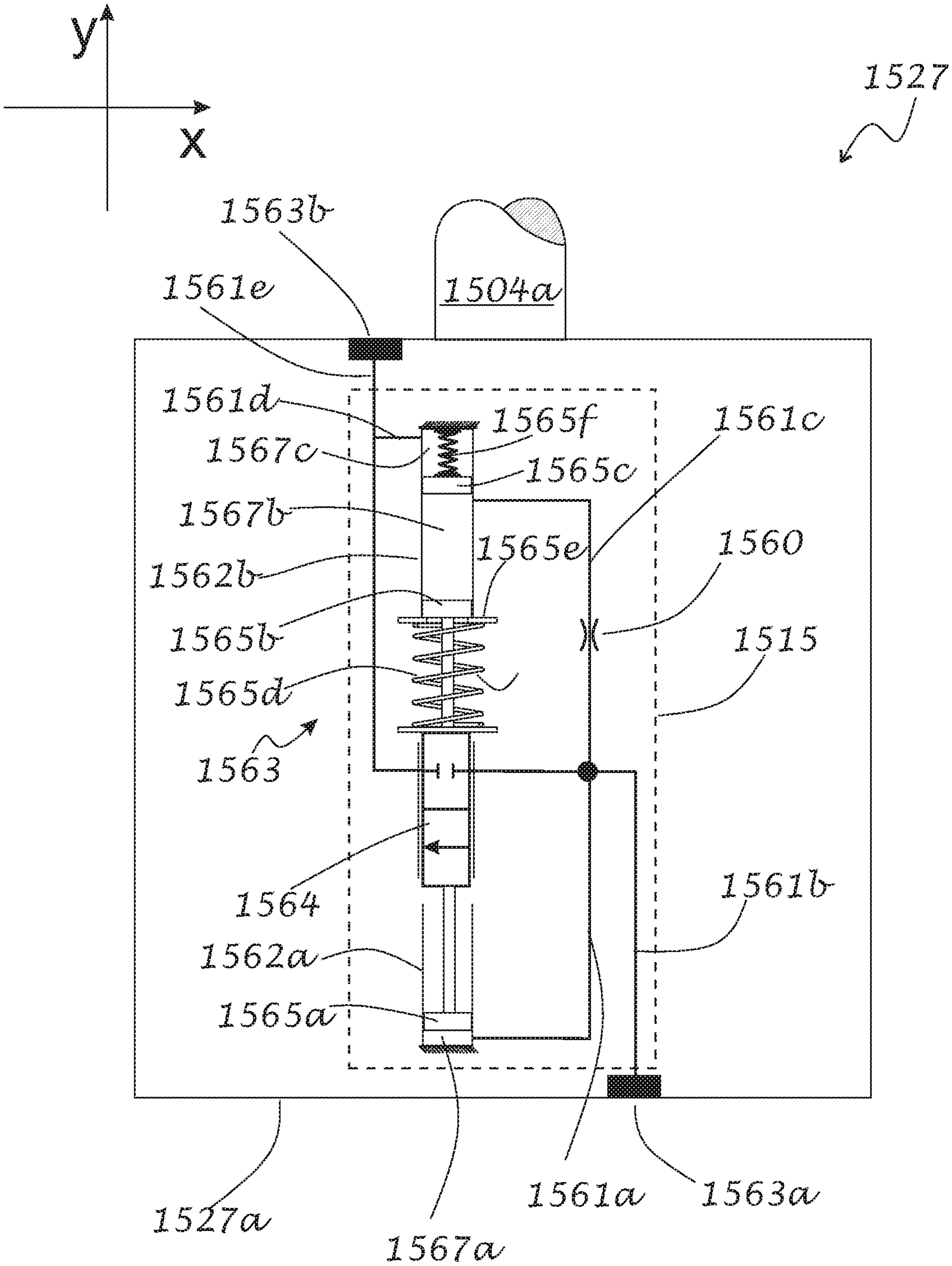
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Fig. 17

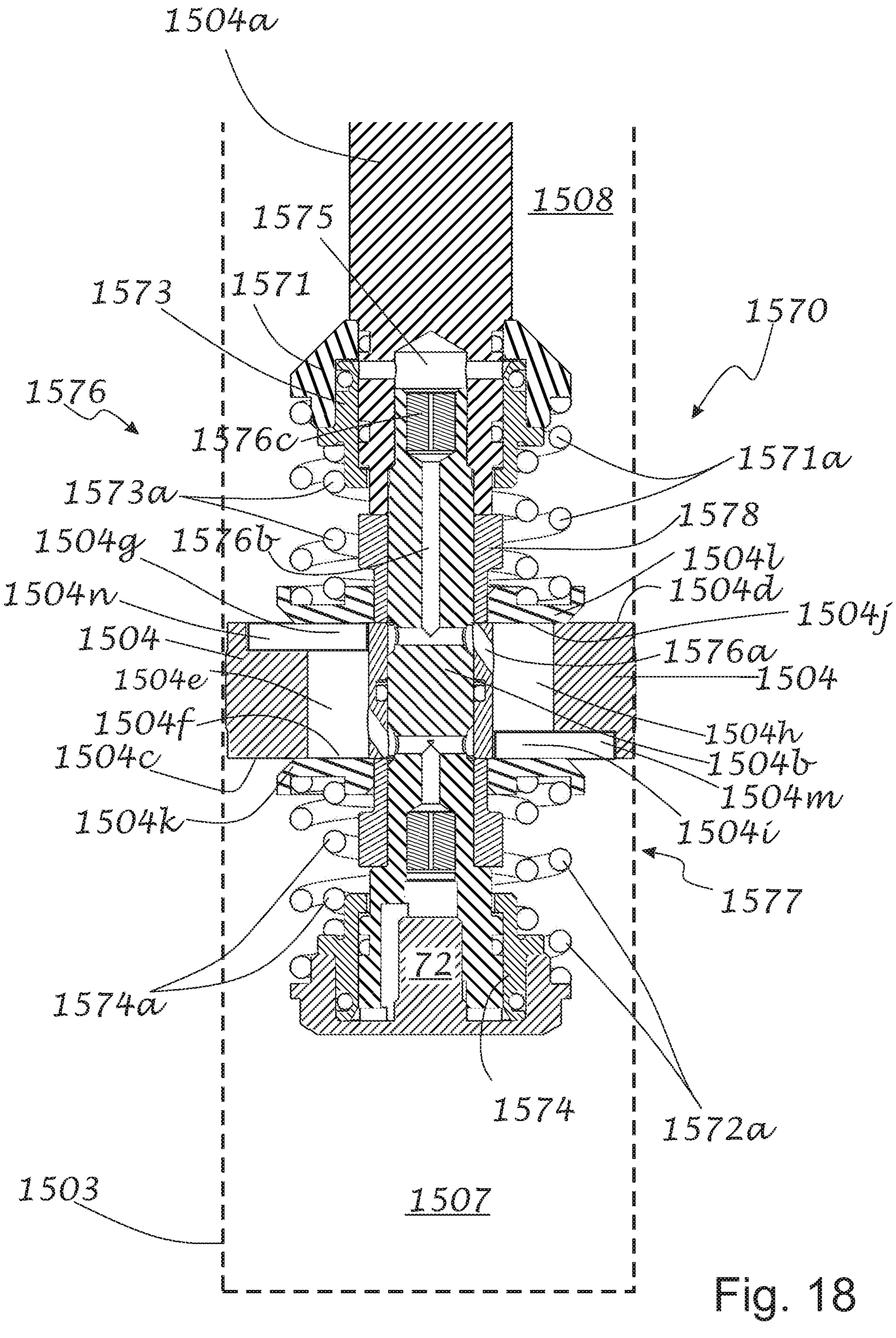
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Fig. 18

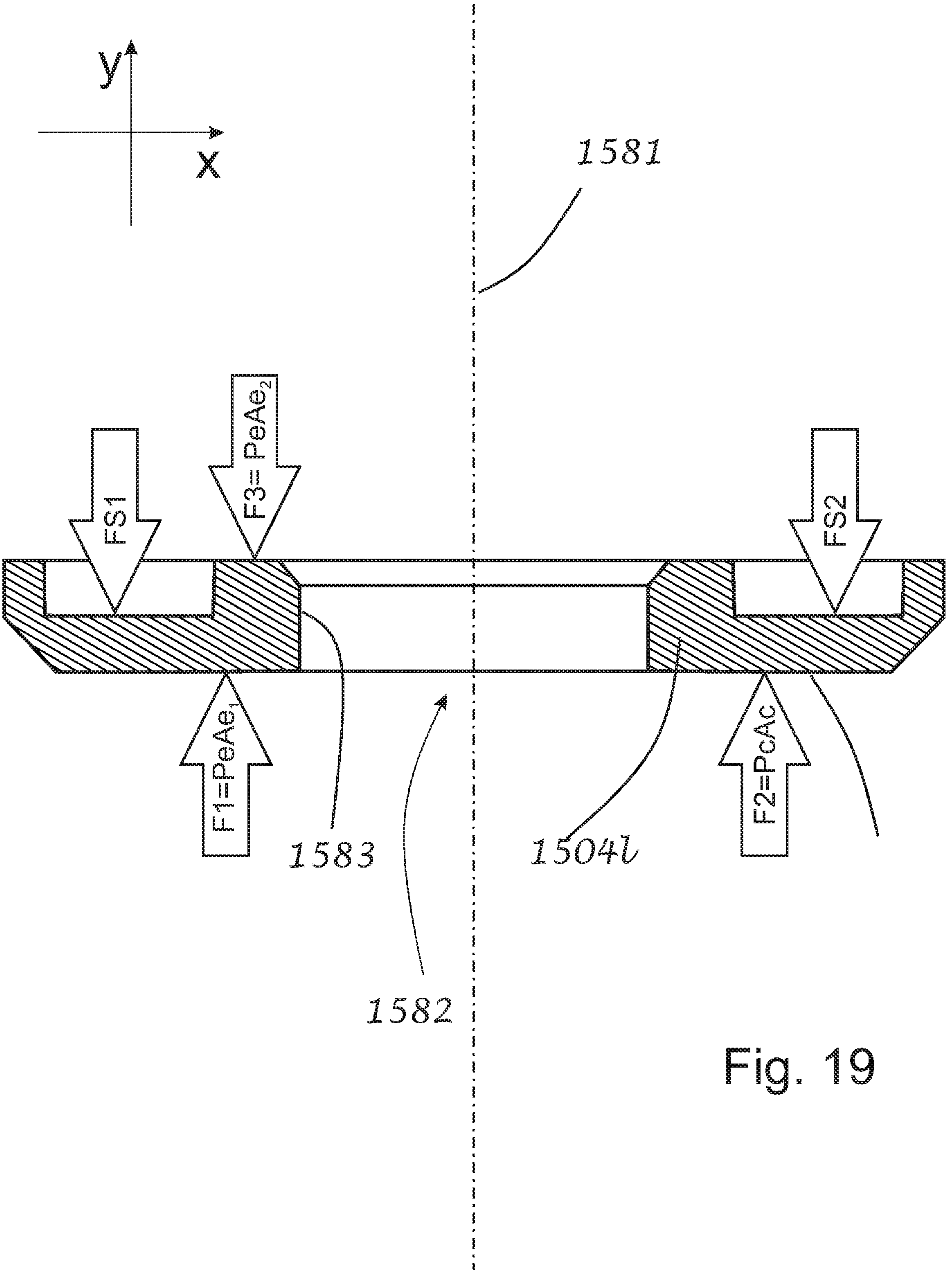

Fig. 19

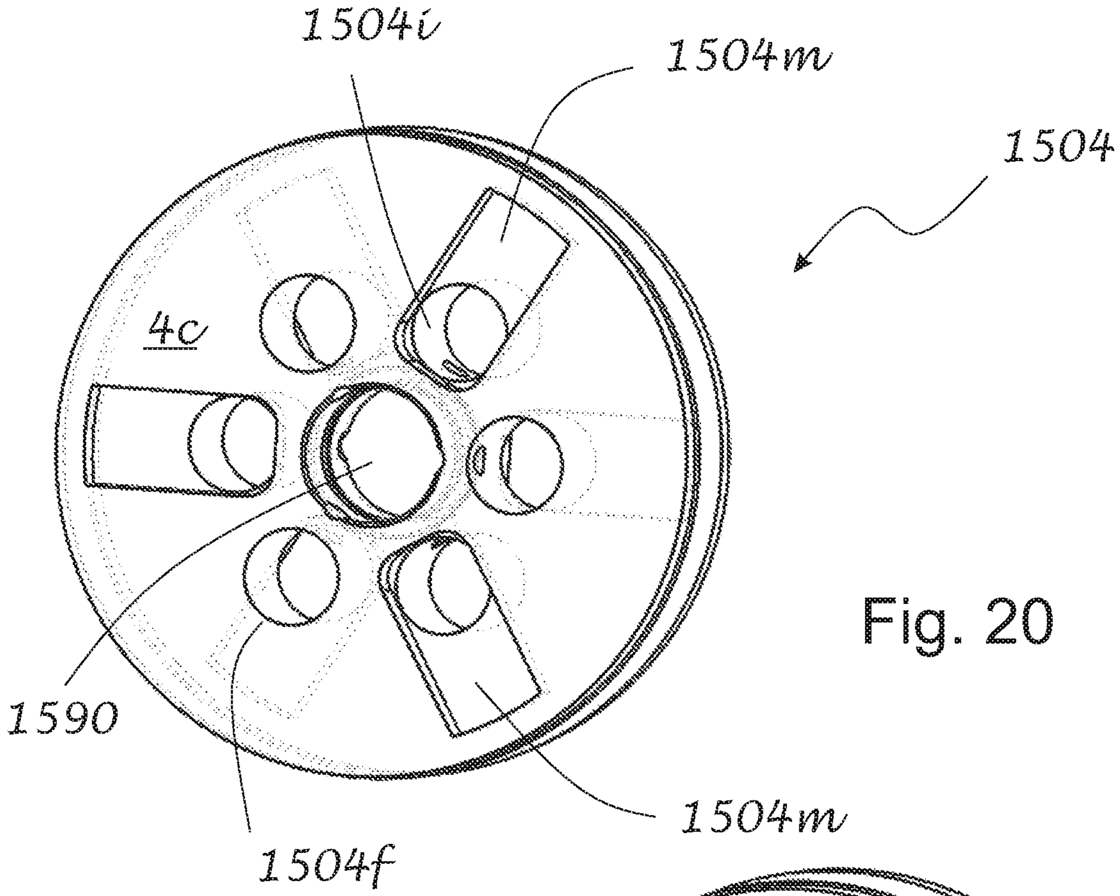
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Fig. 20

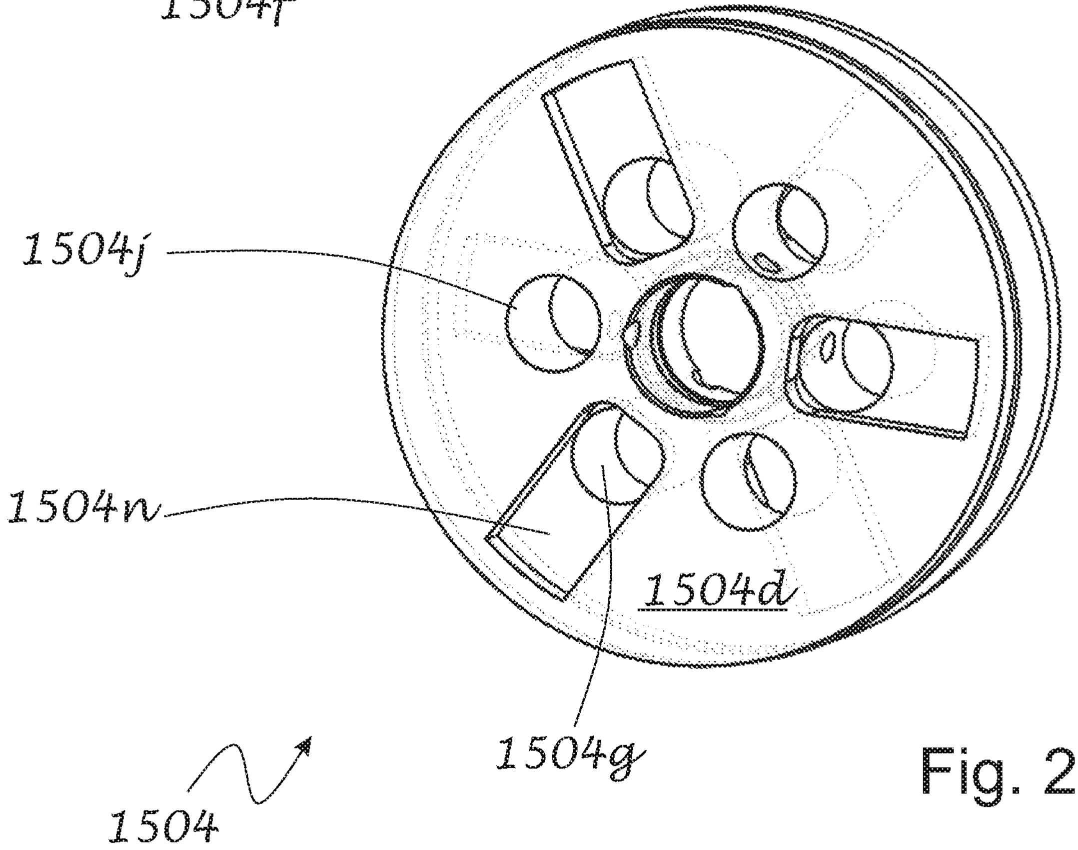
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Fig. 21

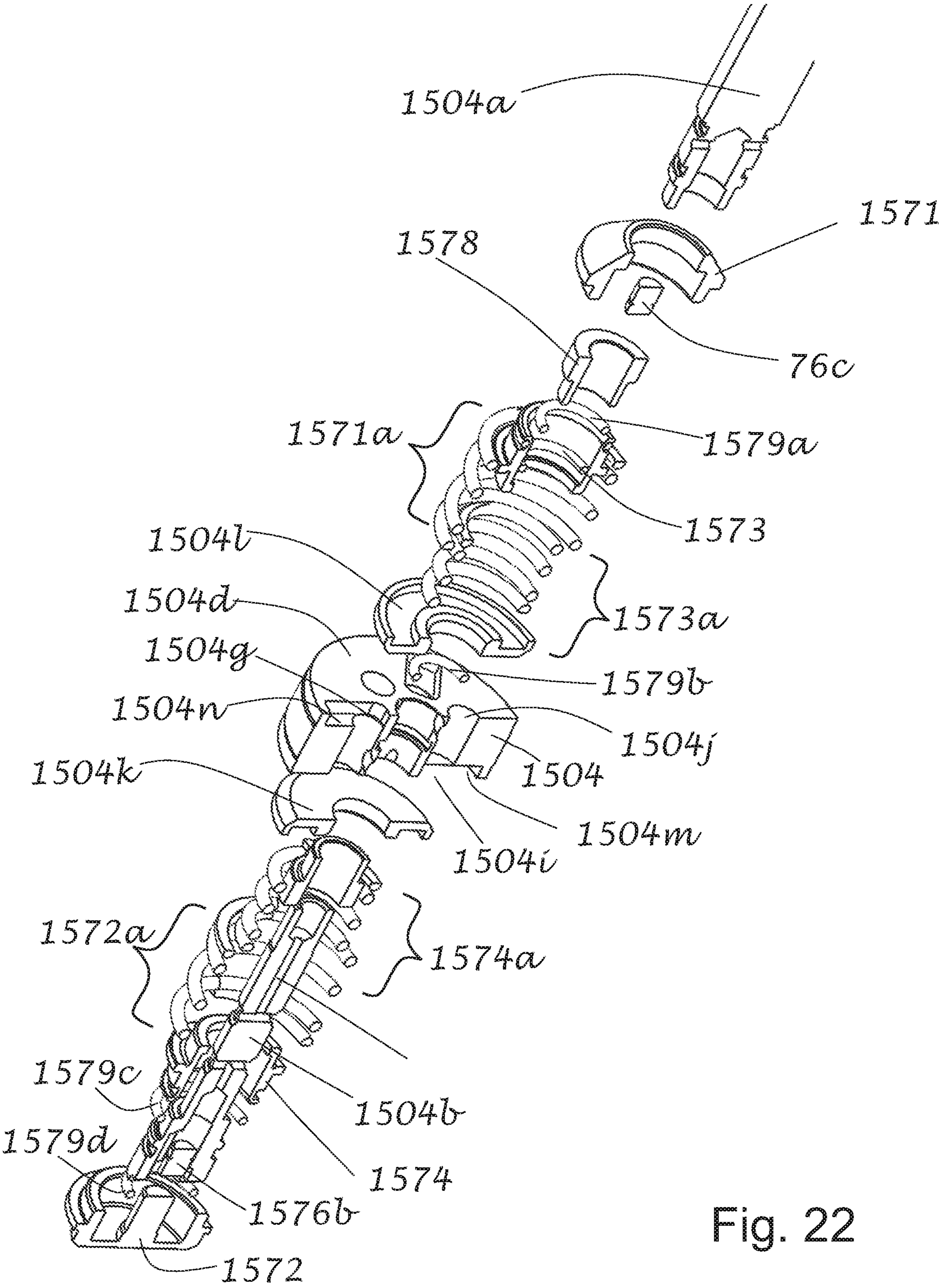
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Fig. 22

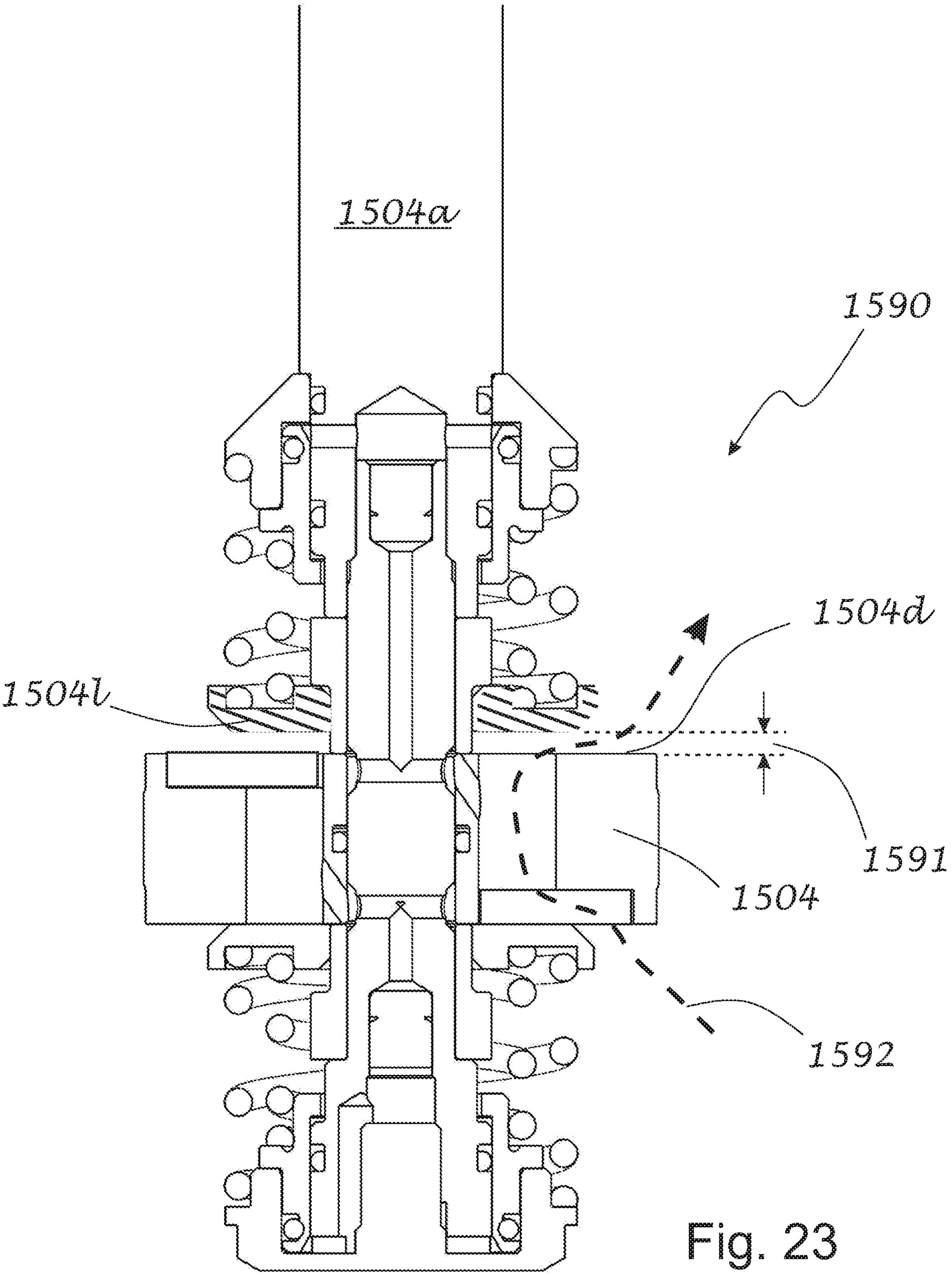

Fig. 23

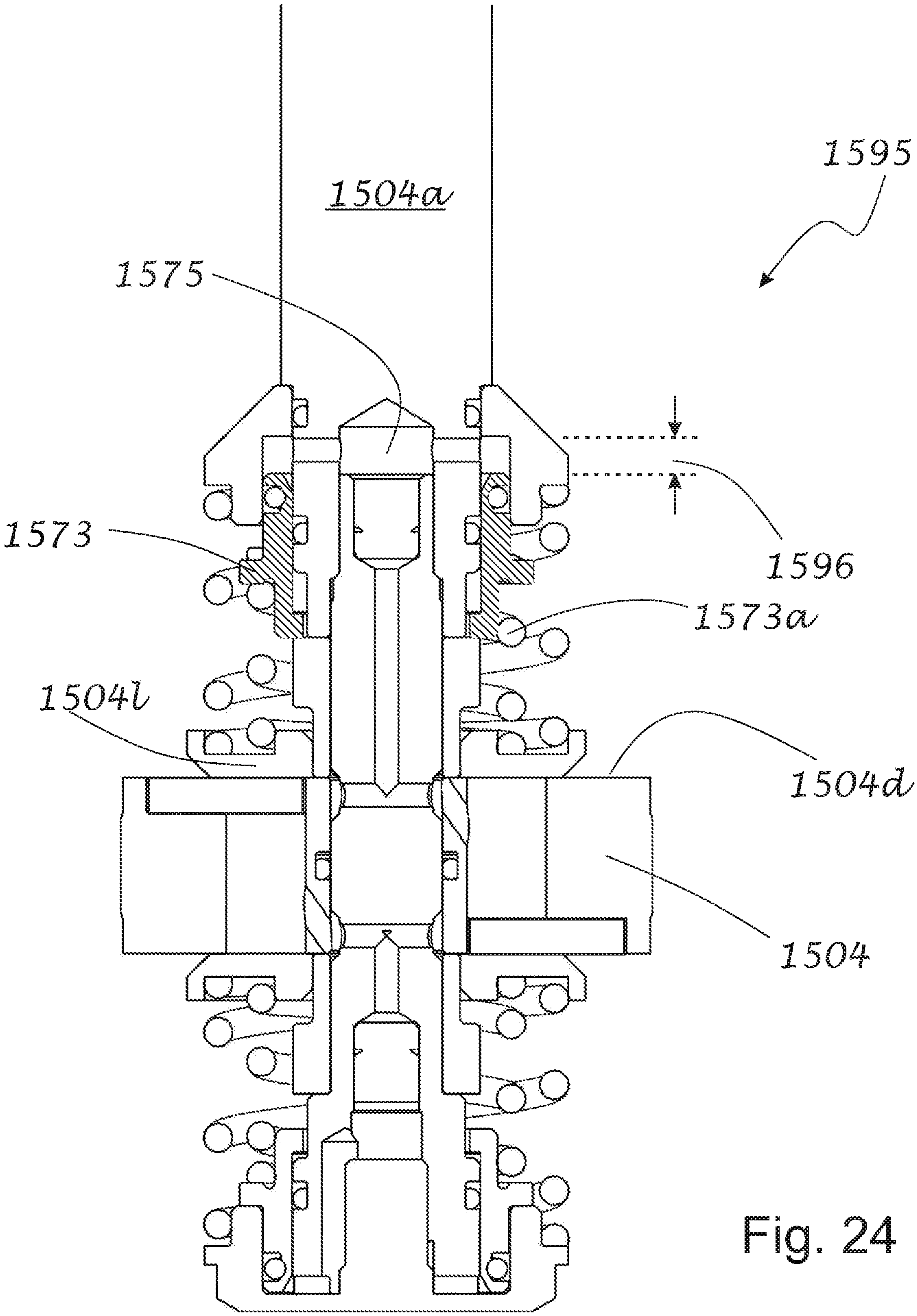

Fig. 24

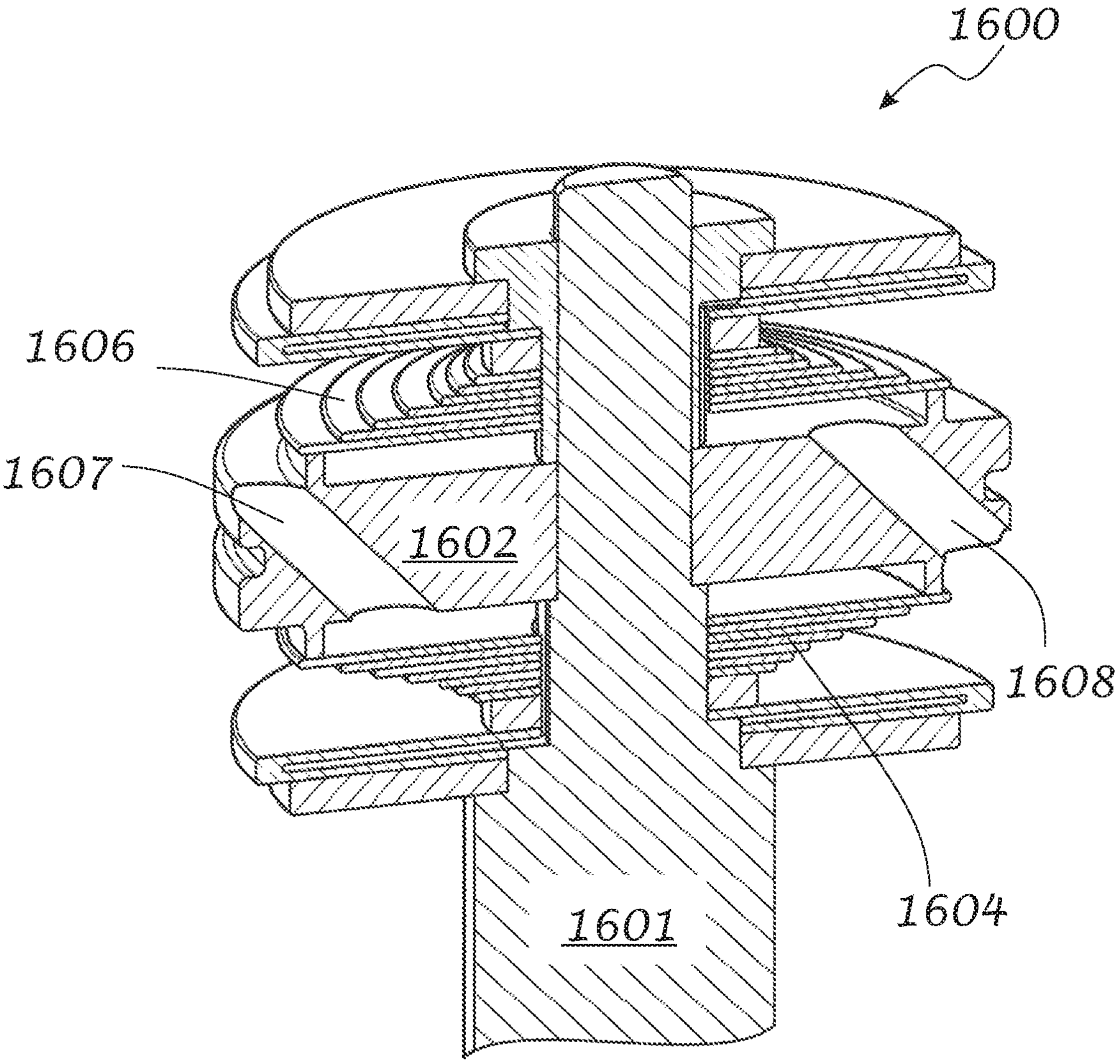

Fig. 25

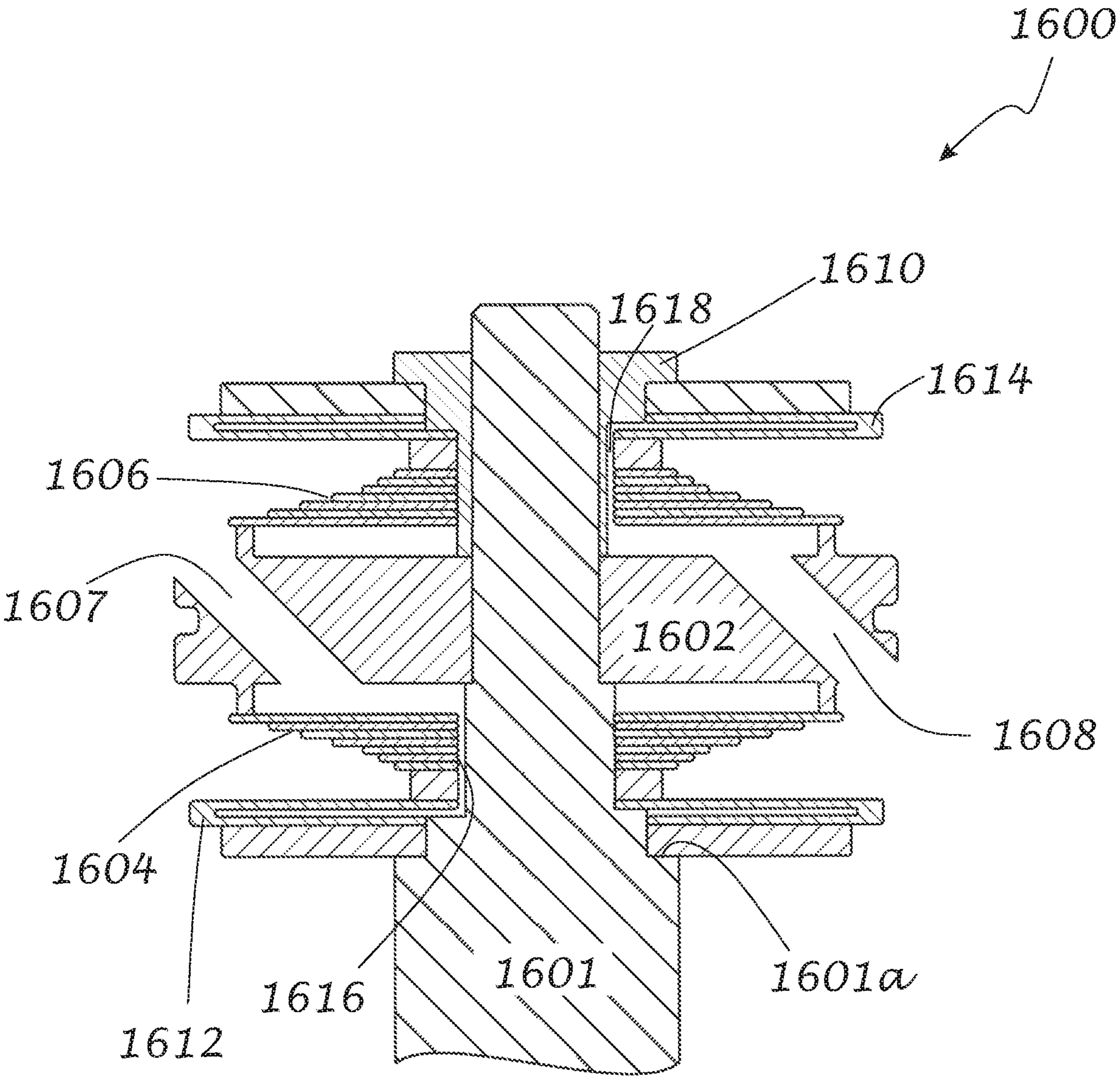
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Fig. 26

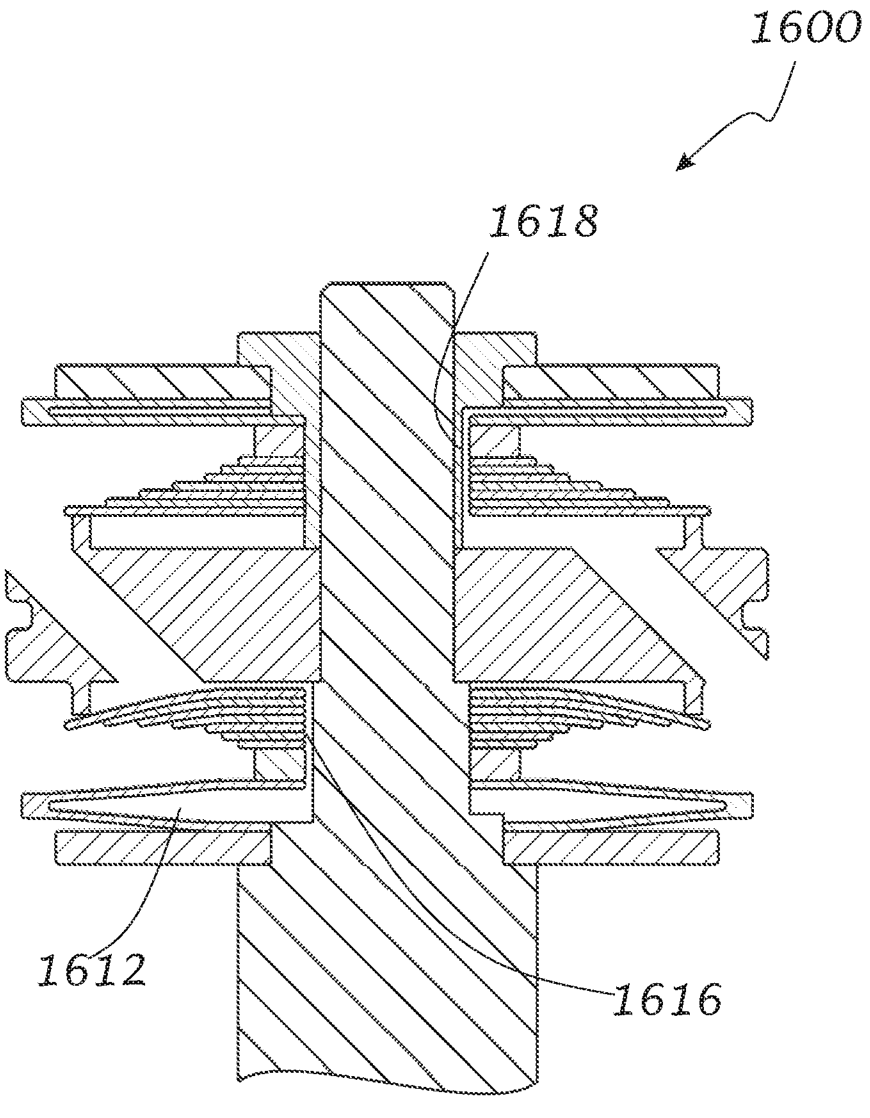

Fig. 27

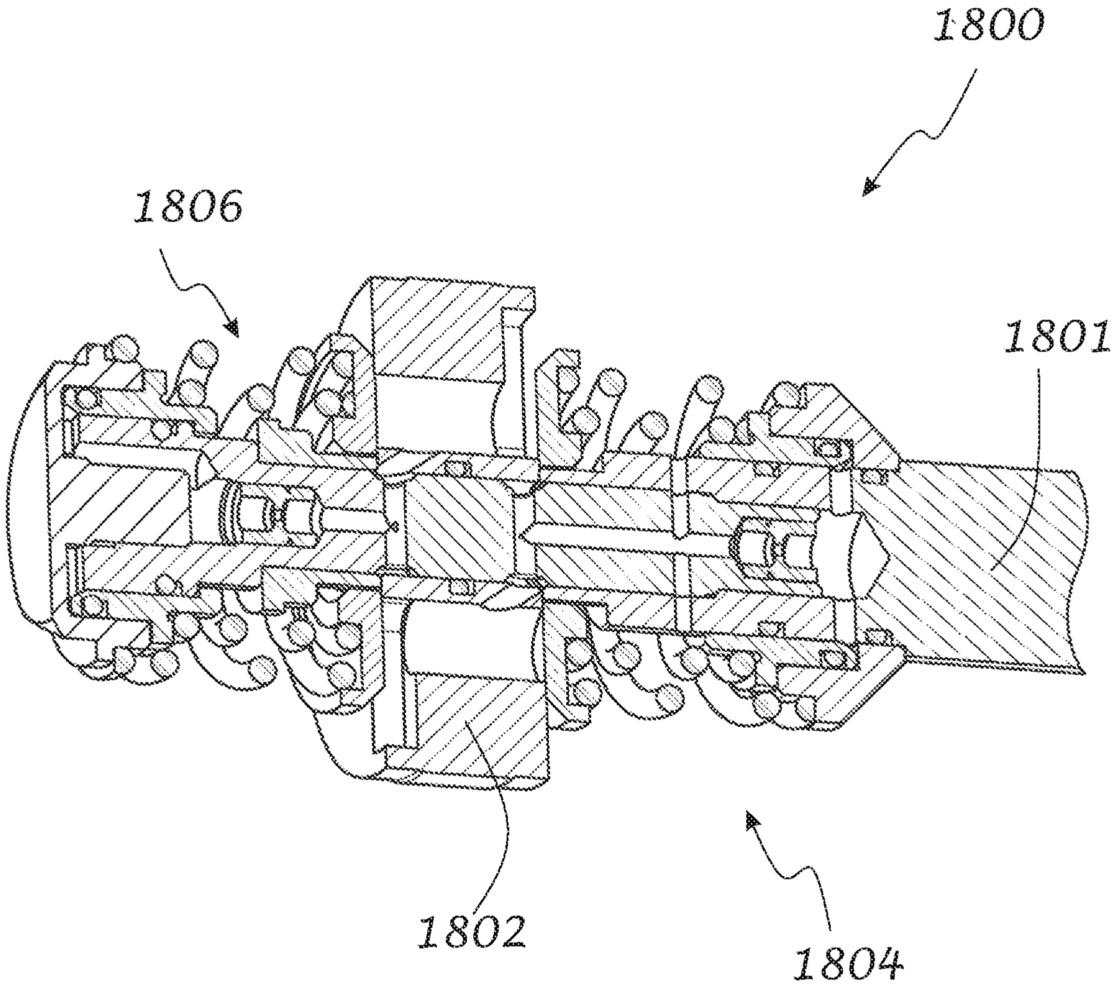

Fig. 28

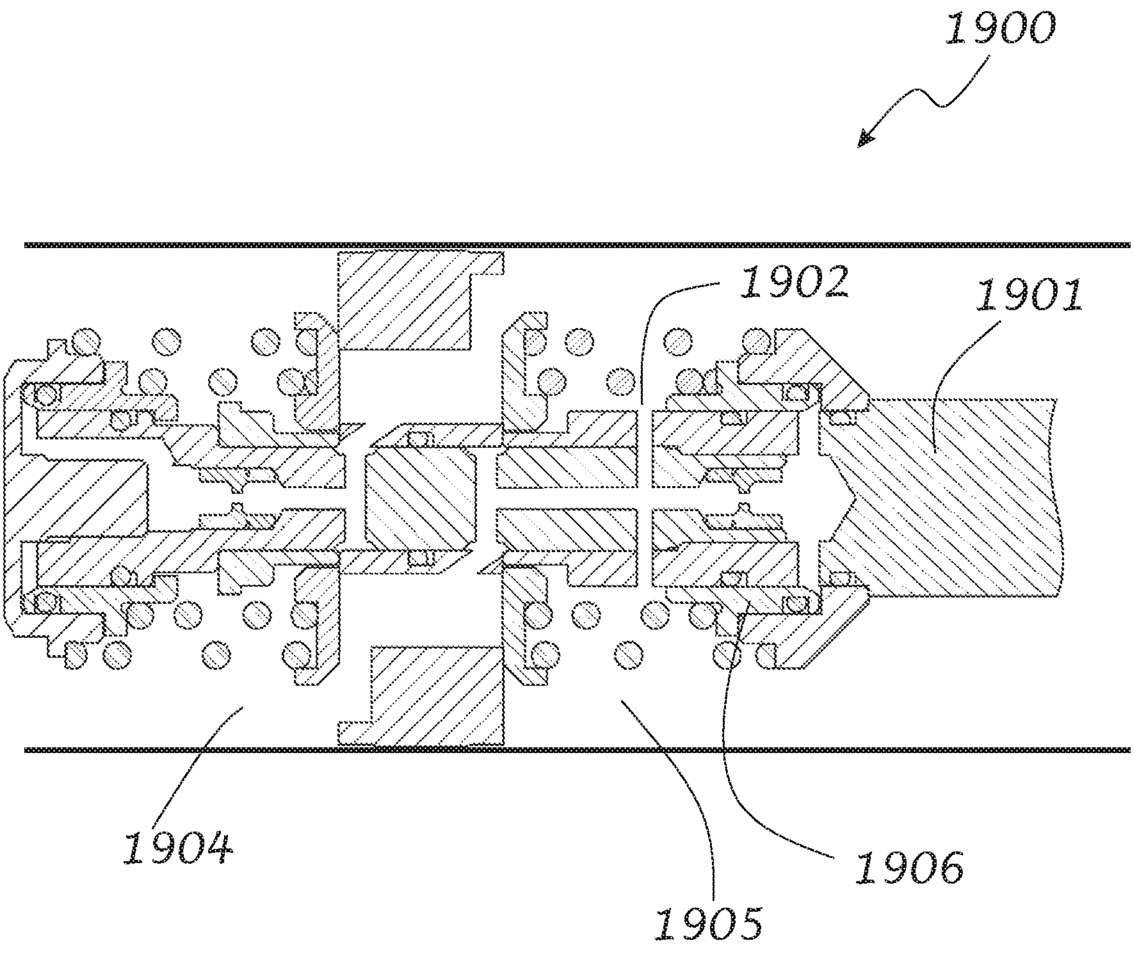

Fig. 29

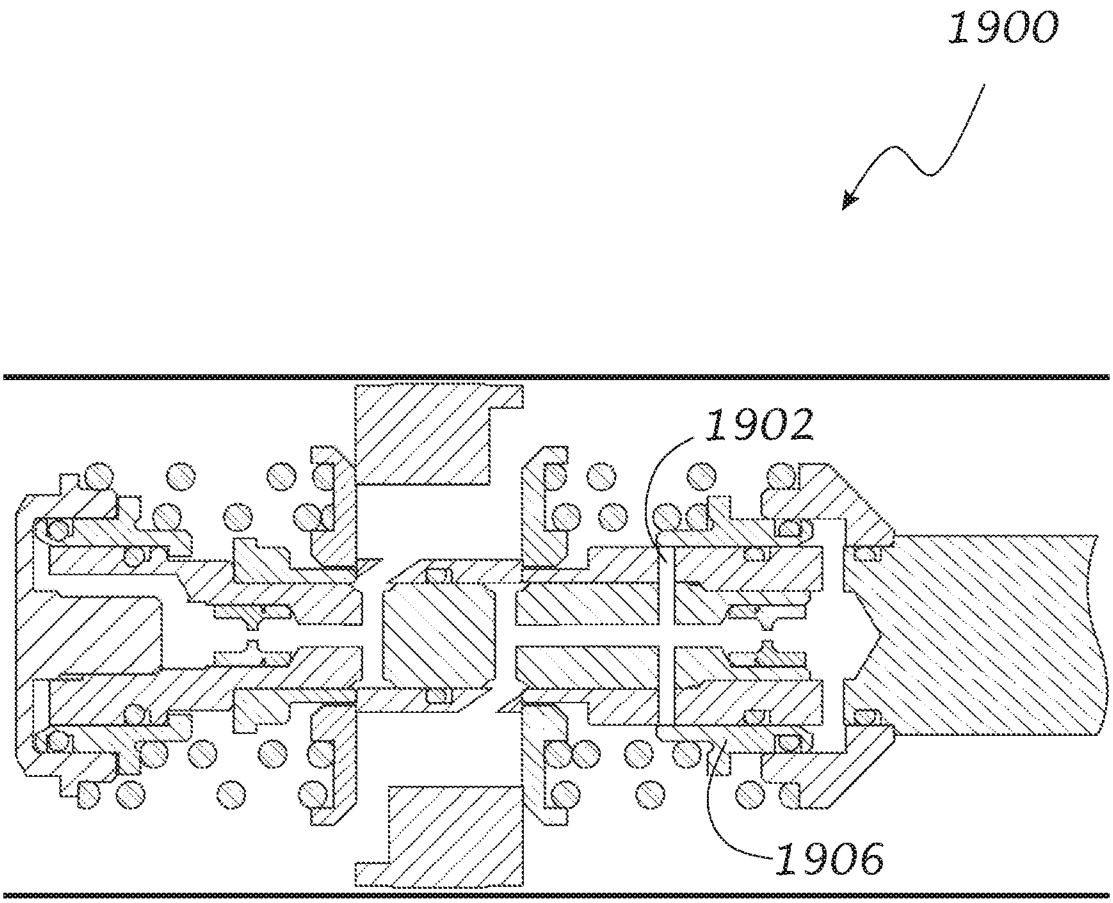

Fig. 30

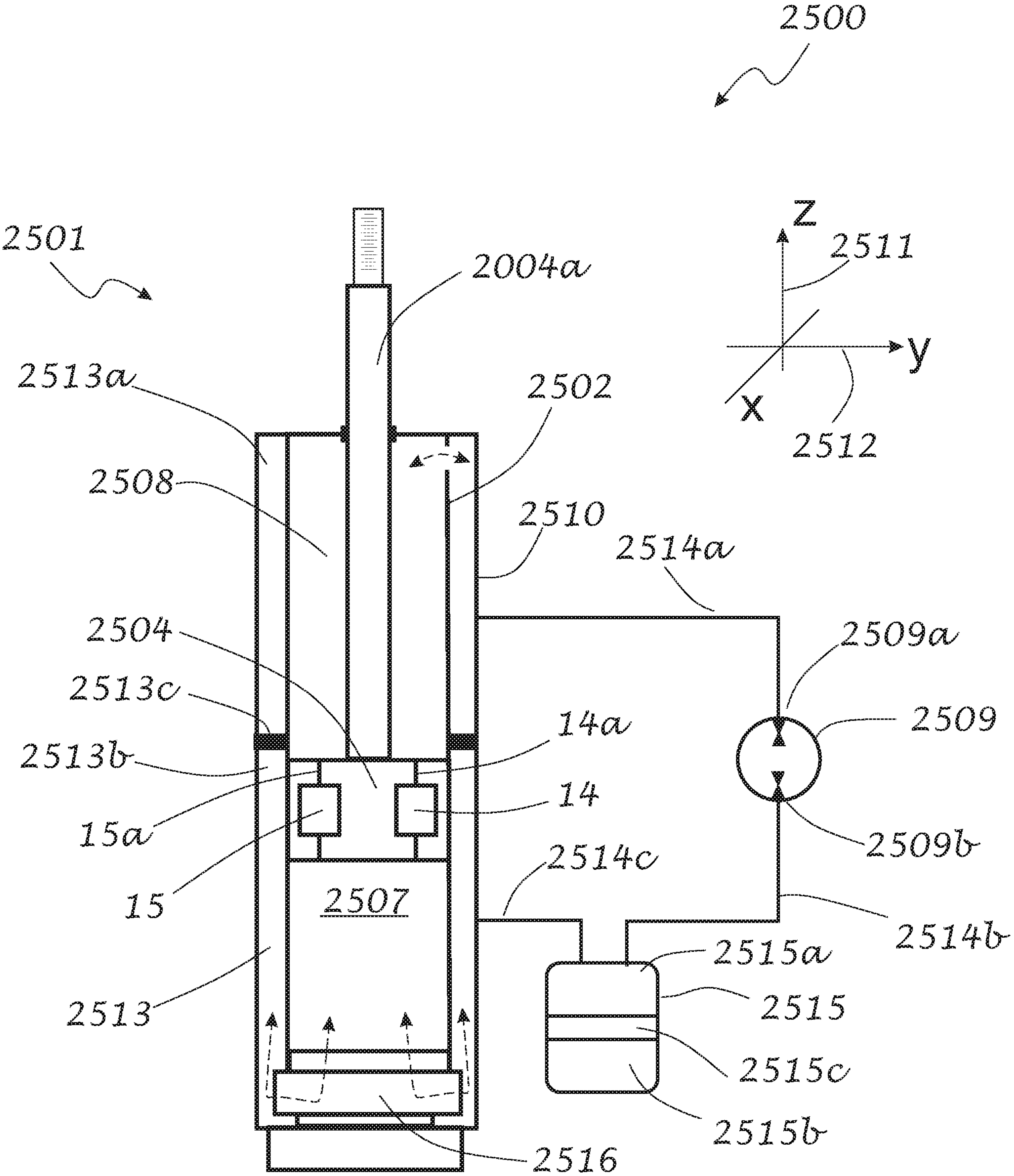
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Fig. 31

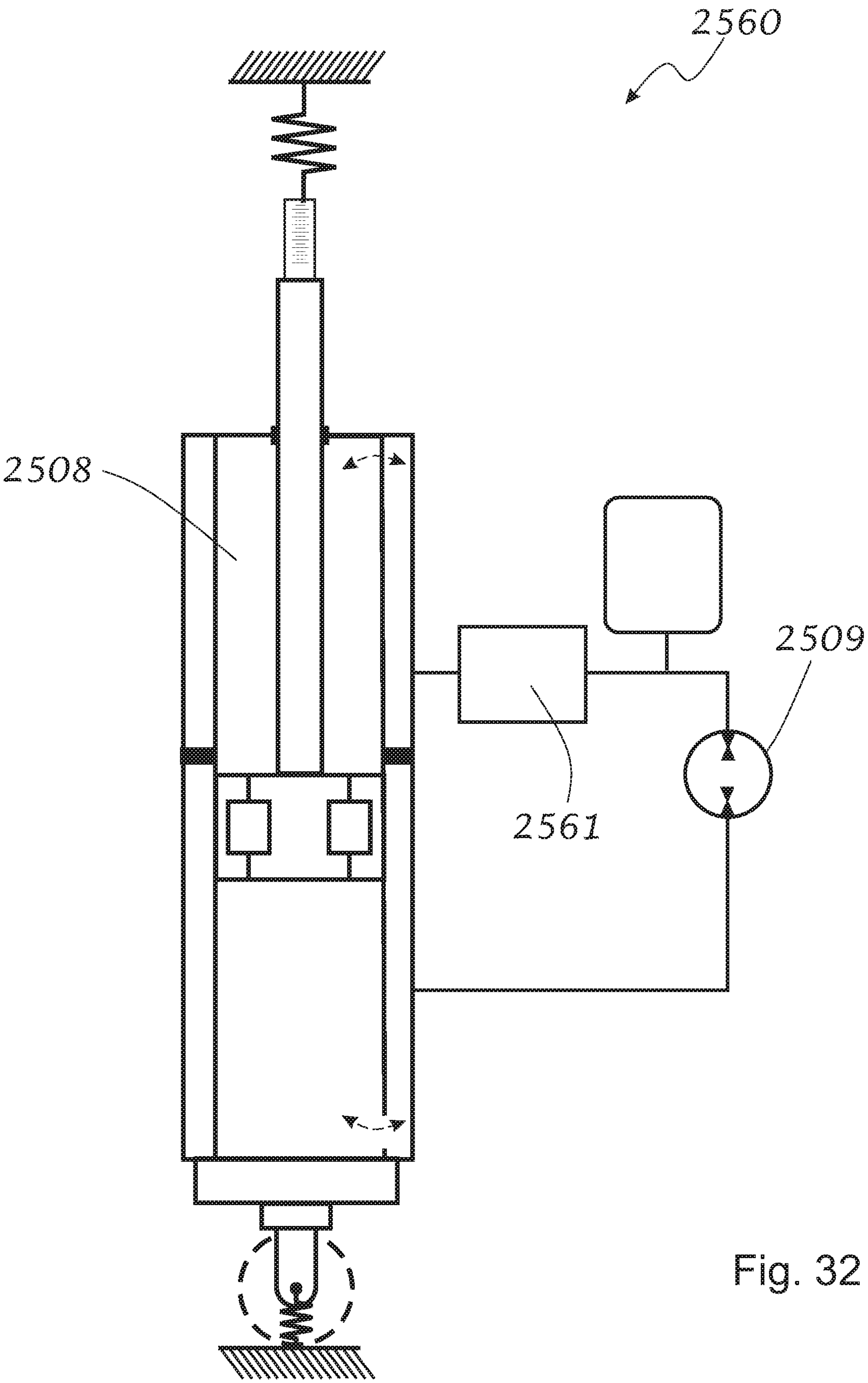
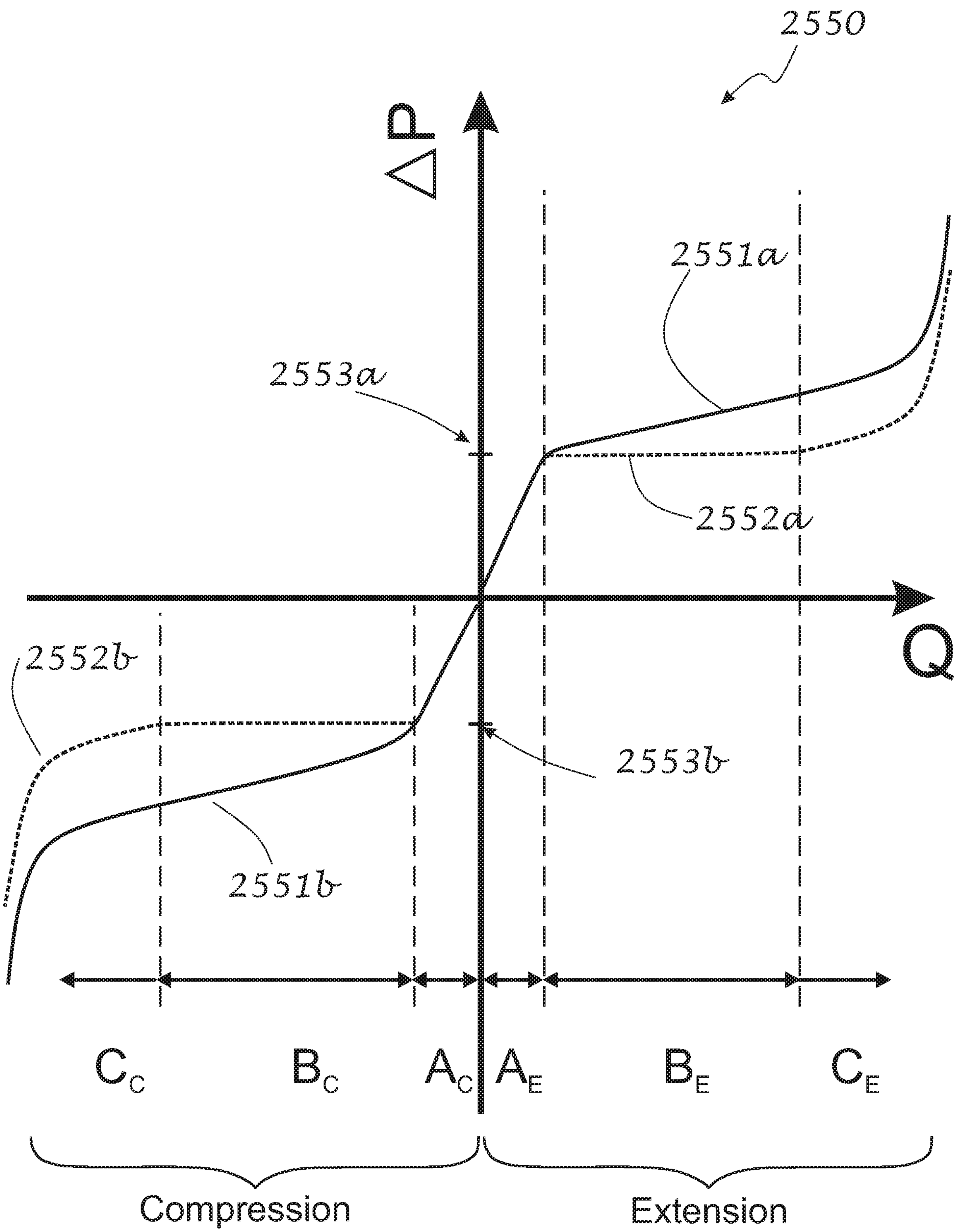


Fig. 32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B_C
A_C
A_E
B_E
C_E
Compression
Extension

Fig. 33

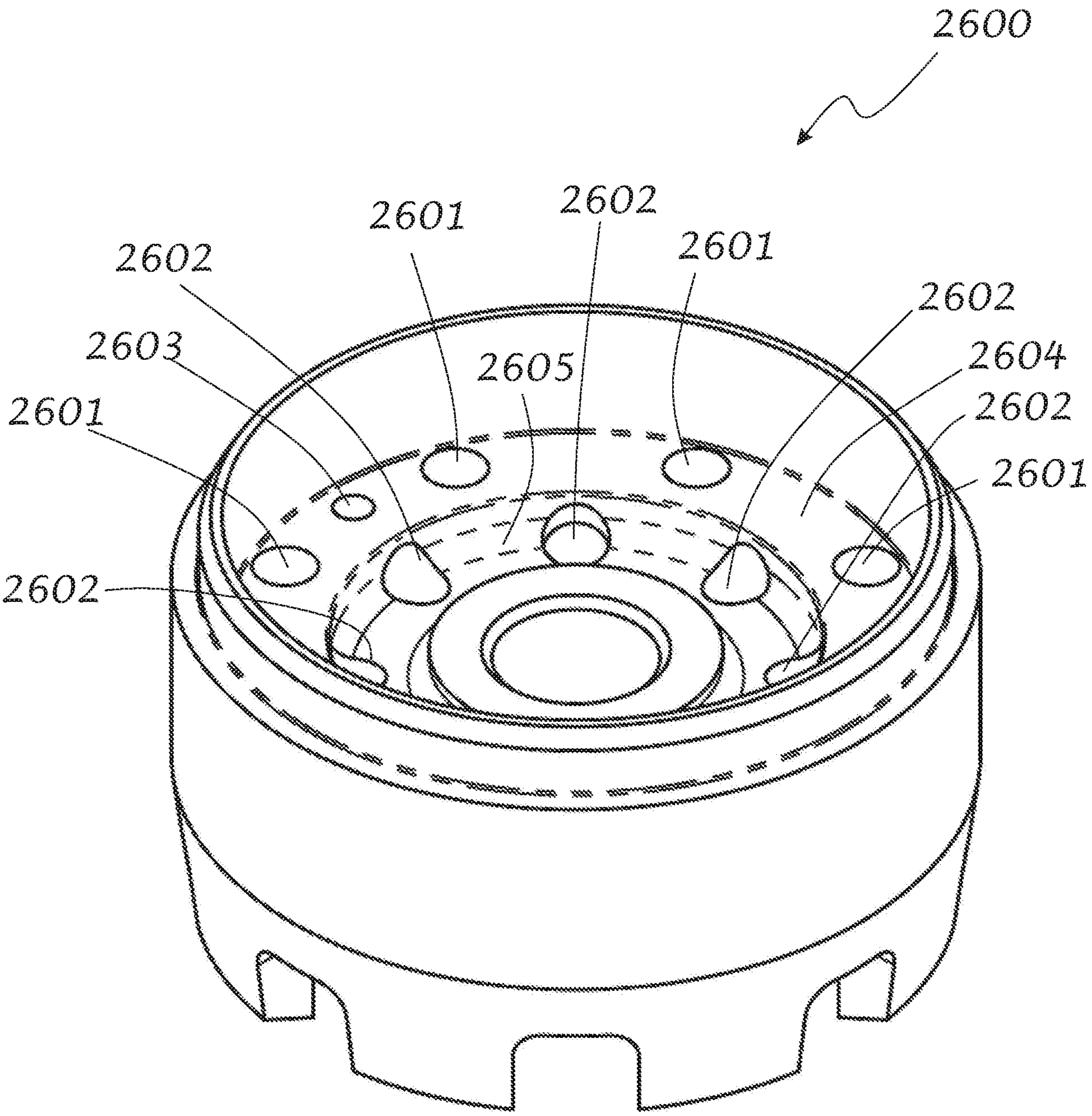
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Fig. 34

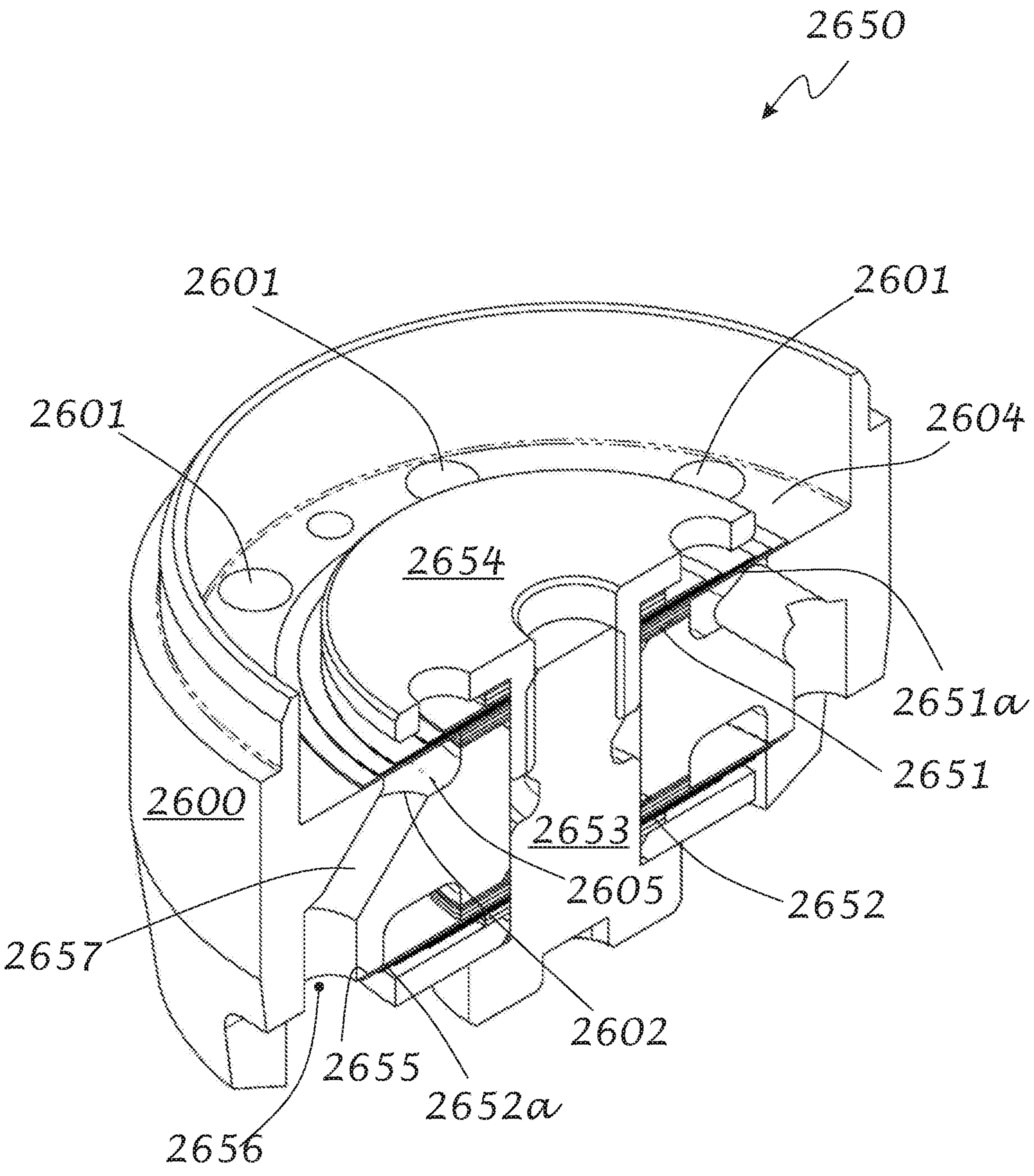
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Fig. 35

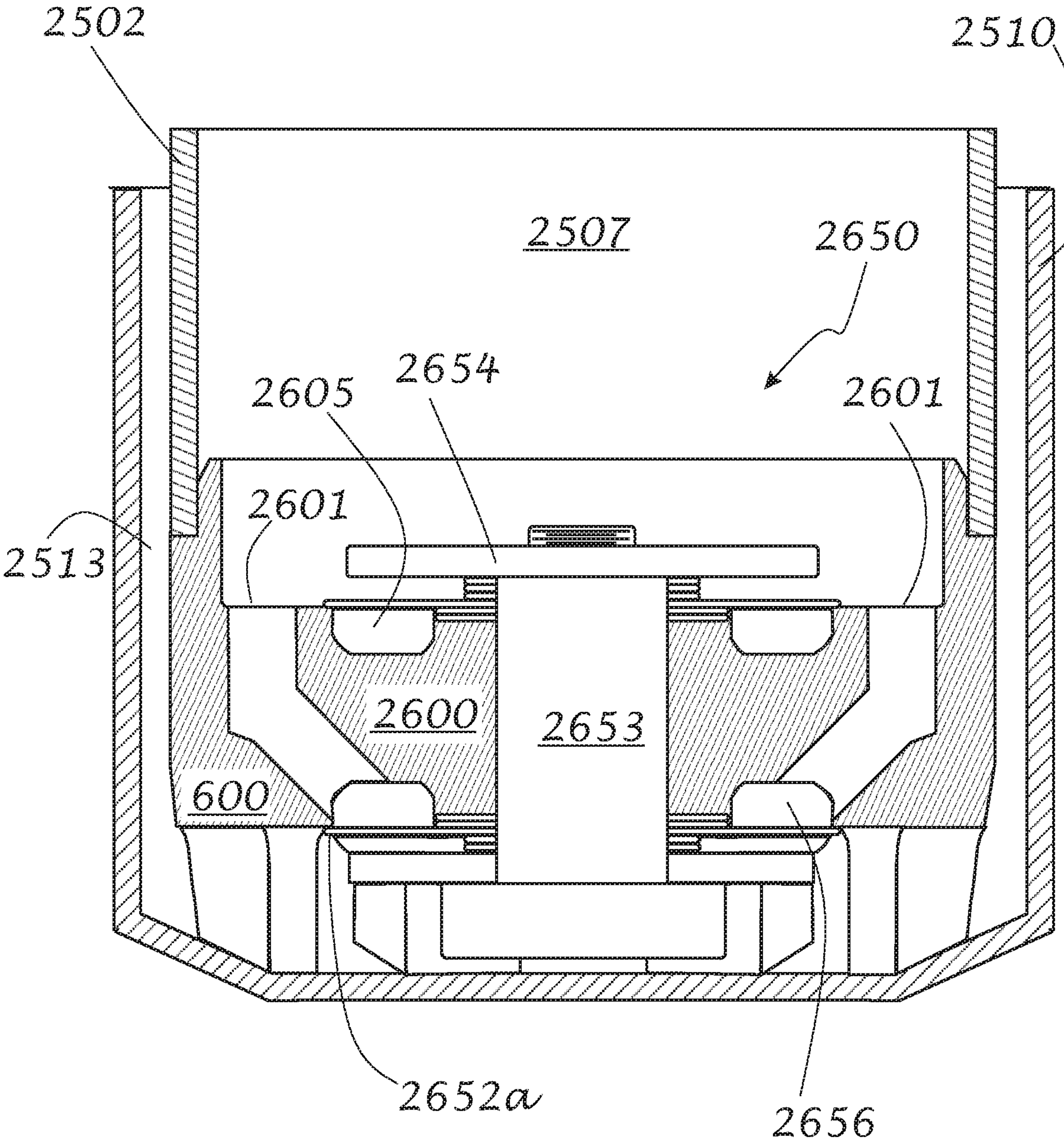
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Fig. 36

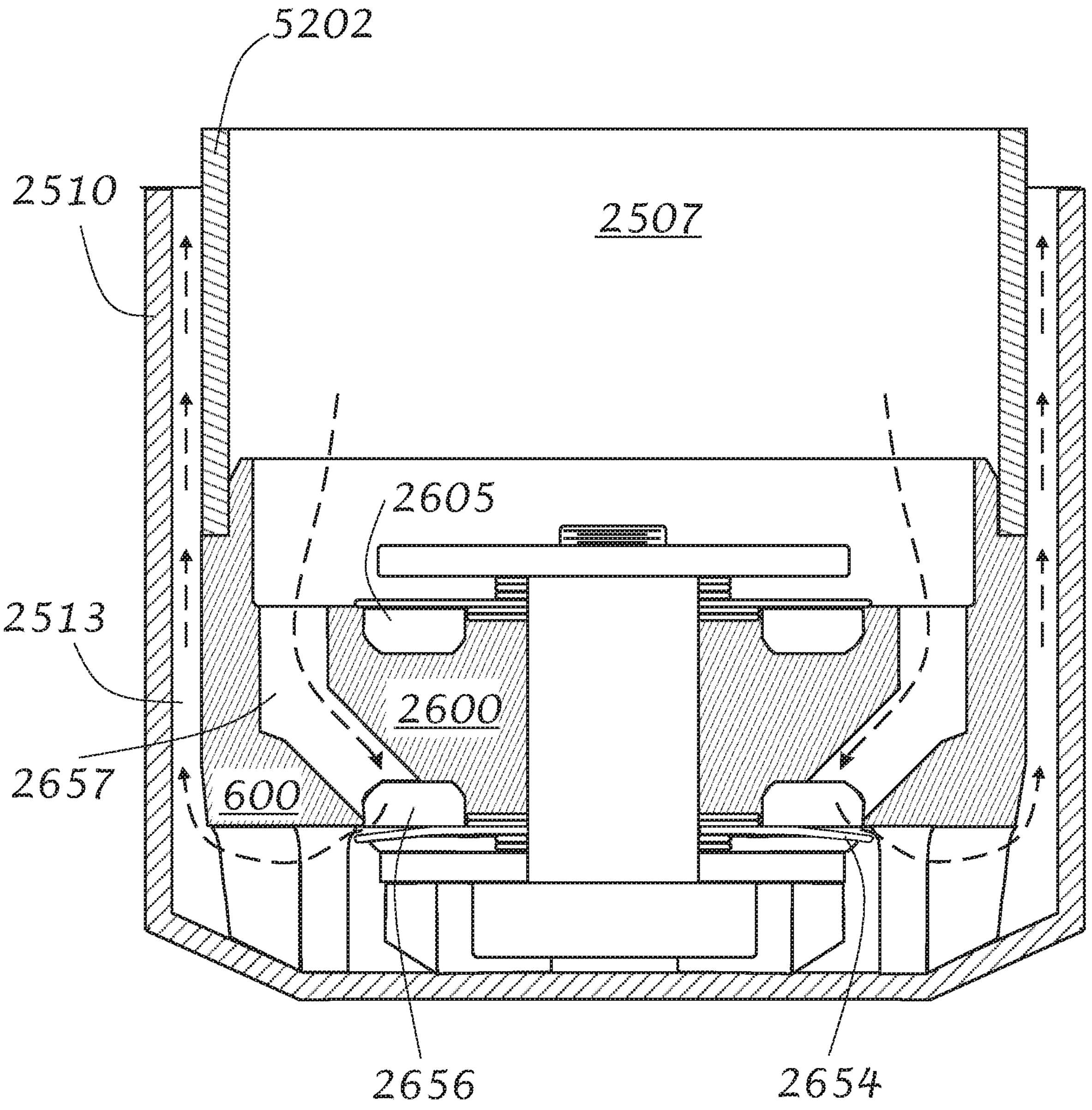

Fig. 37

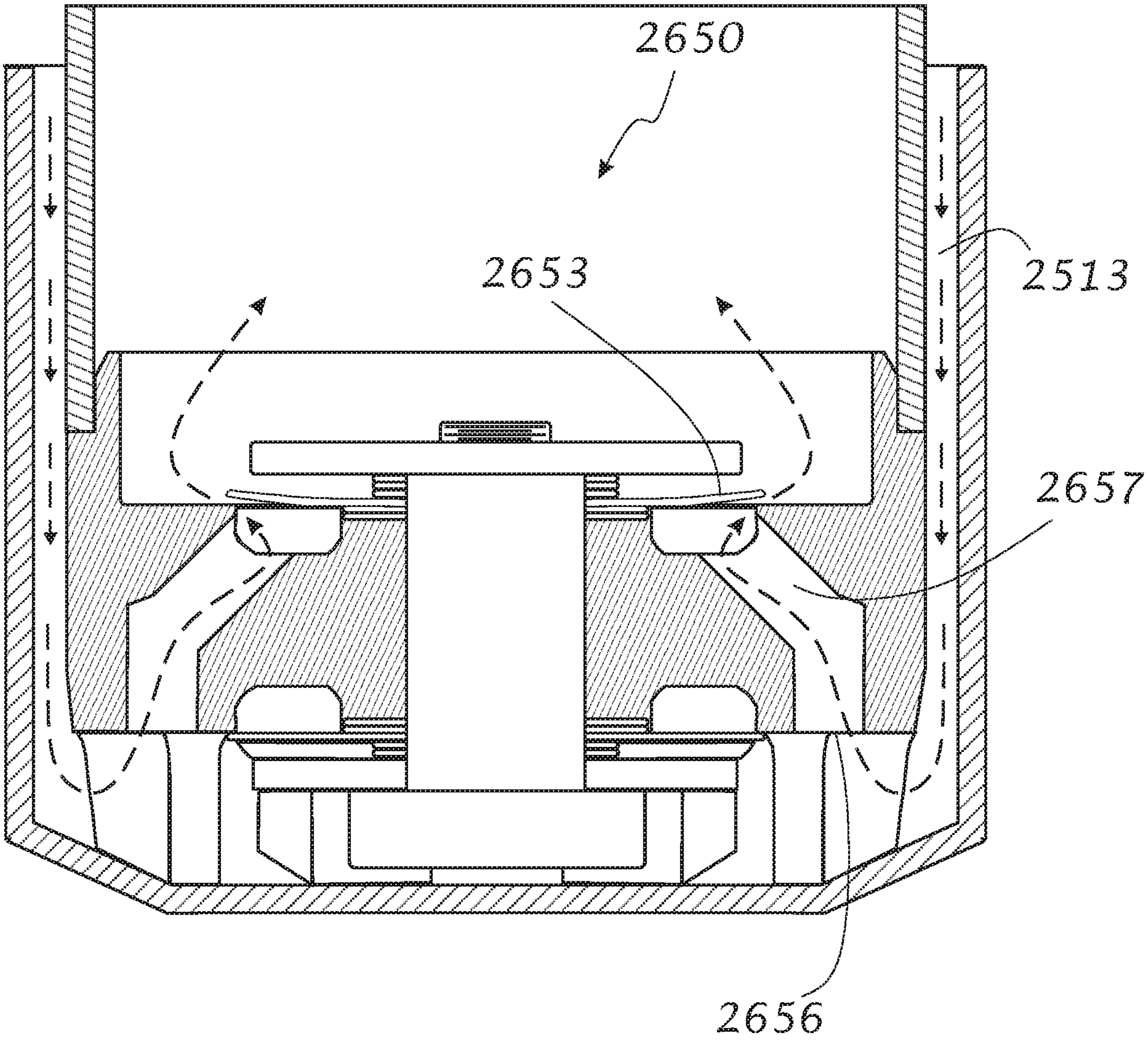

Fig. 38

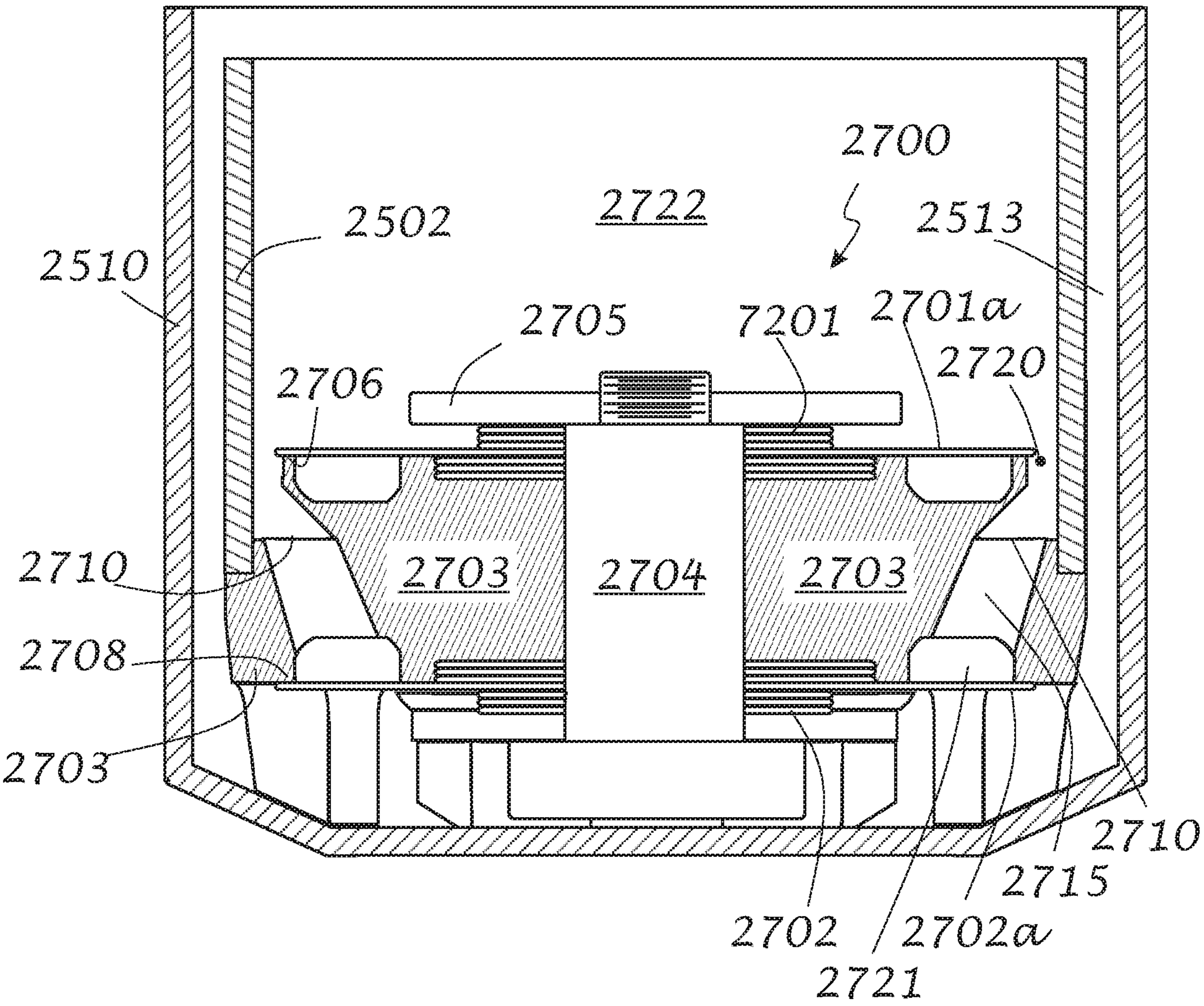
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Fig. 39

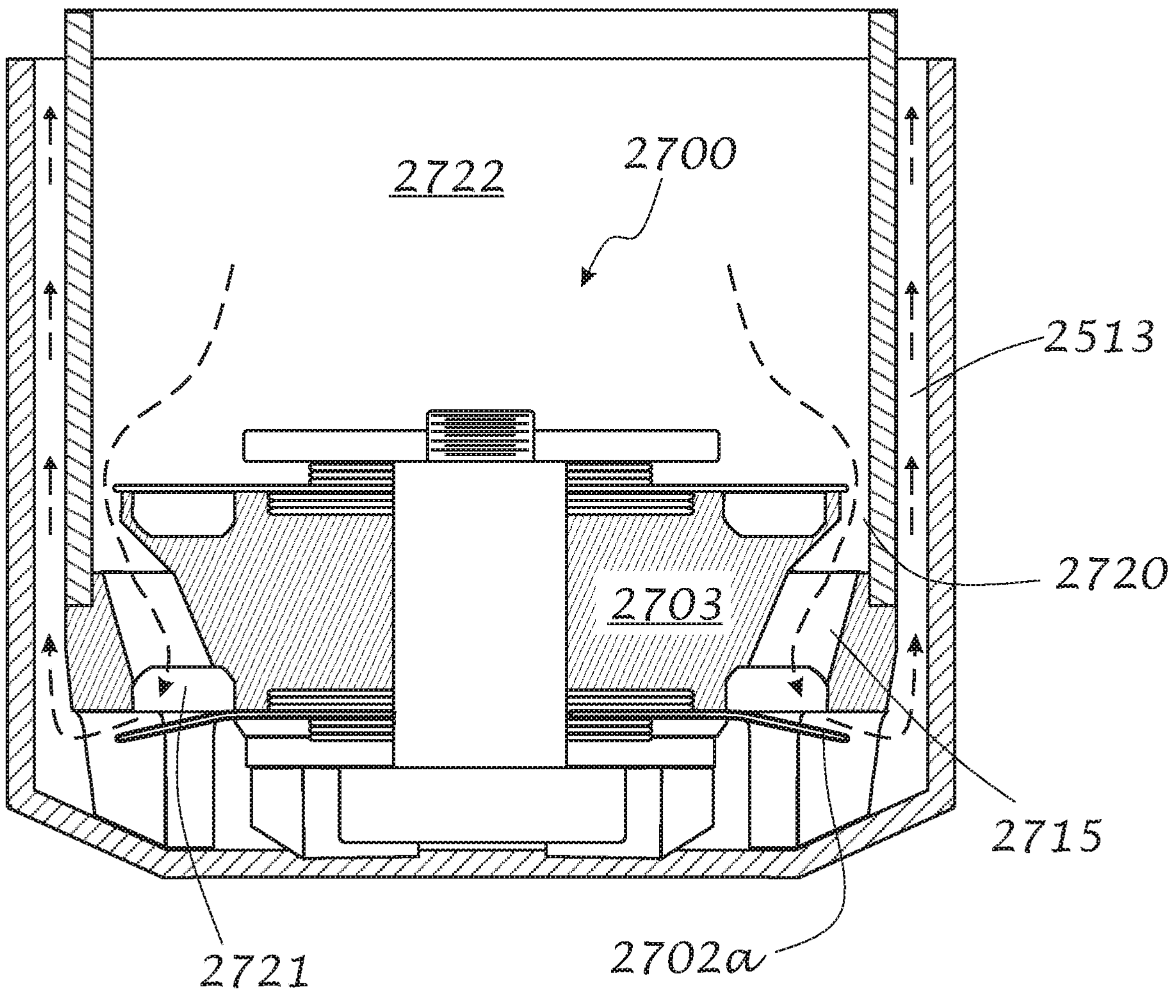

Fig. 40

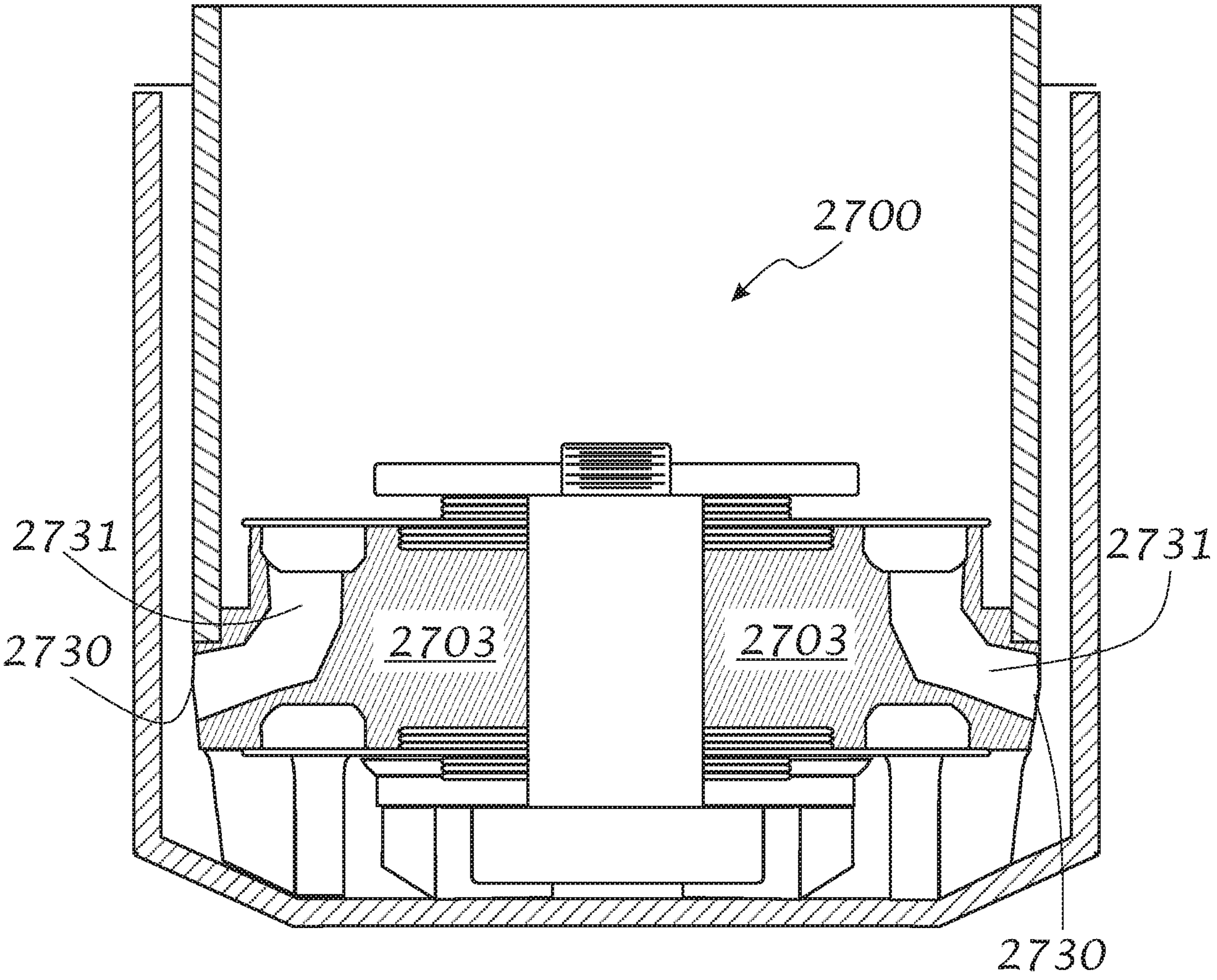

Fig. 41

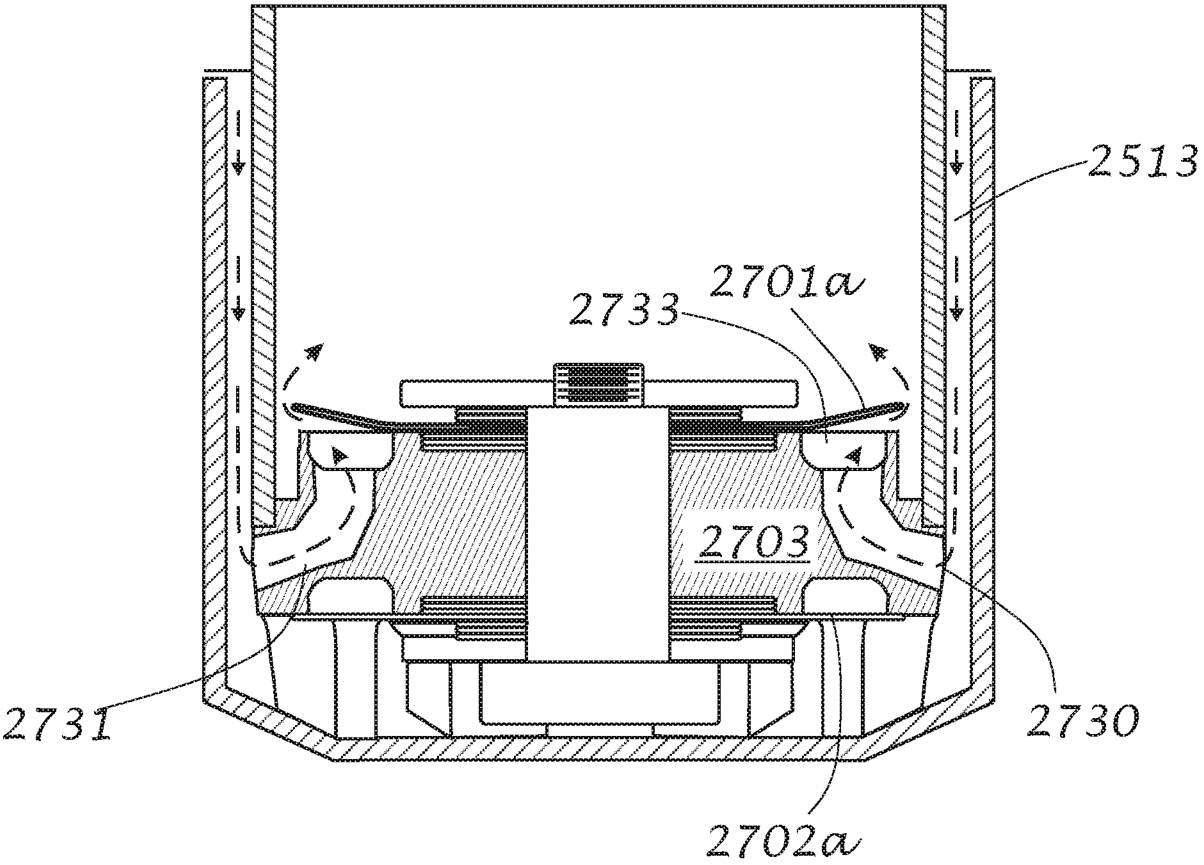

Fig. 42

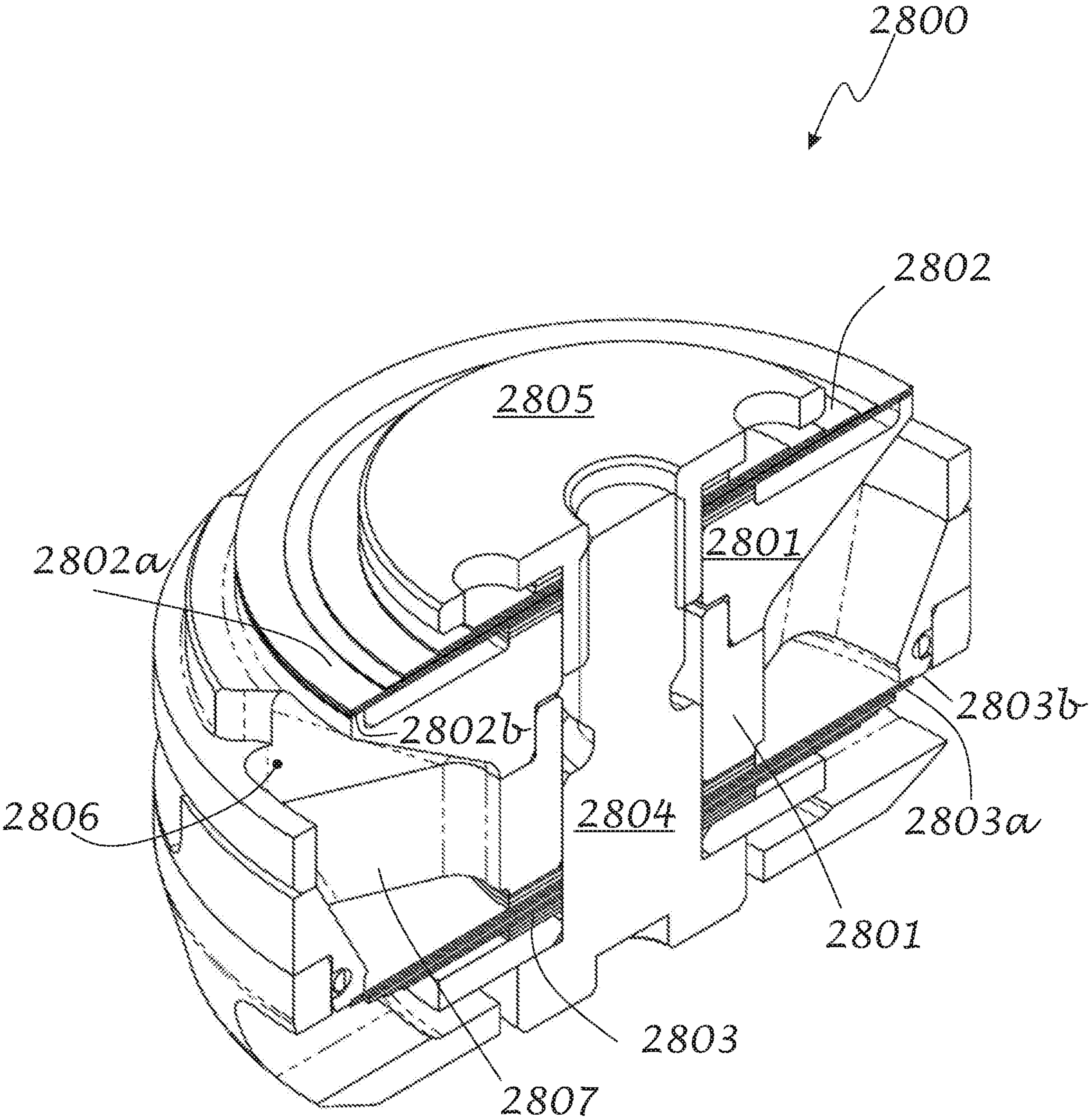
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Fig. 43

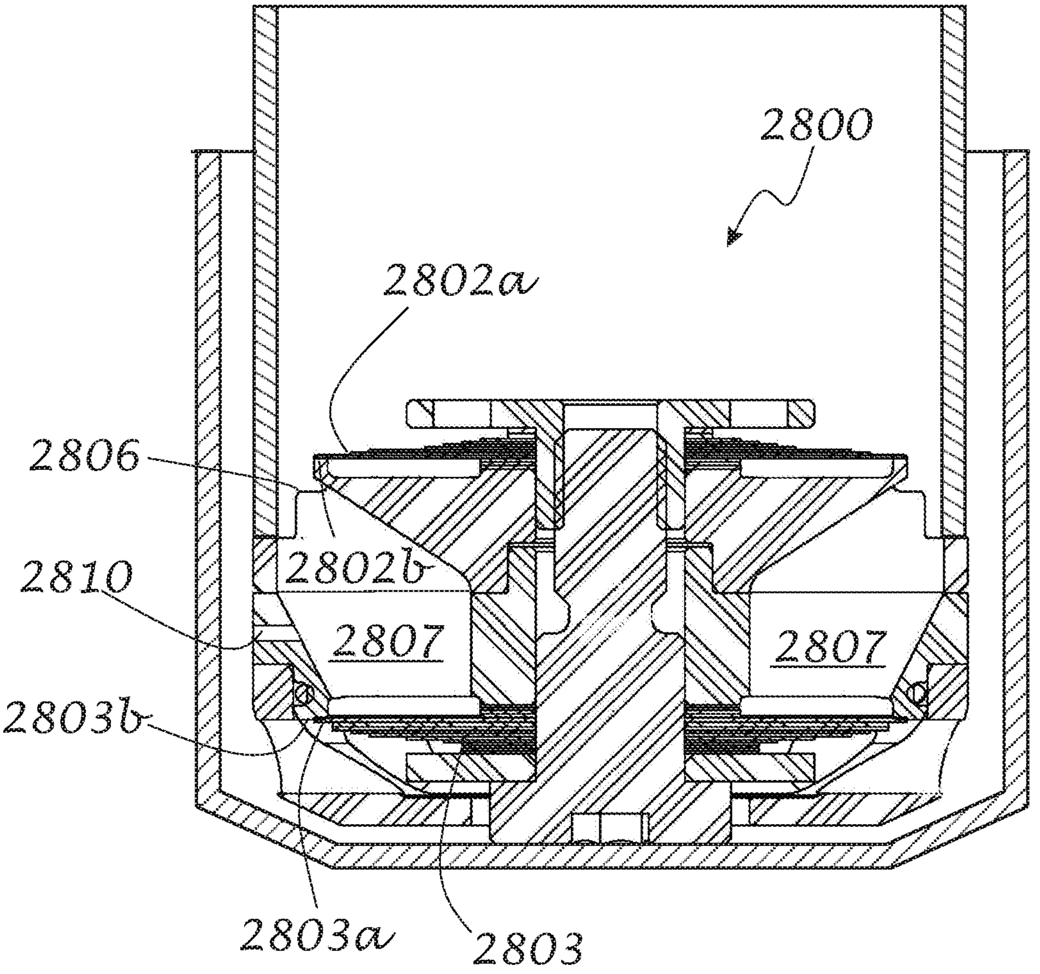

Fig. 44

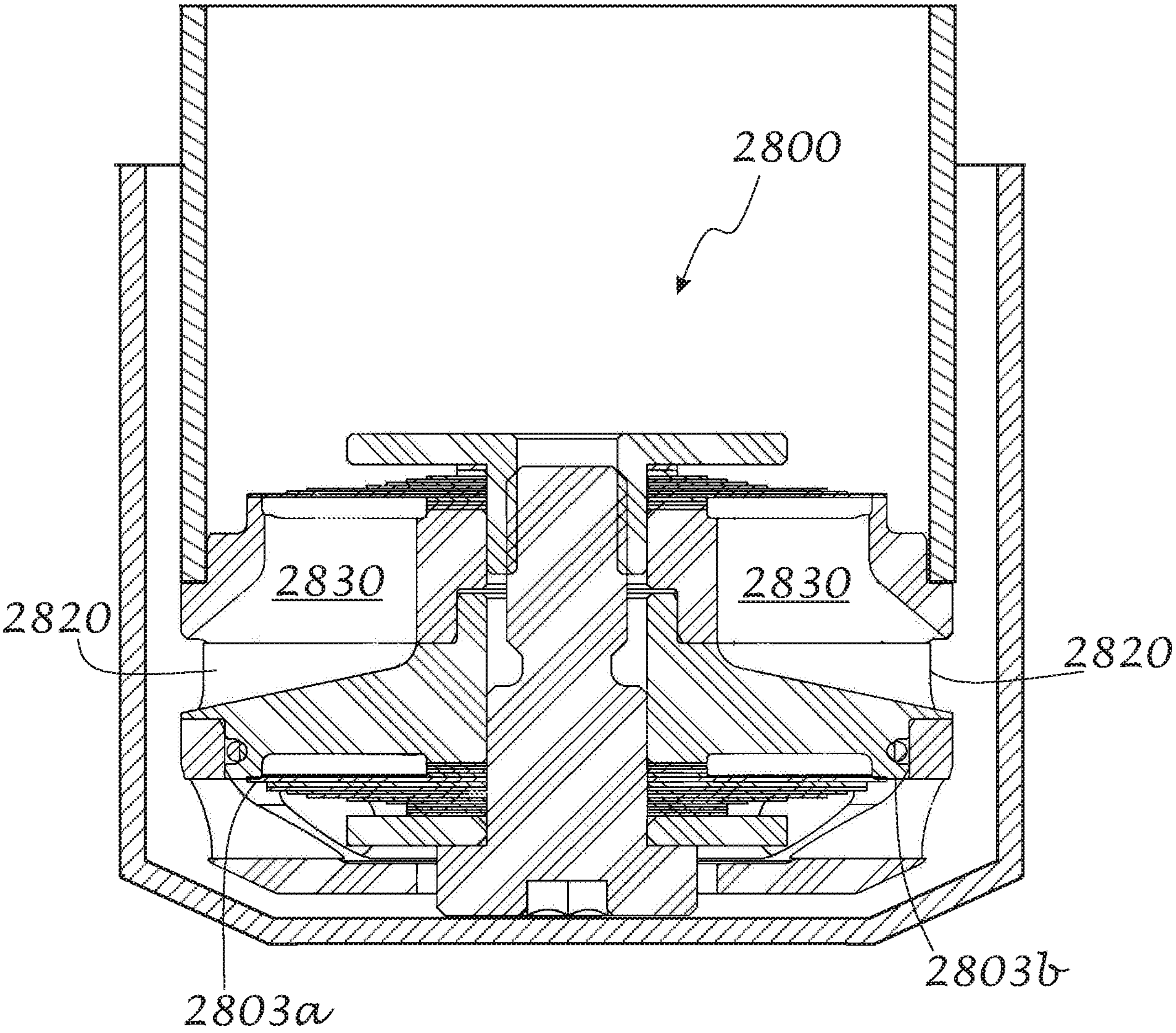

Fig. 45

INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 20/22193

A. CLASSIFICATION OF SUBJECT MATTER

IPC - F16F 9/512 (2020.01)

CPC - F16F 9/5126

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
See Search History document

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
See Search History document

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
See Search History document

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DE 10 2010 023 434 A1 (DAIMLER AG) 15 December 2011 (15.12.2011) entire document	1-14
A	US 2015/0276005 A1 (MANDO CORPORATION) 01 October 2015 (01.10.2015) entire document	1-14
A	US 5,248,014 A (ASHIBA) 28 September 1993 (28.09.1993) entire document	1-14
A	DE 10 2015 104 388 A1 (THYSSENKRUPP AG et al) 29 September 2016 (29.09.2016) entire document	1-14
A	DE 10 2011 102 537 A1 (DAIMLER AG) 16 February 2012 (16.02.2012) entire document	1-14

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:	
"A" document defining the general state of the art which is not considered to be of particular relevance	"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"D" document cited by the applicant in the international application	"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"E" earlier application or patent but published on or after the international filing date	
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"O" document referring to an oral disclosure, use, exhibition or other means	
"P" document published prior to the international filing date but later than the priority date claimed	"&" document member of the same patent family

Date of the actual completion of the international search 05 May 2020	Date of mailing of the international search report 28 JUL 2020
Name and mailing address of the ISA/US Mail Stop PCT, Attn: ISA/US, Commissioner for Patents P.O. Box 1450, Alexandria, Virginia 22313-1450 Facsimile No. 571-273-8300	Authorized officer Lee Young Telephone No. PCT Helpdesk: 571-272-4300

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US 20/22193

Box No. II Observations where certain claims were found unsearchable (Continuation of item 2 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

2. ☐ Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

3. ☐ Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:
SEE EXTRA SHEET

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. ☐ As all searchable claims could be searched without effort justifying additional fees, this Authority did not invite payment of additional fees.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☒ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:
1-14

Remark on Protest

☐ The additional search fees were accompanied by the applicant's protest and, where applicable, the payment of a protest fee.

☐ The additional search fees were accompanied by the applicant's protest but the applicable protest fee was not paid within the time limit specified in the invitation.

☐ No protest accompanied the payment of additional search fees.

-*-Continuation of: Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)-*-

This application contains the following inventions or groups of inventions which are not so linked as to form a single general inventive concept under PCT Rule 13.1. In order for all inventions to be searched, the appropriate additional search fees must be paid.

Group I: Claims 1-14 directed to damping flow control device and a bypass flow control device.

Group II: Claims 15-22 directed to a piston that includes a first inlet port and a first outlet port

Group III: Claims 23-28 directed to a method including the step of operating the hydraulic machine at a rate without opening the pressure balanced blow-off valve.

Group IV: Claims 29-35 directed to a base valve assembly.

The inventions listed as Groups I-IV do not relate to a single general inventive concept under PCT Rule 13.1 because, under PCT Rule 13.2, they lack the same or corresponding special technical features for the following reasons:

SPECIAL TECHNICAL FEATURES

The invention of Group I includes the special technical feature of
a damping flow control device; and
a first bypass flow control device.
not required by the claims of Groups II-IV.

The invention of Group II includes the special technical feature of a
a piston that includes a first inlet port and a first outlet port;
not required by the claims of Groups I and III-IV.

The invention of Group III includes the special technical feature of a
operating the hydraulic machine to increase the pressure in the compression volume to a value greater than the threshold at a second rate lower than the first rate without opening the pressure balanced blow-off valve; and
operating the hydraulic machine to increase the pressure in the compression volume to a value less than the threshold value at a rate greater than the first rate without opening the pressure balanced blow-off valve;
not required by the claims of Groups I-II and IV.

The invention of Group IV includes the special technical feature of
a base valve assembly;
not required by the claims of Groups I-III.

COMMON TECHNICAL FEATURES

Groups I-IV share the common technical features of a piston, a pressure tube with an internal volume divided into a first chamber and a second chamber by the piston and a blow-off valve.
However, this shared technical feature does not represent a contribution over prior art as being anticipated by DE 10 2010 023 434 A1 to DAIMLER AG (hereinafter referred to as DAIMLER), which discloses a piston (7; Fig. 2), a pressure tube (working cylinder 5; Fig. 2) with an internal volume divided into a first chamber (second working space 11; Fig. 2) and a second chamber (first working space 9; Para. [0031]; Fig. 2) by the piston and a blow-off valve (hydraulic transmission element 27; Fig. 2).

Groups I and III share the common technical features of a hydraulic machine. However, this shared technical feature does not represent a contribution over prior art as being anticipated by DAIMLER, which discloses a hydraulic machine (hydraulic motor generator device 25; Para. [0038]; Fig. 2).

As the common technical features were known in the art at the time of the invention, these cannot be considered special technical feature that would otherwise unify the groups.

Therefore, Groups I-IV lack unity under PCT Rule 13 because they do not share a same or corresponding special technical feature.