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- (71) Applicant (for all designated States except US): **OS-
COMP SYSTEMS INC.** [US/US]; 337 Summer Street,
Floor 0, Boston, MA 02210 (US).
- (72) Inventors; and
- (75) Inventors/Applicants (for US only): **SANTOS, Pedro**
[DO/US]; 4 Trowbridge Place, Apt. 1EF, Cambridge, MA

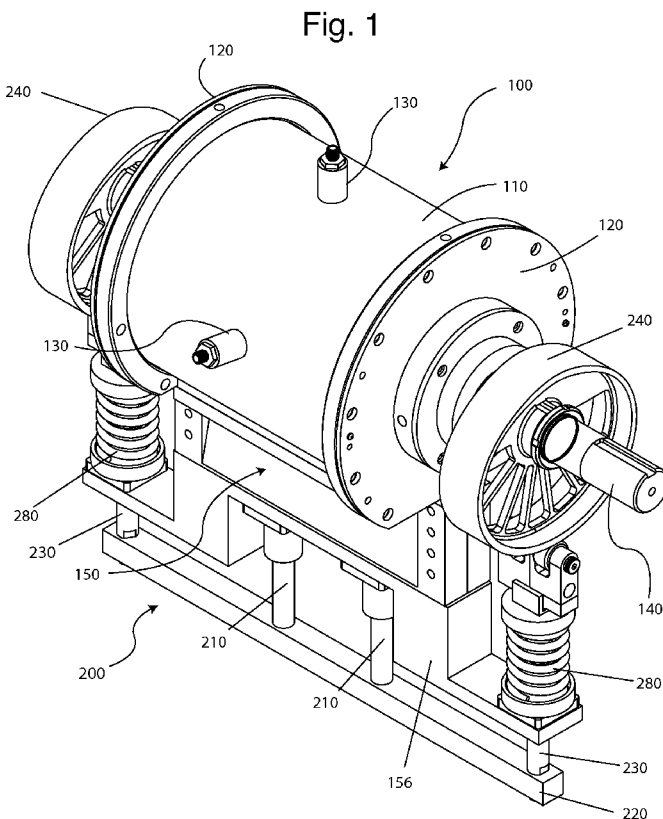
02138 (US). **PITTS, Jeremy** [US/US]; 9 West Broad-
way, Apt. 321, Boston, MA 02127 (US). **NELSON, An-
drew** [US/US]; 18 Day Street, Apt. 410, Somerville, MA
02144 (US). **SANTEN, Johannes** [DE/US]; 93 Mosle
Road, Far Hills, NJ 07931 (US). **WALTON, John**
[US/US]; 532 Beacon Street, Boston, MA 02215 (US).
WESTWOOD, Mitchell [US/US]; 32 Hereford Street,
Boston, MA 02215 (US). **O'HANLEY, Harrison**
[US/US]; 208 Arsilla Road, Ipswich, MA 01938 (US).

(74) Agent: **BLUESTONE, David, H.**; Brinks Hofer Gilson
& Lione, P.O. Box 10087, Chicago, IL 60610 (US).

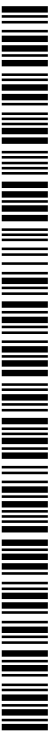
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(54) Title: COMPRESSOR WITH LIQUID INJECTION COOLING



(57) Abstract: A positive displacement compressor designed for near isothermal compression. A rotor includes a curved sealing portion that coincides with a in an interior rotor casing wall. Liquid injectors provide cooling liquid. A gate moves within the compression chamber to either make contact with or be proximate to the rotor as it turns. Gate positioning systems position the gate in this manner, taking into account the shape of the rotor. Outlet valves allow for expulsion of liquids and compressed gas. The unique geometry and relationship between the parts provides for efficiencies and higher pressures not previously found in existing compressor designs.



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COMPRESSOR WITH LIQUID INJECTION COOLING

5 [001] This application claims priority to U.S. provisional application Serial No. 61/378,297, which was filed on August 30, 2010, and U.S. provisional application Serial No. 61/485,006, which was filed on May 11, 2011.

BACKGROUND

1. Technical Field.

10 [002] The invention generally relates to fluid pumps, such as compressors and expanders. More specifically, preferred embodiments utilize a novel rotary compressor design for compressing air, vapor, or gas for high pressure conditions over 200 psi and power ratings above 10 HP.

2. Related Art.

15 [001] Compressors have typically been used for a variety of applications, such as air compression, vapor compression for refrigeration, and compression of industrial gases. Compressors can be split into two main groups, positive displacement and dynamic. Positive displacement compressors reduce the volume of the compression chamber to increase the pressure of the fluid in the chamber. This is done by applying force to a drive shaft that is driving the compression
20 process. Dynamic compressors work by transferring energy from a moving set of blades to the working fluid.

[002] Positive displacement compressors can take a variety of forms. They are typically classified as reciprocating or rotary compressors. Reciprocating compressors are commonly used in industrial applications where higher pressure
25 ratios are necessary. They can easily be combined into multistage machines, although single stage reciprocating compressors are not typically used at pressures above 80 psig. Reciprocating compressors use a piston to compress the vapor, air, or gas, and have a large number of components to help translate the rotation of the drive shaft into the reciprocating motion used for compression. This can lead to
30 increased cost and reduced reliability. Reciprocating compressors also suffer from

high levels of vibration and noise. This technology has been used for many industrial applications such as natural gas compression.

[003] Rotary compressors use a rotating component to perform compression. As noted in the art, rotary compressors typically have the following features in common: (1) they impart energy to the gas being compressed by way of an input shaft moving a single or multiple rotating elements; (2) they perform the compression in an intermittent mode; and (3) they do not use inlet or discharge valves. (Brown, *Compressors: Selection and Sizing*, 3rd Ed., at 6). As further noted in Brown, rotary compressor designs are generally suitable for designs in which less than 20:1 pressure ratios and 1000 CFM flow rates are desired. For pressure ratios above 20:1, Royce suggests that multistage reciprocating compressors should be used instead.

[004] Typical rotary compressor designs include the rolling piston, screw compressor, scroll compressor, lobe, liquid ring, and rotary vane compressors. Each of these traditional compressors has deficiencies for producing high pressure, near isothermal conditions.

[005] The design of a rotating element/rotor/lobe against a radially moving element/piston to progressively reduce the volume of a fluid has been utilized as early as the mid-19th century with the introduction of the "Yule Rotary Steam Engine." Developments have been made to small-sized compressors utilizing this methodology into refrigeration compression applications. However, current Yule-type designs are limited due to problems with mechanical spring durability (returning the piston element) as well as chatter (insufficient acceleration of the piston in order to maintain contact with the rotor).

[006] For commercial applications, such as compressors for refrigerators, small rolling piston or rotary vane designs are typically used. (P N Ananthanarayanan, *Basic Refrigeration and Air Conditioning*, 3rd Ed., at 171-72.) In these designs, a closed oil-lubricating system is typically used.

[007] Rolling piston designs typically allow for a significant amount of leakage between an eccentrically mounted circular rotor, the interior wall of the casing, and/or the vane that contacts the rotor. By spinning the rolling piston faster,

the leakages are deemed acceptable because the desired pressure and flow rate for the application can be easily reached even with these losses. The benefit of a small self-contained compressor is more important than seeking higher pressure ratios.

5 [008] Rotary vane designs typically use a single circular rotor mounted eccentrically in a cylinder slightly larger than the rotor. Multiple vanes are positioned in slots in the rotor and are kept in contact with the cylinder as the rotor turns typically by spring or centrifugal force inside the rotor. The design and operation of these type of compressors may be found in Mark's Standard Handbook for Mechanical Engineers, Eleventh Edition, at 14:33-34.

10 [009] In a sliding-vane compressor design, vanes are mounted inside the rotor to slide against the casing wall. Alternatively, rolling piston designs utilize a vane mounted within the cylinder that slides against the rotor. These designs are limited by the amount of restoring force that can be provided and thus the pressure that can be yielded.

15 [010] Each of these types of prior art compressors has limits on the maximum pressure differential that it can provide. Typical factors include mechanical stresses and temperature rise. One proposed solution is to use multistaging. In multistaging, multiple compression stages are applied sequentially. Intercooling, or cooling between stages, is used to cool the working fluid down to an
20 acceptable level to be input into the next stage of compression. This is typically done by passing the working fluid through a heat exchanger in thermal communication with a cooler fluid. However, intercooling can result in some condensation of liquid and typically requires filtering out of the liquid elements. Multistaging greatly increases the complexity of the overall compression system and
25 adds costs due to the increased number of components required. Additionally, the increased number of components leads to decreased reliability and the overall size and weight of the system are markedly increased.

[011] For industrial applications, single- and double-acting reciprocating compressors and helical-screw type rotary compressors are most commonly used.
30 Single-acting reciprocating compressors are similar to an automotive type piston with compression occurring on the top side of the piston during each revolution of the

crankshaft. These machines can operate with a single-stage discharging between 25 and 125 psig or in two stages, with outputs ranging from 125 to 175 psig or higher. Single-acting reciprocating compressors are rarely seen in sizes above 25 HP. These types of compressors are typically affected by vibration and mechanical stress and require frequent maintenance. They also suffer from low efficiency due to insufficient cooling.

[012] Double-acting reciprocating compressors use both sides of the piston for compression, effectively doubling the machine's capacity for a given cylinder size. They can operate as a single-stage or with multiple stages and are typically sized greater than 10 HP with discharge pressures above 50 psig. Machines of this type with only one or two cylinders require large foundations due to the unbalanced reciprocating forces. Double-acting reciprocating compressors tend to be quite robust and reliable, but are not sufficiently efficient, require frequent valve maintenance, and have extremely high capital costs.

[013] Lubricant-flooded rotary screw compressors operate by forcing fluid between two intermeshing rotors within a housing which has an inlet port at one end and a discharge port at the other. Lubricant is injected into the chamber to lubricate the rotors and bearings, take away the heat of compression, and help to seal the clearances between the two rotors and between the rotors and housing. This style of compressor is reliable with few moving parts. However, it becomes quite inefficient at higher discharge pressures (above approximately 200 psig) due to the intermeshing rotor geometry being forced apart and leakage occurring. In addition, lack of valves and a built-in pressure ratio leads to frequent over or under compression, which translates into significant energy efficiency losses.

[014] Rotary screw compressors are also available without lubricant in the compression chamber, although these types of machines are quite inefficient due to the lack of lubricant helping to seal between the rotors. They are a requirement in some process industries such as food and beverage, semiconductor, and pharmaceuticals, which cannot tolerate any oil in the compressed air used in their processes. Efficiency of dry rotary screw compressors are 15-20% below comparable

injected lubricated rotary screw compressors and are typically used for discharge pressures below 150 psig.

[015] Using cooling in a compressor is understood to improve upon the efficiency of the compression process by extracting heat, allowing most of the energy to be transmitted to the gas and compressing with minimal temperature increase. Liquid injection has previously been utilized in other compression applications for cooling purposes. Further, it has been suggested that smaller droplet sizes of the injected liquid may provide additional benefits.

[016] In U.S. Pat. No. 4,497,185, lubricating oil was intercooled and injected through an atomizing nozzle into the inlet of a rotary screw compressor. In a similar fashion, U.S. Pat. No. 3,795,117 uses refrigerant, though not in an atomized fashion, that is injected early in the compression stages of a rotary screw compressor. Rotary vane compressors have also attempted finely atomized liquid injection, as seen in U.S. Pat. No. 3,820,923.

[017] In each example, cooling of the fluid being compressed was desired. Liquid injection in rotary screw compressors is typically done at the inlet and not within the compression chamber. This provides some cooling benefits, but the liquid is given the entire compression cycle to coalesce and reduce its effective heat transfer coefficient. Additionally, these examples use liquids that have lubrication and sealing as a primary benefit. This affects the choice of liquid used and may adversely affect its heat transfer and absorption characteristics. Further, these styles of compressors have limited pressure capabilities and thus are limited in their potential market applications.

[018] Rotary designs for engines are also known, but suffer from deficiencies that would make them unsuitable for an efficient compressor design. The most well-known example of a rotary engine is the Wankel engine. While this engine has been shown to have benefits over conventional engines and has been commercialized with some success, it still suffers from multiple problems, including low reliability and high levels of hydrocarbon emissions.

[019] Published International Pat. App. No. WO 2010/017199 and U.S. Pat. Pub. No. 2011/0023814 relate to a rotary engine design using a rotor, multiple

gates to create the chambers necessary for a combustion cycle, and an external cam-drive for the gates. The force from the combustion cycle drives the rotor, which imparts force to an external element. Engines are designed for a temperature increase in the chamber and high temperatures associated with the combustion that occurs within an engine. Increased sealing requirements necessary for an effective compressor design are unnecessary and difficult to achieve. Combustion forces the use of positively contacting seals to achieve near perfect sealing, while leaving wide tolerances for metal expansion, taken up by the seals, in an engine. Further, injection of liquids for cooling would be counterproductive and coalescence is not addressed.

[020] Liquid mist injection has been used in compressors, but with limited effectiveness. In U.S. Pat. No. 5,024,588, a liquid injection mist is described, but improved heat transfer is not addressed. In U.S. Pat. Publication. No. U.S. 2011/0023977, liquid is pumped through atomizing nozzles into a reciprocating piston compressor's compression chamber prior to the start of compression. It is specified that liquid will only be injected through atomizing nozzles in low pressure applications. Liquid present in a reciprocating piston compressor's cylinder causes a high risk for catastrophic failure due to hydrolock, a consequence of the incompressibility of liquids when they build up in clearance volumes in a reciprocating piston, or other positive displacement, compressor. To prevent hydrolock situations, reciprocating piston compressors using liquid injection will typically have to operate at very slow speeds, adversely affecting the performance of the compressor.

[021] The prior art lacks compressor designs in which the application of liquid injection for cooling provides desired results for a near-isothermal application. This is in large part due to the lack of a suitable positive displacement compressor design that can both accommodate a significant amount of liquid in the compression chamber and pass that liquid through the compressor outlet without damage.

BRIEF SUMMARY

[022] The presently preferred embodiments are directed to rotary compressor designs. These designs are particularly suited for high pressure applications, typically above 200 psig with compression ratios typically above for
5 existing high-pressure positive displacement compressors.

[023] One illustrative embodiment of the design includes a non-circular-shaped rotor rotating within a cylindrical casing and mounted concentrically on a drive shaft inserted axially through the cylinder. The rotor is symmetrical along the axis traveling from the drive shaft to the casing with cycloid and constant radius
10 portions. The constant radius portion corresponds to the curvature of the cylindrical casing, thus providing a sealing portion. The changing rate of curvature on the other portions provides for a non-sealing portion. In this illustrative embodiment, the rotor is balanced by way of holes and counterweights.

[024] A gate structured similar to a reciprocating rectangular piston is
15 inserted into and withdrawn from the bottom of the cylinder in a timed manner such that the tip of the piston remains in contact with or sufficiently proximate to the surface of the rotor as it turns. The coordinated movement of the gate and the rotor separates the compression chamber into a low pressure and high pressure region.

[025] As the rotor rotates inside the cylinder, the compression volume is
20 progressively reduced and compression of the fluid occurs. At the same time, the intake side is filled with gas through the inlet. An inlet and exhaust are located to allow fluid to enter and exit the chamber at appropriate times. During the compression process, atomized liquid is injected into the compression chamber in such a way that a high and rapid rate of heat transfer is achieved between the gas
25 being compressed and the injected cooling liquid. This results in near isothermal compression, which enables a much higher efficiency compression process.

[026] The rotary compressor embodiments sufficient to achieve near isothermal compression are capable of achieving high pressure compression at higher efficiencies. It is capable of compressing gas only, a mixture of gas and

liquids, or for pumping liquids. As one of ordinary skill in the art would appreciate, the design can also be used as an expander.

[027] The particular rotor and gate designs may also be modified depending on application parameters. For example, different cycloidal and constant radii may be employed. Alternatively, double harmonic or other functions may be used for the variable radius. The gate may be of one or multiple pieces. It may implement a contacting tip-seal, liquid channel, or provide a non-contacting seal by which the gate is proximate to the rotor as it turns.

[028] Several embodiments provide mechanisms for driving the gate external to the main casing. In one embodiment, a spring-backed cam drive system is used. In others, a belt-based system with or without springs may be used. In yet another, a dual cam follower gate positioning system is used. Further, an offset gate guide system may be used. Further still, linear actuator, magnetic drive, and scotch yoke systems may be used.

[029] The presently preferred embodiments provide advantages not found in the prior art. The design is tolerant of liquid in the system, both coming through the inlet and injected for cooling purposes. High compression ratios are achievable due to effective cooling techniques. Lower vibration levels and noise are generated. Valves are used to minimize inefficiencies resulting from over- and under-compression common in existing rotary compressors. Seals are used to allow higher pressures and slower speeds than typical with other rotary compressors. The rotor design allows for balanced, concentric motion, reduced acceleration of the gate, and effective sealing between high pressure and low pressure regions of the compression chamber.

BRIEF DESCRIPTION OF THE DRAWINGS

[030] The invention can be better understood with reference to the following drawings and description. The components in the figures are not necessarily to scale, emphasis instead being placed upon illustrating the principles of the invention. Moreover, in the figures, like referenced numerals designate corresponding parts throughout the different views.

[031] Figure 1 is a perspective view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

[032] Figure 2 is a right-side view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

5 [033] Figure 3 is a left-side view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

[034] Figure 4 is a front view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

10 [035] Figure 5 is a back view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

[036] Figure 6 is a top view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

[037] Figure 7 is a bottom view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

15 [038] Figure 8 is a cross-sectional view of a rotary compressor with a spring-backed cam drive in accordance with an embodiment of the present invention.

[039] Figure 9 is a perspective view of rotary compressor with a belt-driven, spring-biased gate positioning system in accordance with an embodiment of the present invention.

20 [040] Figure 10 is a perspective view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

25 [041] Figure 11 is a right-side view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

[042] Figure 12 is a left-side view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

30 [043] Figure 13 is a front view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

[044] Figure 14 is a back view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

5 [045] Figure 15 is a top view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

[046] Figure 16 is a bottom view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

10 [047] Figure 17 is a cross-sectional view of a rotary compressor with a dual cam follower gate positioning system in accordance with an embodiment of the present invention.

[048] Figure 18 is perspective view of a rotary compressor with a belt-driven gate positioning system in accordance with an embodiment of the present invention.

15 [049] Figure 19 is perspective view of a rotary compressor with an offset gate guide positioning system in accordance with an embodiment of the present invention.

[050] Figure 20 is a right-side view of a rotary compressor with an offset gate guide positioning system in accordance with an embodiment of the present invention.

20 [051] Figure 21 is a front view of a rotary compressor with an offset gate guide positioning system in accordance with an embodiment of the present invention.

25 [052] Figure 22 is a cross-sectional view of a rotary compressor with an offset gate guide positioning system in accordance with an embodiment of the present invention.

[053] Figure 23 is perspective view of a rotary compressor with a linear actuator gate positioning system in accordance with an embodiment of the present invention.

[054] Figures 24A and B are right side and cross-section views, respectively, of a rotary compressor with a magnetic drive gate positioning system in accordance with an embodiment of the present invention

5 [055] Figure 25 is perspective view of a rotary compressor with a scotch yoke gate positioning system in accordance with an embodiment of the present invention.

[056] Figures 26A-F are cross-sectional views of the inside of an embodiment of a rotary compressor with a contacting tip seal in a compression cycle in accordance with an embodiment of the present invention.

10 [057] Figures 27A-F are cross-sectional views of the inside of an embodiment of a rotary compressor without a contacting tip seal in a compression cycle in accordance with another embodiment of the present invention.

[058] Figure 28 is perspective, cross-sectional view of a rotary compressor in accordance with an embodiment of the present invention.

15 [059] Figure 29 is a left-side view of an additional liquid injectors embodiment of the present invention.

[060] Figure 30 is a cross-section view of a rotor design in accordance with an embodiment of the present invention.

20 [061] Figures 31A-D are cross-sectional views of rotor designs in accordance with various embodiments of the present invention.

[062] Figures 32A and B are perspective and right-side views of a drive shaft, rotor, and gate in accordance with an embodiment of the present invention.

[063] Figure 33 is a perspective view of a gate with exhaust ports in accordance with an embodiment of the present invention.

25 [064] Figure 34A and B are a perspective view and magnified view of a gate with notches, respectively, in accordance with an embodiment of the present invention.

[065] Figure 35 is a cross-sectional, perspective view a gate with a rolling tip in accordance with an embodiment of the present invention.

30 [066] Figure 36 is a cross-sectional front view of a gate with a liquid injection channel in accordance with an embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[067] To the extent that the following terms are utilized herein, the following definitions are applicable:

[068] **Balanced rotation:** the center of mass of the rotating mass is located on the axis of rotation.

[069] **Chamber volume:** any volume that can contain fluids for compression.

[070] **Compressor:** a device used to increase the pressure of a compressible fluid. The fluid can be either gas or vapor, and can have a wide molecular weight range.

[071] **Concentric:** the center or axis of one object coincides with the center or axis of a second object

[072] **Concentric rotation:** rotation in which one object's center of rotation is located on the same axis as the second object's center of rotation.

[073] **Positive displacement compressor:** a compressor that collects a fixed volume of gas within a chamber and compresses it by reducing the chamber volume.

[074] **Proximate:** sufficiently close to restrict fluid flow between high pressure and low pressure regions. Restriction does not need to be absolute; some leakage is acceptable.

[075] **Rotor:** A rotating element driven by a mechanical force to rotate about an axis. As used in a compressor design, the rotor imparts energy to a fluid.

[076] **Rotary compressor:** A positive-displacement compressor that imparts energy to the gas being compressed by way of an input shaft moving a single or multiple rotating elements

[077] Figures 1 through 7 show external views of an embodiment of the present invention in which a rotary compressor includes spring backed cam drive gate positioning system. Main housing 100 includes a main casing 110 and end plates 120, each of which includes a hole through which drive shaft 140 passes axially. Liquid injector assemblies 130 are located on holes in the main casing 110.

The main casing includes a hole for the inlet flange 160, and a hole for the gate casing 150.

[078] Gate casing 150 is connected to and positioned below main casing 110 at a hole in main casing 110. The gate casing 150 is comprised of two portions: an inlet side 152 and an outlet side 154. As shown in Figure 28, the outlet side 154 includes outlet ports 435, which are holes which lead to outlet valves 440.

Alternatively, an outlet valve assembly may be used.

[079] Referring back to Figures 1-7, the spring-backed cam drive gate positioning system 200 is attached to the gate casing 150 and drive shaft 140. The gate positioning system 200 moves gate 600 in conjunction with the rotation of rotor 500. A movable assembly includes gate struts 210 and cam struts 230 connected to gate support arm 220 and bearing support plate 156. The bearing support plate 156 seals the gate casing 150 by interfacing with the inlet and outlet sides through a bolted gasket connection. Bearing support plate 156 is shaped to seal gate casing 150, mount bearing housings 270 in a sufficiently parallel manner, and constrain compressive springs 280. Bearing housings 270, also known as pillow blocks, are concentric to the gate struts 210 and the cam struts 230.

[080] Two cam followers 250 are located tangentially to each cam 240, providing a downward force on the gate. Drive shaft 140 turns cams 240, which transmits force to the cam followers 250. The cam followers 250 may be mounted on a through shaft, which is supported on both ends, or cantilevered and only supported on one end. The cam followers 250 are attached to cam follower supports 260, which transfer the force into the cam struts 230. As cams 240 turn, the cam followers 250 are pushed down, thus moving the cam struts 230 down. This moves the gate support arm 220 and the gate strut 210 down. This, in turn, moves the gate 600 down.

[081] Springs 280 provide a restorative upward force to keep the gate 600 timed appropriately to seal against the rotor 500. As the cams 240 continue to turn and no longer effectuate a downward force on the cam followers 250, springs 280 provide an upward force. As shown in this embodiment, compression springs are utilized. As one of ordinary skill in the art would appreciate, tension springs and the

shape of the bearing support plate 156 may be altered to provide for the desired upward or downward force. The upward force of the springs 280 pushes the cam follower support 260 and thus the gate support arm 220 up which in turn moves the gate 600 up.

5 [082] Due to the varying pressure angle between the cam followers 250 and cams 240, the preferred embodiment may utilize an exterior cam profile that differs from the rotor 500 profile. This variation in profile allows for compensation for the changing pressure angle to ensure that the tip of the gate 600 remains proximate to the rotor 500 throughout the entire compression cycle.

10 [083] Line A in Figures 3, 6, and 7 shows the location for the cross-sectional view of the compressor in Figure 8. As shown in Figure 8, the main casing 110 has a cylindrical shape. Liquid injector housings 132 are attached to, or may be cast as a part of, the main casing 110 to provide for openings in the rotor casing 400. Because it is cylindrically shaped in this embodiment, the rotor casing 400 may also be referenced as the cylinder. The interior wall defines a rotor casing volume 410.
15 The rotor 500 concentrically rotates with drive shaft 140 and is affixed to the drive shaft 140 by way of key 540 and press fit.

[084] Figure 9 shows an embodiment of the present invention in which a timing belt with spring gate positioning system is utilized. This embodiment 290
20 incorporates two timing belts 292 each of which is attached to the drive shaft 140 by way of sheaves 294. The timing belts 292 are attached to secondary shafts 142 by way of sheaves 295. Gate strut springs 296 are mounted around gate struts. Rocker arms 297 are mounted to rocker arm supports 299. The sheaves 295 are connected to rocker arm cams 293 to push the rocker arms 297 down. As the inner rings push
25 down on one side of the rocker arms 297, the other side pushes up against the gate support bar 298. The gate support bar 298 pushes up against the gate struts and gate strut springs 296. This moves the gate up. The springs 296 provide a downward force pushing the gate down.

[085] Figures 10 through 17 show external views of a rotary compressor
30 embodiment utilizing a dual cam follower gate positioning system. The main housing 100 includes a main casing 110 and end plates 120, each of which includes

a hole through which a drive shaft 140 passes axially. Liquid injector assemblies 130 are located on holes in the main casing 110. The main casing 110 also includes a hole for the inlet flange 160 and a hole for the gate casing 150. The gate casing 150 is mounted to and positioned below the main casing 110 as discussed above.

5 [086] A dual cam follower gate positioning system 300 is attached to the gate casing 150 and drive shaft 140. The dual cam follower gate positioning system 300 moves the gate 600 in conjunction with the rotation of the rotor 500. In a preferred embodiment, the size and shape of the cams is nearly identical to the rotor in cross-sectional size and shape. In other embodiments, the rotor, cam
10 shape, curvature, cam thickness, and variations in the thickness of the lip of the cam may be adjusted to account for variations in the attack angle of the cam follower. Further, large or smaller cam sizes may be used. For example, a similar shape but smaller size cam may be used to reduce roller speeds.

[087] A movable assembly includes gate struts 210 and cam struts 230
15 connected to gate support arm 220 and bearing support plate 156. In this embodiment, the bearing support plate 157 is straight. As one of ordinary skill in the art would appreciate, the bearing support plate can utilize different geometries, including structures designed to or not to perform sealing of the gate casing 150. In this embodiment, the bearing support plate 157 serves to seal the bottom of the gate
20 casing 150 through a bolted gasket connection. Bearing housings 270, also known as pillow blocks, are mounted to bearing support plate 157 and are concentric to the gate struts 210 and the cam struts 230.

[088] Drive shaft 140 turns cams 240, which transmit force to the cam
25 followers 250, including upper cam followers 252 and lower cam followers 254. The cam followers 250 may be mounted on a through shaft, which is supported on both ends, or cantilevered and only supported on one end. In this embodiment, four cam followers 250 are used for each cam 240. Two lower cam followers 252 are located below and follow the outside edge of the cam 240. They are mounted using a through shaft. Two upper cam followers 254 are located above the previous
30 two and follow the inside edge of the cams 240. They are mounted using a cantilevered connection.

[089] The cam followers 250 are attached to cam follower supports 260, which transfer the force into the cam struts 230. As the cams 240 turn, the cam struts 230 move up and down. This moves the gate support arm 220 and gate struts 210 up and down, which in turn, moves the gate 600 up and down.

5 [090] Line A in Figures 11, 12, 15, and 16 show the location for the cross-sectional view of the compressor in Figure 17. As shown in Figure 17, the main casing 110 has a cylindrical shape. Liquid injector housings 132 are attached to or may be cast as a part of the main casing 110 to provide for openings in the rotor casing 400. The rotor 500 concentrically rotates around drive shaft 140.

10 [091] An embodiment using a belt driven system 310 is shown in Figure 18. Timing belts 292 are connected to the drive shaft 140 by way of sheaves 294. The timing belts 292 are each also connected to secondary shafts 142 by way of another set of sheaves 295. The secondary shafts 142 drive the external cams 240, which are placed below the gate casing 150 in this embodiment. Sets of upper and lower cam
15 followers 254 and 252 are applied to the cams 240, which provide force to the movable assembly including gate struts 210 and gate support arm 220. As one of ordinary skill in the art would appreciate, belts may be replaced by chains or other materials.

[092] An embodiment of the present invention using an offset gate guide
20 system is shown in Figures 19 through 22 and 33. Outlet of the compressed gas and injected fluid is achieved through a ported gate system 602 comprised of two parts bolted together to allow for internal lightening features. Fluid passes through channels 630 in the upper portion of the gate 602 and travels to the lengthwise sides to outlet through an exhaust port 344 in a timed manner with relation to the angle of
25 rotation of the rotor 500 during the cycle. Discrete point spring-backed scraper seals 326 provide sealing of the gate 602 in the single piece gate casing 336. Liquid injection is achieved through a variety of flat spray nozzles 322 and injector nozzles 130 across a variety of liquid injector port 324 locations and angles.

[093] Reciprocating motion of the two-piece gate 602 is controlled through
30 the use of an offset spring-backed cam follower control system 320 to achieve gate motion in concert with rotor rotation. Single cams 342 drive the gate system

downwards through the transmission of force on the cam followers 250 through the cam struts 338. This results in controlled motion of the crossarm 334, which is connected by bolts (some of which are labeled as 328) with the two-piece gate 602. The crossarm 334 mounted linear bushings 330, which reciprocate along the length
5 of cam shafts 332, control the motion of the gate 602 and the crossarm 334. The cam shafts 332 are fixed in a precise manner to the main casing through the use of cam shaft support blocks 340. Compression springs 346 are utilized to provide a returning force on the crossarm 334, allowing the cam followers 250 to maintain constant rolling contact with the cams, thereby achieving controlled reciprocating
10 motion of the two-piece gate 602.

[094] Figure 23 shows an embodiment using a linear actuator system 350 for gate positioning. A pair of linear actuators 352 is used to drive the gate. In this embodiment, it is not necessary to mechanically link the drive shaft to the gate as with other embodiments. The linear actuators 352 are controlled so as to raise and
15 lower the gate in accordance with the rotation of the rotor. The actuators may be electronic, hydraulic, belt-driven, electromagnetic, gas-driven, variable-friction, or other means. The actuators may be computer controlled or controlled by other means.

[095] Figures 24A and B show a magnetic drive system 360. The gate
20 system may be driven, or controlled, in a reciprocating motion through the placement of magnetic field generators, whether they are permanent magnets or electromagnets, on any combination of the rotor 500, gate 600, and/or gate casing 150. The purpose of this system is to maintain a constant distance from the tip of the gate 600 to the surface of the rotor 500 at all angles throughout the cycle. In a
25 preferred magnetic system embodiment, permanent magnets 366 are mounted into the ends of the rotor 500 and retained. In addition, permanent magnets 364 are installed and retained in the gate 600. Poles of the magnets are aligned so that the magnetic force generated between the rotor's magnets 366 and the gate's magnets 364 is a repulsive force, forcing the gate 600 down throughout the cycle to control its
30 motion and maintain constant distance. To provide an upward, returning force on the gate 600, additional magnets (not shown) are installed into the bottom of the

gate 600 and the bottom of the gate casing 150 to provide an additional repulsive force. The magnetic drive systems are balanced to precisely control the gate's reciprocating motion.

[096] Alternative embodiments may use an alternate pole orientation to provide attractive forces between the gate and rotor on the top portion of the gate and attractive forces between the gate and gate casing on the bottom portion of the gate. In place of the lower magnet system, springs may be used to provide a repulsive force. In each embodiment, electromagnets may be used in place of permanent magnets. In addition, switched reluctance electromagnets may also be utilized. In another embodiment, electromagnets may be used only in the rotor and gate. Their poles may switch at each inflection point of the gate's travel during its reciprocating cycle, allowing them to be used in an attractive and repulsive method.

[097] Alternatively, direct hydraulic or indirect hydraulic (hydropneumatic) can be used to apply motive force/energy to the gate to drive it and position it adequately. Solenoid or other flow control valves can be used to feed and regulate the position and movement of the hydraulic or hydropneumatic elements. Hydraulic force may be converted to mechanical force acting on the gate through the use of a cylinder based or direct hydraulic actuators using membranes/diaphragms.

[098] Figure 25 shows an embodiment using a scotch yoke gate positioning system 370. Here, a pair of scotch yokes 372 is connected to the drive shaft and the bearing support plate. A roller rotates at a fixed radius with respect to the shaft. The roller follows a slot within the yoke 372, which is constrained to a reciprocating motion. The yoke geometry can be manipulated to a specific shape that will result in desired gate dynamics.

[099] As one of skill in the art would appreciate, these alternative drive mechanisms do not require any particular number of linkages between the drive shaft and the gate. For example, a single spring, belt, linkage bar, or yoke could be used. Depending on the design implementation, more than two such elements could be used.

[0100] Figures 26A-26F show a compression cycle of an embodiment utilizing a tip seal 620. As the drive shaft 140 turns, the rotor 500 and gate strut 210 push up gate 600 so that it is timed with the rotor 500. As the rotor 500 turns clockwise, the gate 600 rises up until the rotor 500 is in the 12 o'clock position shown in Figure 26C. As the rotor 500 continues to turn, the gate 600 moves downward until it is back at the 6 o'clock position in Figure 26F. The gate 600 separates the portion of the cylinder that is not taken up by rotor 500 into two components: an intake component 412 and a compression component 414.

[0101] Figures 26A-F depict steady state operation. Accordingly, in Figure 26A, where the rotor 500 is in the 6 o'clock position, the compression volume 414, which constitutes a subset of the rotor casing volume 410, already has received fluid. In Figure 26B, the rotor 500 has turned clockwise and gate 600 has risen so that the tip seal 620 makes contact with the rotor 500 to separate the intake volume 412, which also constitutes a subset of the rotor casing volume 410, from the compression volume 414. Embodiments using the roller tip 650 discussed below instead of tip seal 620 would operate similarly. As the rotor 500 turns, as shown further in Figures 26C-E, the intake volume 412 increases, thereby drawing in more fluid from inlet 420, while the compression volume 414 decreases. As the volume of the compression volume 414 decreases, the pressure increases. The pressurized fluid is then expelled by way of an outlet 430. At a point in the compression cycle when a desired high pressure is reached, the outlet valve opens and the high pressure fluid can leave the compression volume 414. In this embodiment, the valve outputs both the compressed gas and the liquid injected into the compression chamber.

[0102] Figures 27A - 27F show an embodiment in which the gate 600 does not use a tip seal. Instead, the gate 600 is timed to be proximate to the rotor 500 as it turns. The close proximity of the gate 600 to the rotor 500 leaves only a very small path for high pressure fluid to escape. Close proximity in conjunction with the presence of liquid (due to the liquid injectors 136 or an injector placed in the gate itself) allow the gate 600 to effectively create an intake fluid component 412 and

a compression component 414. Embodiments incorporating notches 640 would operate similarly.

[0103] Figure 28 shows a cross-sectional perspective view of the rotor casing 400, the rotor 500, and the gate 600. The inlet port 420 shows the path that gas can enter. The outlet 430 is comprised of several holes that serve as outlet ports 435 that lead to outlet valves 440. The gate casing 150 consists of an inlet side 152 and an outlet side 154. A return pressure path (not shown) may be connected to the inlet side 152 of the gate casing 150 and the inlet port 420 to ensure that there is no back pressure build up against gate 600 due to leakage through the gate seals. As one of ordinary skill in the art would appreciate, it is desirable to achieve a hermetic seal, although perfect hermetic sealing is not necessary.

[0104] Figure 29 shows an alternative embodiment in which flat spray liquid injector housings 170 are located on the main casing 110 at approximately the 3 o'clock position. These injectors can be used to inject liquid directly onto the inlet side of the gate 600, ensuring that it does not reach high temperatures. These injectors also help to provide a coating of liquid on the rotor 500, helping to seal the compressor.

[0105] As discussed above, the preferred embodiments utilize a rotor that concentrically rotates within a rotor casing. In the preferred embodiment, the rotor 500 is a right cylinder with a non-circular cross-section that runs the length of the main casing 110. Figure 30 shows a cross-sectional view of the sealing and non-sealing portions of the rotor 500. The profile of the rotor 500 is comprised of three sections. The radii in sections I and III are defined by a cycloidal curve. This curve also represents the rise and fall of the gate and defines an optimum acceleration profile for the gate. Other embodiments may use different curve functions to define the radius such as a double harmonic function. Section II employs a constant radius 570, which corresponds to the maximum radius of the rotor. The minimum radius 580 is located at the intersection of sections I and III, at the bottom of rotor 500. In a preferred embodiment, Φ is 23.8 degrees. In alternative embodiments, other angles may be utilized depending on the desired size of the compressor, the desired acceleration of the gate, and desired sealing area.

[0106] The radii of the rotor 500 in the preferred embodiment can be calculated using the following functions:

$$r(t) = \begin{cases} r_I = r_{\min} + h \left[\frac{t_I}{T} + \sin\left(\frac{2\pi t_I}{T}\right) \right] \\ r_{II} = r_{\max} \\ r_{III} = r_{\min} + h \left[\frac{t_{III}}{T} + \sin\left(\frac{2\pi t_{III}}{T}\right) \right] \end{cases} \quad \#$$

5

[0107] In a preferred embodiment, the rotor 500 is symmetrical along one axis. It may generally resemble a cross-sectional egg shape. The rotor 500 includes a hole 530 in which the drive shaft 140 and a key 540 may be mounted. The rotor 500 has a sealing section 510, which is the outer surface of the rotor 500 corresponding to section II, and a non-sealing section 520, which is the outer surface of the rotor 500 corresponding to sections I and III. The sections I and III have a smaller radius than sections II creating a compression volume.

10

[0108] The sealing portion 510 is shaped to correspond to the curvature of the rotor casing 400, thereby creating a dwell seal that effectively minimizes communication between the outlet 430 and inlet 420. Physical contact is not required for the dwell seal. Instead, it is sufficient to create a tortuous path that minimizes the amount of fluid that can pass through. In a preferred embodiment, the gap between the rotor and the casing in this embodiment is less than 0.008 inches. As one of ordinary skill in the art would appreciate, this gap may be altered depending on tolerances, both in machining the rotor 500 and rotor housing 400, temperature, material properties, and other specific application requirements.

15

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[0109] Additionally, as discussed below, liquid is injected into the compression chamber. By becoming entrained in the gap between the sealing portion 510 and the rotor casing 400, the liquid can increase the effectiveness of the dwell seal.

25

[0110] As shown in Figure 31A, the rotor 500 is balanced with cut out shapes and counterweights. Holes, some of which are marked as 550, lighten the rotor 500. Counterweights, one of which is labeled as 560, are made of a denser

material than the remainder of the rotor 500. The shapes of the counterweights can vary and do not need to be cylindrical.

[0111] The rotor design provides several advantages. As shown in the embodiment of Figure 31A, the rotor 500 includes 7 cutout holes 550 on one side and two counterweights 560 on the other side to allow the center of mass to match the center of rotation. An opening 530 includes space for the drive shaft and a key. This weight distribution is designed to achieve balanced, concentric motion. The number and location of cutouts and counterweights may be changed depending on structural integrity, weight distribution, and balanced rotation parameters.

[0112] The cross-sectional shape of the rotor 500 allows for concentric rotation about the drive shaft's axis of rotation, a dwell seal 510 portion, and open space on the non-sealing side for increased gas volume for compression. Concentric rotation provides for rotation about the drive shaft's principal axis of rotation and thus smoother motion and reduced noise.

[0113] An alternative rotor design 502 is shown in Figure 31B. In this embodiment, a different arc of curvature is implemented utilizing three holes 550 and a circular opening 530. Another alternative design 504 is shown in Figure 31C. Here, a solid rotor shape is used and a larger hole 530 (for a larger drive shaft) is implemented. Yet another alternative rotor design 506 is shown in Figure 31D incorporating an asymmetrical shape, which would smooth the volume reduction curve, allowing for increased time for heat transfer to occur at higher pressures. Alternative rotor shapes may be implemented for different curvatures or needs for increased volume in the compression chamber.

[0114] The rotor surface may be smooth in embodiments with contacting tip seals to minimize wear on the tip seal. In alternative embodiments, it may be advantageous to put surface texture on the rotor to create turbulence that may improve the performance of non-contacting seals. In other embodiments, the rotor casing's interior cylindrical wall may further be textured to produce additional turbulence, both for sealing and heat transfer benefits. This texturing could be achieved through machining of the parts or by utilizing a surface coating. Another

method of achieving the texture would be through blasting with a waterjet, sandblast, or similar device to create an irregular surface.

[0115] The main casing 110 may further utilize a removable cylinder liner. This liner may feature microsurfacing to induce turbulence for the benefits noted
5 above. The liner may also act as a wear surface to increase the reliability of the rotor and casing. The removable liner could be replaced at regular intervals as part of a recommended maintenance schedule. The rotor may also include a liner.

[0116] The exterior of the main casing 110 may also be modified to meet application specific parameters. For example, in subsea applications, the casing may
10 require to be significantly thickened to withstand exterior pressure, or placed within a secondary pressure vessel. Other applications may benefit from the exterior of the casing having a rectangular or square profile to facilitate mounting exterior objects or stacking multiple compressors. Liquid may be circulated in the casing interior to achieve additional heat transfer or to equalize pressure in the case of subsea
15 applications for example.

[0117] As shown in Figure 32A and B, the combination of the rotor 500 (here depicted with rotor end caps 590), the gate 600, and drive shaft 140, provide for a more efficient manner of compressing fluids in a cylinder. The gate is aligned along the length of the rotor to separate and define the inlet portion and
20 compression portion as the rotor turns.

[0118] The drive shaft 140 is mounted to endplates 120 in the preferred embodiment using one spherical roller bearing in each endplate 120. More than one bearing may be used in each endplate 120, in order to increase total load capacity. A grease pump (not shown) is used to provide lubrication to the bearings.
25 Various types of other bearings may be utilized depending on application specific parameters, including roller bearings, ball bearings, needle bearings, conical bearings, cylindrical bearings, journal bearings, etc. Different lubrication systems using grease, oil, or other lubricants may also be used. Further, dry lubrication systems or materials may be used. Additionally, applications in which dynamic
30 imbalance may occur may benefit from multi-bearing arrangements to support stray axial loads.

[0119] Operation of gates in accordance with embodiments of the present invention are shown in Figures 8, 17, 22, 24B, 26A-F, 27A-F, 28, 32A-B, and 33-36. As shown in Figures 26A-F and 27A-F, gate 600 creates a pressure boundary between an intake volume 412 and a compression volume 414. The intake volume
5 412 is in communication with the inlet 420. The compression volume 414 is in communication with the outlet 430. Resembling a reciprocating, rectangular piston, the gate 600 rises and falls in time with the turning of the rotor 500.

[0120] The gate 600 may include an optional tip seal 620 that makes contact with the rotor 500, providing an interface between the rotor 500 and the gate 600.

10 Tip seal 620 consists of a strip of material at the tip of the gate 600 that rides against rotor 500. The tip seal 620 could be made of different materials, including polymers, graphite, and metal, and could take a variety of geometries, such as a curved, flat, or angled surface. The tip seal 620 may be backed by pressurized fluid or a spring force provided by springs or elastomers. This provides a return force to
15 keep the tip seal 620 in sealing contact with the rotor 500.

[0121] Different types of contacting tips may be used with the gate 600. As shown in Figure 35, a roller tip 650 may be used. The roller tip 650 rotates as it makes contact with the turning rotor 500. Also, tips of differing strengths may be used. For example, a tip seal 620 or roller tip 650 may be made of softer metal that
20 would gradually wear down before the rotor 500 surfaces would wear.

[0122] Alternatively, a non-contacting seal may be used. Accordingly, the tip seal may be omitted. In these embodiments, the topmost portion of the gate 600 is placed proximate, but not necessarily in contact with, the rotor 500 as it turns. The amount of allowable gap may be adjusted depending on application parameters.

25 [0123] As shown in Figures 34A and 34B, in an embodiment in which the tip of the gate 600 does not contact the rotor 500, the tip may include notches 640 that serve to keep gas pocketed against the tip of the gate 600. The entrained fluid, in either gas or liquid form, assists in providing a non-contacting seal. As one of ordinary skill in the art would appreciate, the number and size of the notches is a
30 matter of design choice dependent on the compressor specifications.

[0124] Alternatively, liquid may be injected from the gate itself. As shown in Figure 36, a cross-sectional view of a portion of a gate, one or more channels 660 from which a fluid may pass may be built into the gate. In one such embodiment, a liquid can pass through a plurality of channels 660 to form a liquid seal between the topmost portion of the gate 600 and the rotor 500 as it turns. In another embodiment, residual compressed fluid may be inserted through one or more channels 660. Further still, the gate 600 may be shaped to match the curvature of portions of the rotor 500 to minimize the gap between the gate 600 and the rotor 500.

[0125] Preferred embodiments enclose the gate in a gate casing. As shown in Figures 8 and 17, the gate 600 is encompassed by the gate casing 150, including notches, one of which is shown as item 158. The notches hold the gate seals, which ensure that the compressed fluid will not release from the compression volume 414 through the interface between gate 600 and gate casing 150 as gate 600 moves up and down. The gate seals may be made of various materials, including polymers, graphite or metal. A variety of different geometries may be used for these seals. Various embodiments could utilize different notch geometries, including ones in which the notches may pass through the gate casing, in part or in full.

[0126] The seals may use energizing forces provided by springs or elastomers with the assembly of the gate casing 150 inducing compression on the seals. Pressurized fluid may also be used to energize the seals.

[0127] A rotor face seal may also be placed on the rotor 500 to provide for an interface between the rotor 500 and the endplates 120. An outer rotor face seal is placed along the exterior edge of the rotor 500, preventing fluid from escaping past the end of the rotor 500. A secondary inner rotor face seal is placed on the rotor face at a smaller radius to prevent any fluid that escapes past the outer rotor face seal from escaping the compressor entirely. This seal may use the same or other materials as the gate seal. Various geometries may be used to optimize the effectiveness of the seals. These seals may use energizing forces provided by springs, elastomers or pressurized fluid.

[0128] Minimizing the possibility of fluids leaking to the exterior of the main housing 100 is desirable. Various seals, such as gaskets and o-rings, are used to seal external connections between parts. For example, in a preferred embodiment, a double o-ring seal is used between the main casing 110 and endplates 120. Further seals are utilized around the drive shaft 140 to prevent leakage of any fluids making it past the rotor face seals. A lip seal is used to seal the drive shaft 140 where it passes through the endplates 120. Other forms of seals could also be used, such as mechanical or labyrinth seals.

[0129] It is desirable to achieve near isothermal compression. To provide cooling during the compression process, liquid injection is used. In preferred embodiments, the liquid is atomized to provide increased surface area for heat absorption. In other embodiments, different spray applications or other means of injecting liquids may be used.

[0130] Liquid injection is used to cool the fluid as it is compressed, increasing the efficiency of the compression process. Cooling allows most of the input energy to be used for compression rather than heat generation in the gas. The liquid has dramatically superior heat absorption characteristics compared to gas, allowing the liquid to absorb heat and minimize temperature increase of the working fluid, achieving near isothermal compression. As shown in Figures 8 and 17, liquid injector assemblies 130 are attached to the main casing 110. Liquid injector housings 132 include an adapter for the liquid source 134 (if it is not included with the nozzle) and a nozzle 136. Liquid is injected by way of a nozzle 136 directly into the rotor casing volume 410.

[0131] The amount and timing of liquid injection may be controlled by a variety of implements including a computer-based controller capable of measuring the liquid drainage rate, liquid levels in the chamber, and/or any rotational resistance due to liquid accumulation through a variety of sensors. Valves or solenoids may be used in conjunction with the nozzles to selectively control injection timing. Variable orifice control may also be used to regulate the amount of liquid injection and other characteristics.

[0132] Analytical and experimental results are used to optimize the number, location, and spray direction of the injectors 136. These injectors 136 may be located in the periphery of the cylinder. Liquid injection may also occur through the rotor or gate. The current embodiment of the design has two nozzles located at 12 o'clock and 10 o'clock. Different application parameters will also influence preferred nozzle arrays.

[0133] The nozzle array is designed for a high flow rate of greater than 5 gallons per minute and to be capable of extremely small droplet sizes of 150 microns or less at a low differential pressure of less than 100 psi. Two exemplary nozzles are Spraying Systems Co. Part Number: 1/4HHSJ-SS12007 and Bex Spray Nozzles Part Number: 1/4YS12007. The preferred flow rate and droplet size ranges will vary with application parameters. Alternative nozzle styles may also be used. For example, one embodiment may use micro-perforations in the cylinder through which to inject liquid, counting on the small size of the holes to create sufficiently small droplets. Other embodiments may include various off the shelf or custom designed nozzles which, when combined into an array, meet the injection requirements necessary for a given application.

[0134] As discussed above, the rate of heat transfer is improved by using such atomizing nozzles to inject very small droplets of liquid into the compression chamber. Because the rate of heat transfer is proportional to the surface area of liquid across which heat transfer can occur, the creation of smaller droplets improves cooling. Numerous cooling liquids may be used. For example, water, triethylene glycol, and various types of oils and other hydrocarbons may be used. Ethylene glycol, propylene glycol, methanol or other alcohols in case phase change characteristics are desired may be used. Refrigerants such as ammonia and others may also be used. Further, various additives may be combined with the cooling liquid to achieve desired characteristics. Along with the heat transfer and heat absorption properties of the liquid helping to cool the compression process, vaporization of the liquid may also be utilized in some embodiments of the design to take advantage of the large cooling effect due to phase change.

[0135] The effect of liquid coalescence is also addressed in the preferred embodiments. Liquid accumulation can provide resistance against the compressing mechanism, eventually resulting in hydrolock in which all motion of the compressor is stopped, causing potentially irreparable harm. As is shown in the embodiments of Figures 8 and 17, the inlet 420 and outlet 430 are located at the bottom of the rotor casing 400 on opposite sides of the gate 600, thus providing an efficient location for both intake of fluid to be compressed and exhausting of compressed fluid and the injected liquid. A valve is not necessary at the inlet 420. The inclusion of a dwell seal allows the inlet 420 to be an open port, simplifying the system and reducing inefficiencies associated with inlet valves. However, if desirable, an inlet valve could also be incorporated. Additional features may be added at the inlet to induce turbulence to provide enhanced thermal transfer and other benefits. Hardened materials may be used at the inlet and other locations of the compressor to protect against cavitation when liquid/gas mixtures enter into choke and other cavitation-inducing conditions.

[0136] Alternative embodiments may include an inlet located at positions other than shown in the figures. Additionally, multiple inlets may be located along the periphery of the cylinder. These could be utilized in isolation or combination to accommodate inlet streams of varying pressures and flow rates. The inlet ports can also be enlarged or moved, either automatically or manually, to vary the displacement of the compressor.

[0137] In these embodiments, multi-phase compression is utilized, thus the outlet system allows for the passage of both gas and liquid. Placement of outlet 430 near the bottom of the rotor casing 400 provides for a drain for the liquid. This minimizes the risk of hydrolock found in other liquid injection compressors. A small clearance volume allows any liquids that remain within the chamber to be accommodated. Gravity assists in collecting and eliminating the excess liquid, preventing liquid accumulation over subsequent cycles. Additionally, the sweeping motion of the rotor helps to ensure that most liquid is removed from the compressor during each compression cycle.

[0138] Outlet valves allow gas and liquid to flow out of the compressor once the desired pressure within the compression chamber is reached. Due to the presence of liquid in the working fluid, valves that minimize or eliminate changes in direction for the outflowing working fluid are desirable. This prevents the hammering effect of liquids as they change direction. Additionally, it is desirable to minimize clearance volume.

[0139] Reed valves may be desirable as outlet valves. As one of ordinary skill in the art would appreciate, other types of valves known or as yet unknown may be utilized. Hoerbiger type R, CO, and Reed valves may be acceptable. Additionally, CT, HDS, CE, CM or Poppet valves may be considered. Other embodiments may use valves in other locations in the casing that allow gas to exit once the gas has reached a given pressure. In such embodiments, various styles of valves may be used. Passive or directly-actuated valves may be used and valve controllers may also be implemented.

[0140] In the presently preferred embodiments, the outlet valves are located near the bottom of the casing and serve to allow exhausting of liquid and compressed gas from the high pressure portion. In other embodiments, it may be useful to provide additional outlet valves located along periphery of main casing in locations other than near the bottom. Some embodiments may also benefit from outlets placed on the endplates. In still other embodiments, it may be desirable to separate the outlet valves into two types of valves - one predominately for high pressured gas, the other for liquid drainage. In these embodiments, the two or more types of valves may be located near each other, or in different locations.

[0141] As shown in Figures 8 and 17, the sealing portion 510 of the rotor effectively precludes fluid communication between the outlet and inlet ports by way of the creation of a dwell seal. The interface between the rotor 500 and gate 600 further precludes fluid communication between the outlet and inlet ports through use of a non-contacting seal or tip seal 620. In this way, the compressor is able to prevent any return and venting of fluid even when running at low speeds. Existing rotary compressors, when running at low speeds, have a leakage path from the outlet

to the inlet and thus depend on the speed of rotation to minimize venting/leakage losses through this flowpath.

[0142] The high pressure working fluid exerts a large horizontal force on the gate 600. Despite the rigidity of the gate struts 210, this force will cause the gate 600 to bend and press against the inlet side of the gate casing 152. Specialized coatings that are very hard and have low coefficients of friction can coat both surfaces to minimize friction and wear from the sliding of the gate 600 against the gate casing 152. A fluid bearing can also be utilized. Alternatively, pegs (not shown) can extend from the side of the gate 600 into gate casing 150 to help support the gate 600 against this horizontal force.

[0143] The large horizontal forces encountered by the gate may also require additional considerations to reduce sliding friction of the gate's reciprocating motion. Various types of lubricants, such as greases or oils may be used. These lubricants may further be pressurized to help resist the force pressing the gate against the gate casing. Components may also provide a passive source of lubrication for sliding parts via lubricant-impregnated or self-lubricating materials. In the absence of, or in conjunction with, lubrication, replaceable wear elements may be used on sliding parts to ensure reliable operation contingent on adherence to maintenance schedules. As one of ordinary skill in the art would appreciate, replaceable wear elements may also be utilized on various other wear surfaces within the compressor.

[0144] The compressor structure may be comprised of materials such as aluminum, carbon steel, stainless steel, titanium, tungsten, or brass. Materials may be chosen based on corrosion resistance, strength, density, and cost. Seals may be comprised of polymers, such as PTFE, HDPE, PEEK™, acetal copolymer, etc., graphite, cast iron, or ceramics. Other materials known or unknown may be utilized. Coatings may also be used to enhance material properties.

[0145] As one of ordinary skill in the art can appreciate, various techniques may be utilized to manufacture and assemble the invention that may affect specific features of the design. For example, the main casing 110 may be manufactured using a casting process. In this scenario, the nozzle housings 132, gate casing 150,

or other components may be formed in singularity with the main casing 110. Similarly, the rotor 500 and drive shaft 140 may be built as a single piece, either due to strength requirements or chosen manufacturing technique.

5 [0146] Further benefits may be achieved by utilizing elements exterior to the compressor envelope. A flywheel may be added to the drive shaft 140 to smooth the torque curve encountered during the rotation. A flywheel or other exterior shaft attachment may also be used to help achieve balanced rotation. Applications requiring multiple compressors may combine multiple compressors on a single drive shaft with rotors mounted out of phase to also achieve a smoothed torque curve. A bell housing or other shaft coupling may be used to attach the drive shaft to a driving force such as engine or electric motor to minimize effects of misalignment and increase torque transfer efficiency. Accessory components such as pumps or generators may be driven by the drive shaft using belts, direct couplings, gears, or other transmission mechanisms. Timing gears or belts may 10 further be utilized to synchronize accessory components where appropriate.

[0147] After exiting the valves the mix of liquid and gases may be separated through any of the following methods or a combination thereof: 1. Interception through the use of a mesh, vanes, intertwined fibers; 2. Inertial impaction against a surface; 3. Coalescence against other larger injected droplets; 4. Passing through a liquid curtain; 5. Bubbling through a liquid reservoir; 6. Brownian motion to aid in coalescence; 7. Change in direction; 8. Centrifugal motion for coalescence into walls and other structures; 9. Inertia change by rapid deceleration; and 10. Dehydration through the use of adsorbents or absorbents. 20

[0148] At the outlet of the compressor, a pulsation chamber may consist of cylindrical bottles or other cavities and elements, may be combined with any of the aforementioned separation methods to achieve pulsation dampening and attenuation as well as primary or final liquid coalescence. Other methods of separating the liquid and gases may be used as well. 25

[0149] The presently preferred embodiments could be modified to operate as an expander. Further, although descriptions have been used to describe the top and bottom and other directions, the orientation of the elements (e.g. the gate 600 at 30

the bottom of the rotor casing 400) should not be interpreted as limitations on the present invention.

[0150] While the foregoing written description of the invention enables one of ordinary skill to make and use what is considered presently to be the best mode thereof, those of ordinary skill will understand and appreciate the existence of variations, combinations, and equivalents of the specific embodiment, method, and examples herein. The invention should therefore not be limited by the above described embodiment, method, and examples, but by all embodiments and methods within the scope and spirit of the invention.

[0151] It is therefore intended that the foregoing detailed description be regarded as illustrative rather than limiting, and that it be understood that it is the following claims, including all equivalents, that are intended to define the spirit and scope of this invention. To the extent that “at least one” is used to highlight the possibility of a plurality of elements that may satisfy a claim element, this should not be interpreted as requiring “a” to mean singular only. “A” or “an” element may still be satisfied by a plurality of elements unless otherwise stated.

CLAIMS

1. A positive displacement compressor, comprising;
a cylindrical rotor casing, the rotor casing having an inlet port, an outlet port,
5 and an inner wall defining a rotor casing volume;
a rotor, the rotor having a sealing portion that corresponds to a curvature of
the inner wall of the rotor casing;
a liquid injector connected with the rotor casing to inject liquids into the
rotor casing volume; and
10 a gate, having a first end and a second end, and operable to move within the
rotor casing to located the first end proximate to the rotor as it turns;
wherein the gate separates an inlet volume and a compression volume in the
rotor casing volume, the inlet port suctions in gas, and the outlet expels both liquid
and gas.

15 2. The positive displacement compressor of claim 1, further comprising a
drive shaft, and wherein the rotor is concentrically mounted to the drive shaft.

3. The positive displacement compressor of claim 2 further comprising:
at least one cam mounted for concentric rotation around the drive shaft;
20 at least one cam follower connected to the cam;
at least one spring; and
a gate support arm connected to the spring and the cam follower.

4. The positive displacement compressor of claim 2 further comprising:
25 at least one cam mounted for concentric rotation around the drive shaft,
at least one cam follower bearing connected to an outer rim of the at least
one cam;
at least one cam follower bearing connected to an inner rim of the at least
one cam; and

a gate support arm connected to the cam follower bearing and a gate casing, the gate casing encompassing the gate;

wherein the rotation of the at least one cam operates to raise and lower the gate.

5

5. The positive displacement compressor of claim 1 further comprising a magnetic drive system for positioning the gate, the magnetic drive system comprising a plurality of rotor magnets located on the rotor and at least one gate magnet located one the gate.

10

6. The positive displacement compressor of claim 2 further comprising:

a belt connected with the drive shaft and a secondary shaft;

a cam connected to the secondary shaft;

a plurality of cam follower bearings connected to the cam; and

15

a gate support arm connected to the plurality of cam follower bearings and the gate.

7. The positive displacement compressor of claim 1 wherein the gate further comprises a channel through which liquid can pass into the rotor casing volume.

20

8. The positive displacement compressor of claim 1 further comprising an outlet port located near the cross-sectional bottom of the cylindrical rotor casing.

9. A positive displacement compressor, comprising:

25

a compression chamber, including a cylindrical-shaped casing having a first end, a second end, and an inner curved surface;

a shaft located axially in the compression chamber

a non-circular rotor mounted for concentric rotation around the shaft, the non-circular rotor having a sealing portion, the sealing portion having a curved

surface that corresponds with the inner curved surface of the cylindrical-shaped casing, and a non-sealing portion;

a gate, the gate having a first end and a second end; and

a gate positioning system, the gate positioning system operable to locate the first end of the gate proximate to the non-circular rotor as the rotor turns.

5

10. The positive displacement compressor of claim 9, further comprising at least one liquid injection nozzle located to provide injected fluids into the compression chamber.

10

11. The positive displacement compressor of claim 10, wherein the at least one liquid injection nozzle provides an atomized liquid spray.

12. The positive displacement compressor of claim 9, further comprising at least one outlet valve in fluid communication with the compression chamber to allow for the expulsion of liquids and gas.

15

13. The positive displacement compressor of claim 7, further comprising an outlet port for the compression chamber located on one side of the gate and an inlet port for the compression chamber located on the opposite side of the gate.

20

14. A rotor for a positive displacement compressor, the rotor comprising: a cylinder having a oval-shaped cross section, the oval shaped cross section including a sealing portion and a non-sealing portion, the sealing portion having a constant radius, the non-sealing portion having a variable radius defined by a cycloidal function;

25

at least one lightening hole in the cylinder; and

at least one counterweight in the cylinder.

30

15. A positive displacement compressor, comprising:

a cylindrical rotor casing defining a rotor casing volume, the rotor casing having a first end and a second end aligned horizontally;

a drive shaft mounted coaxially through the rotor casing;

an oval-shaped rotor mounted for concentric rotation about a drive shaft's axis of rotation, the rotor having a sealing portion that aligns with the curvature of and is proximate to an inside surface of the rotor casing and a non-sealing portion;

at least one liquid injector mounted to an opening in the rotor casing and operable to inject an atomized liquid into the rotor casing volume;

a gate located at the bottom of the rotor casing and operable to move up and down;

an inlet located on the rotor casing on one side of the gate;

an outlet port located on the rotor casing on the opposite side of the gate;

an outlet valve in fluid communication with the outlet port, the outlet valve operable to expel liquid and gas;

wherein the gate moves up and down so that a portion is proximate to the rotor as it turns, the gate defining an inlet fluid portion of the rotor casing volume and a compression portion of the rotor casing volume, the inlet portion in communication with the inlet, and the compression portion in communication with the outlet port.

16. A method for compressing a fluid, the method comprising:

providing a rotary compressor, the rotary compressor having a rotor, rotor casing, intake volume, a compression volume, and outlet valve;

receiving air into the intake volume;

rotating the rotor to increase the intake volume and decrease the compression volume;

injecting cooling liquid into the chamber;

rotating the rotor to further increase and decrease the compression volume;

opening the outlet valve to release compressed gas and liquid; and

separating the liquid from the compressed gas.

17. The method of claim 16 wherein injected cooling liquid is atomized when injected, absorbs heat, and is directed toward the outlet valve.

18. The method of claim 16 wherein the rotor concentrically rotates.

5

19. A positive displacement compressor, comprising:

a compression chamber, including a cylindrical-shaped casing having a first end and a second end, the first and second end aligned horizontally;

a shaft located axially in the compression chamber;

10

a rotor concentrically mounted to the shaft;

liquid injectors located to inject liquid into the compression chamber; and

a dual purpose outlet operable to release gas and liquid.

20. The positive displacement compressor of claim 19, wherein the rotor includes a curved portion that forms a seal with the cylindrical-shaped casing, and balancing holes.

15

Fig. 1

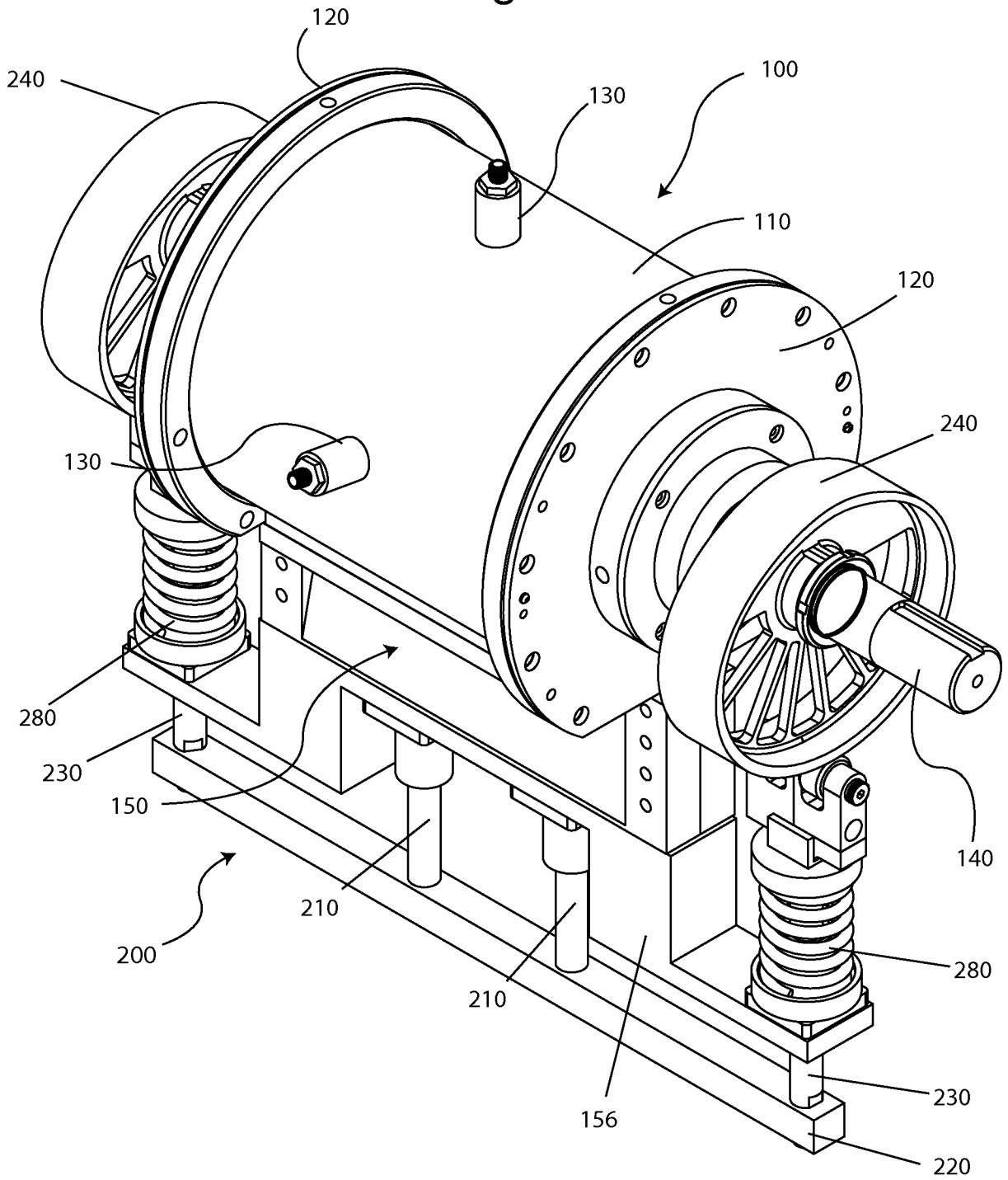


Fig. 2

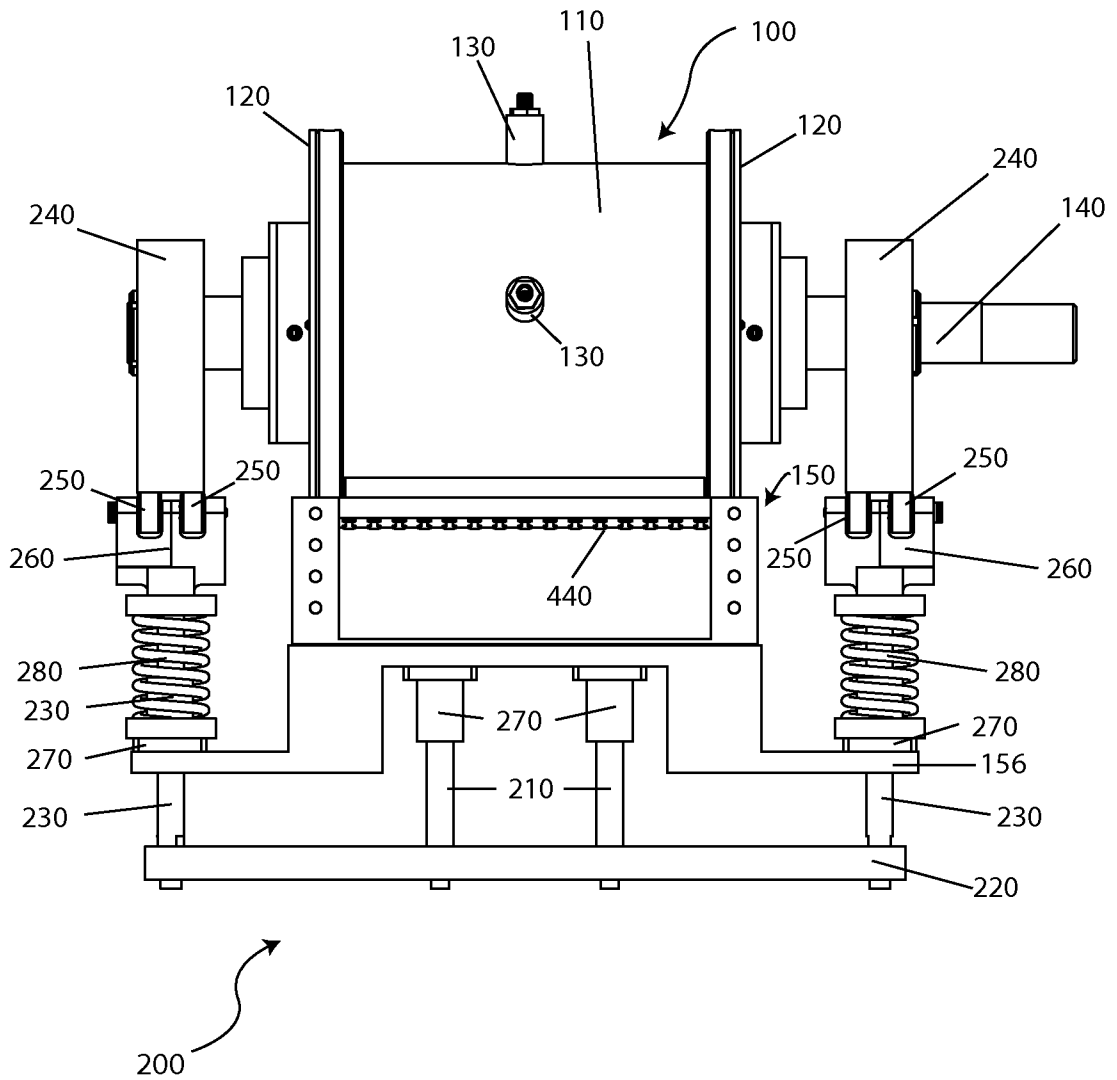


Fig. 4

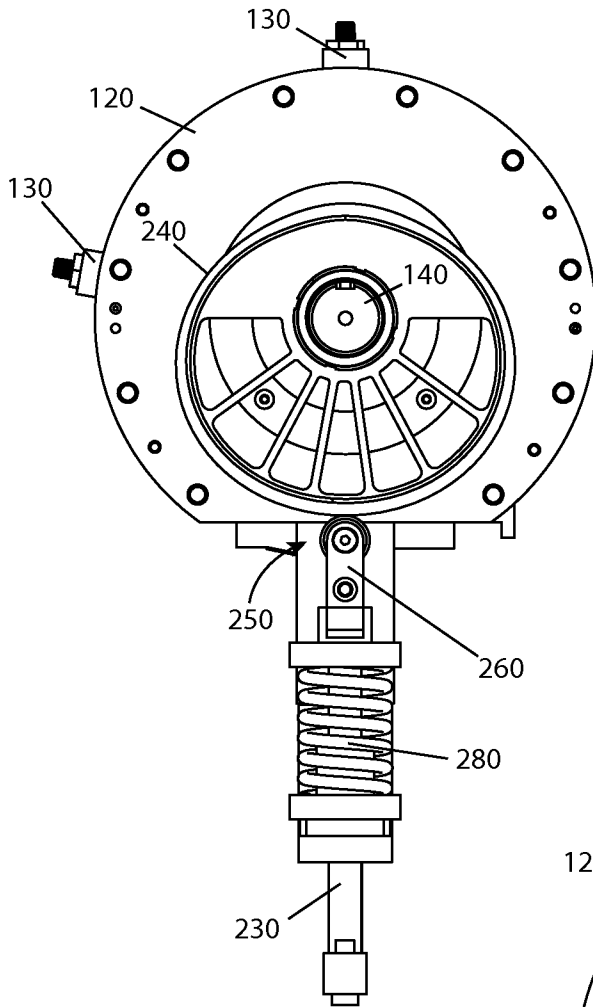
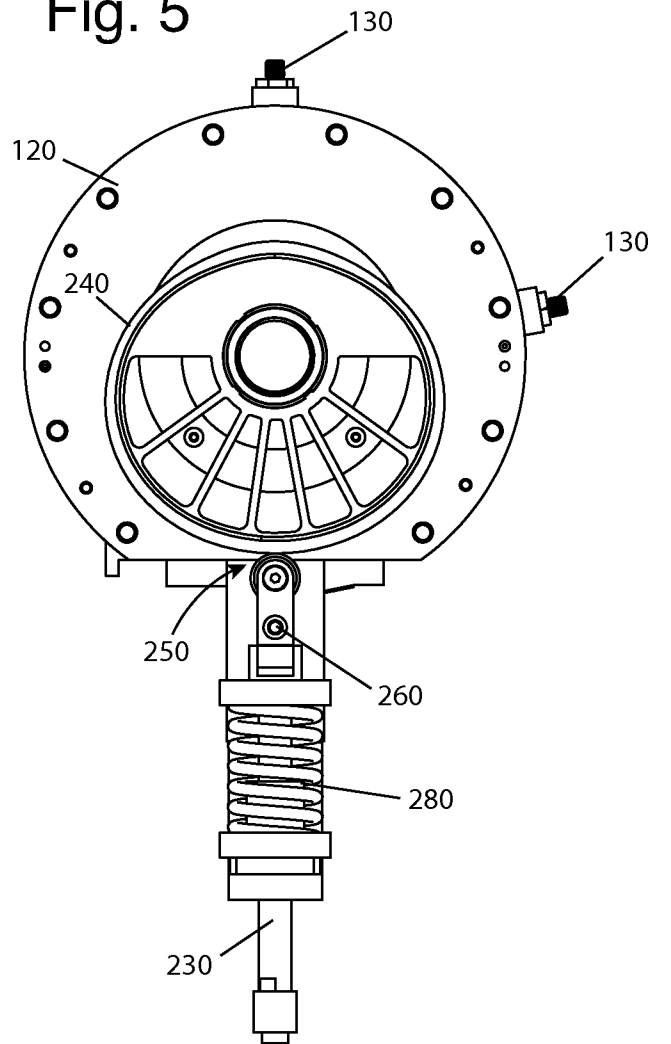


Fig. 5



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Fig. 6

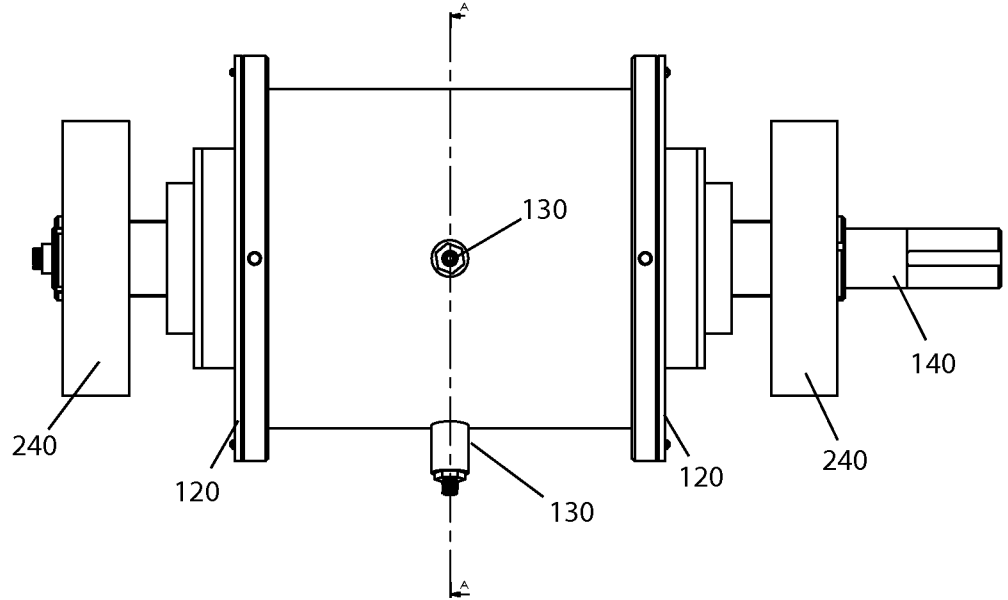
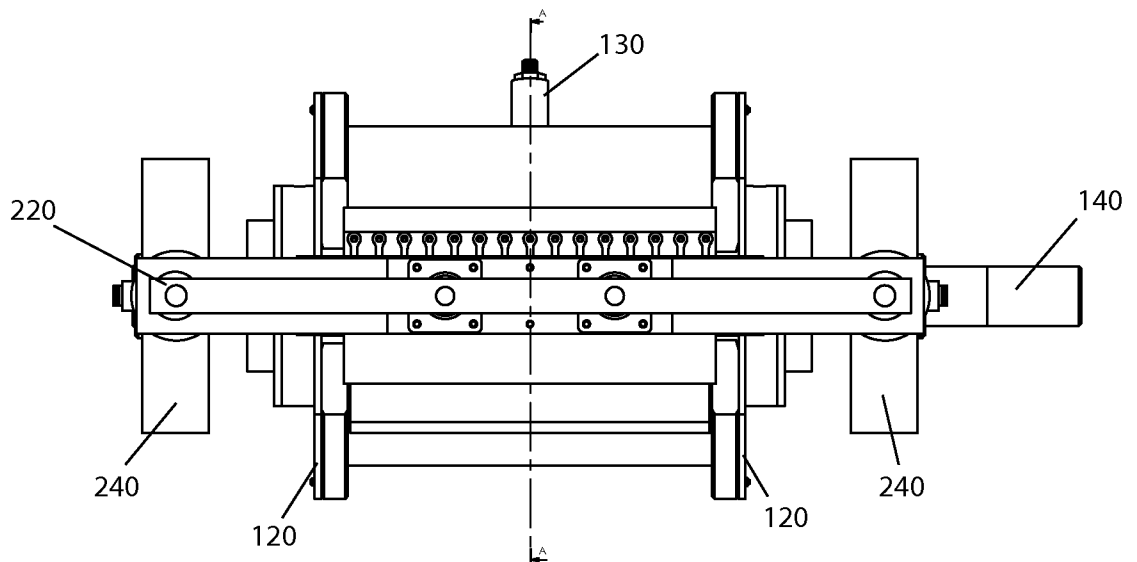
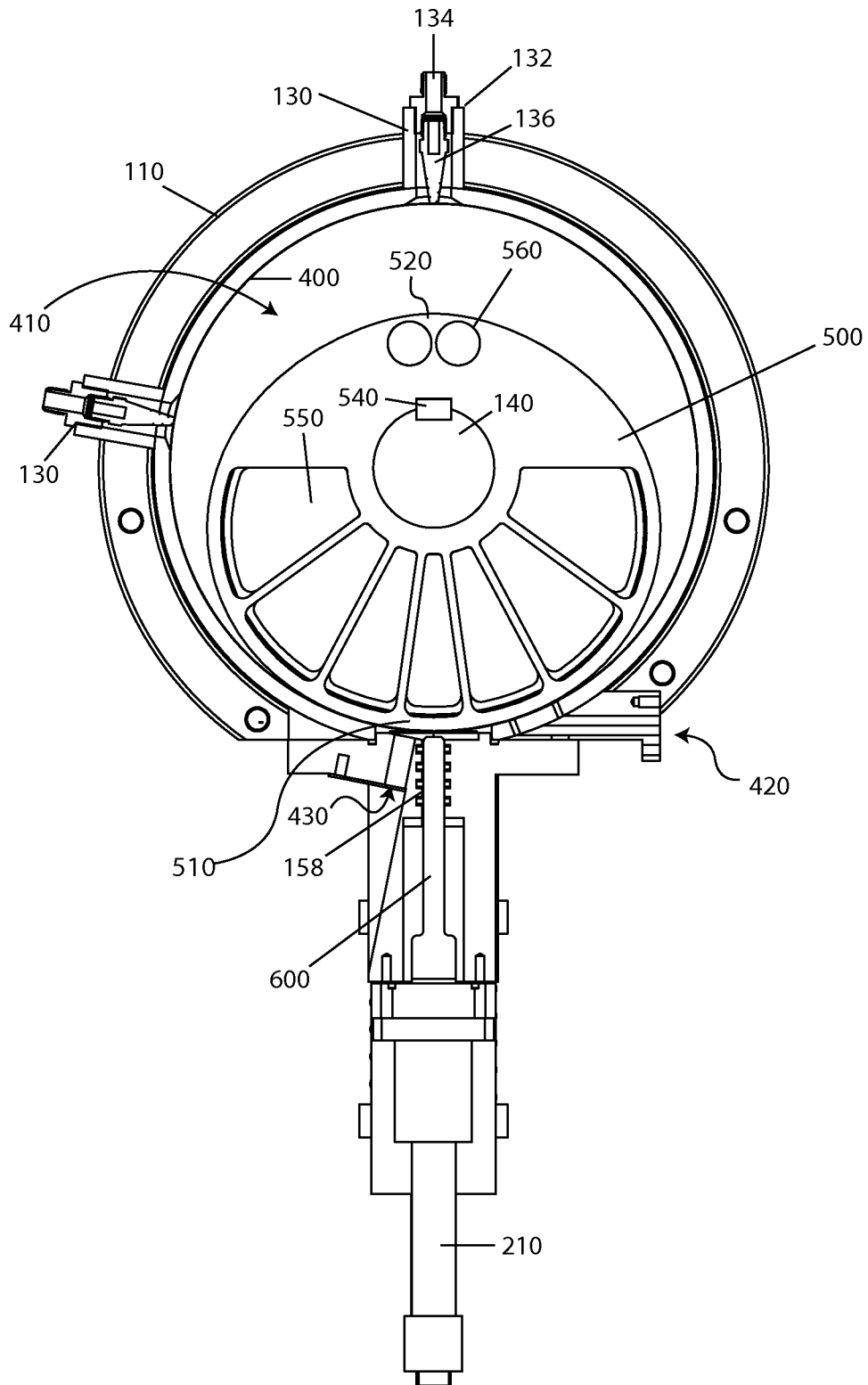


Fig. 7



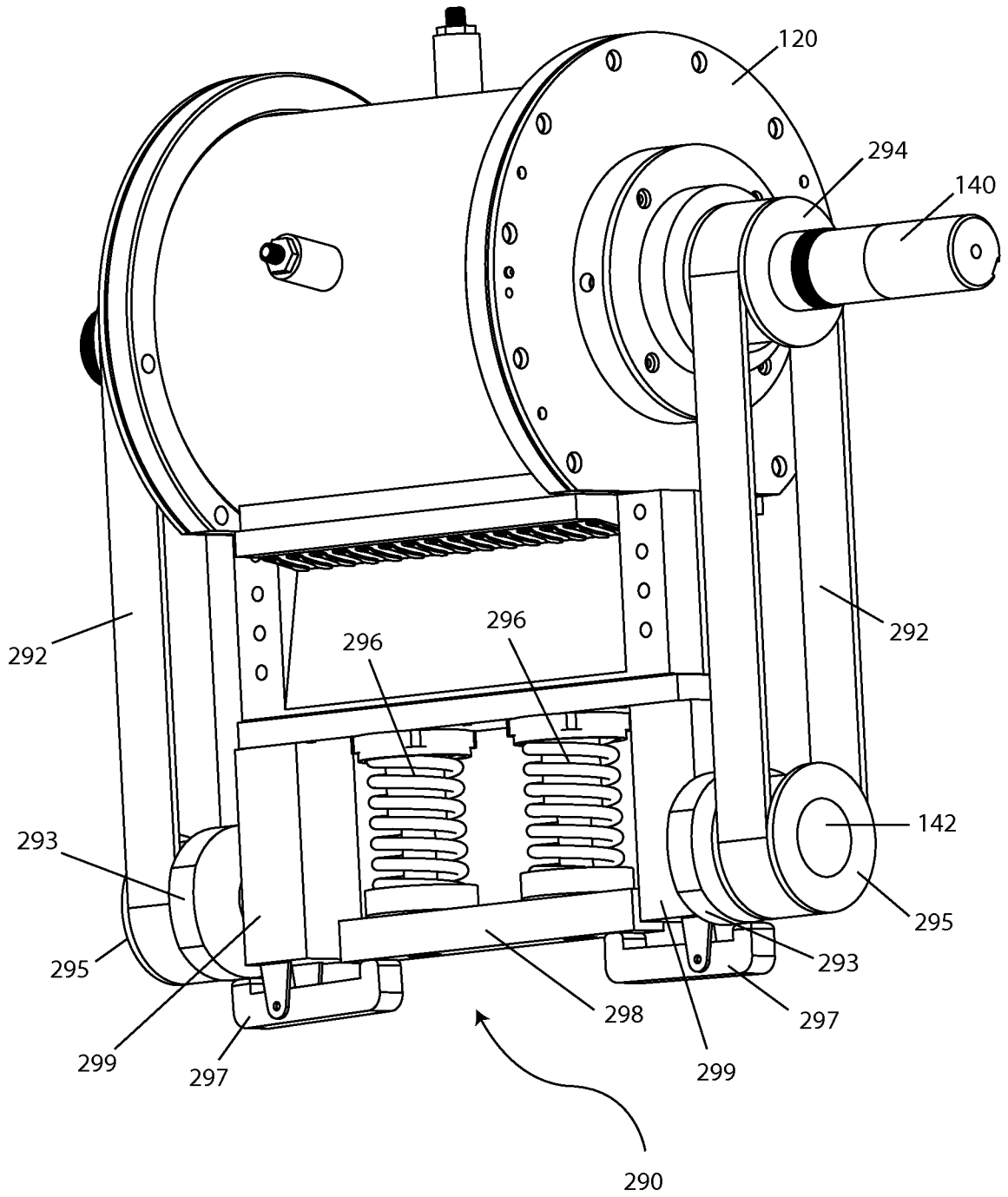
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Fig. 8



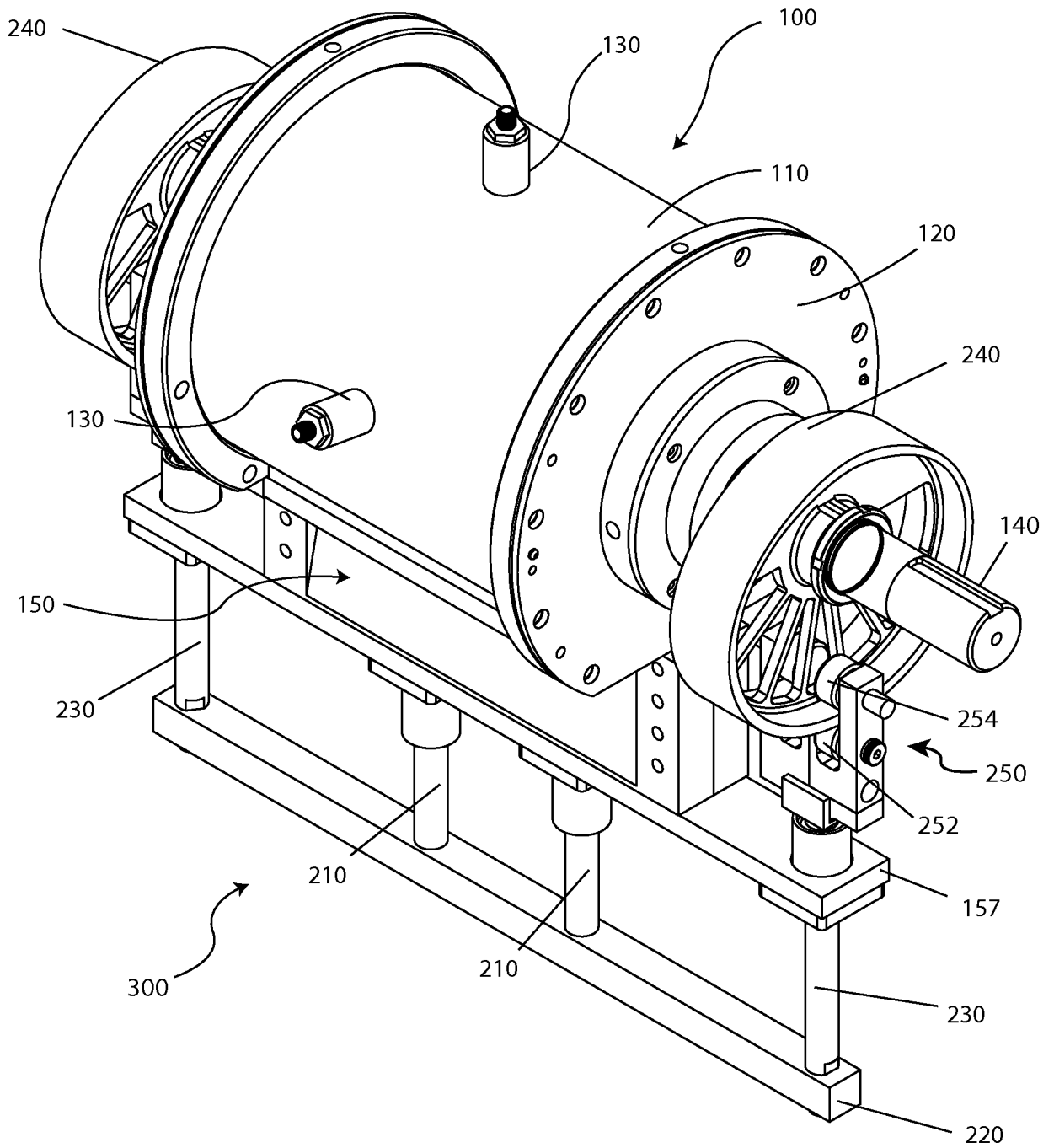
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Fig. 9



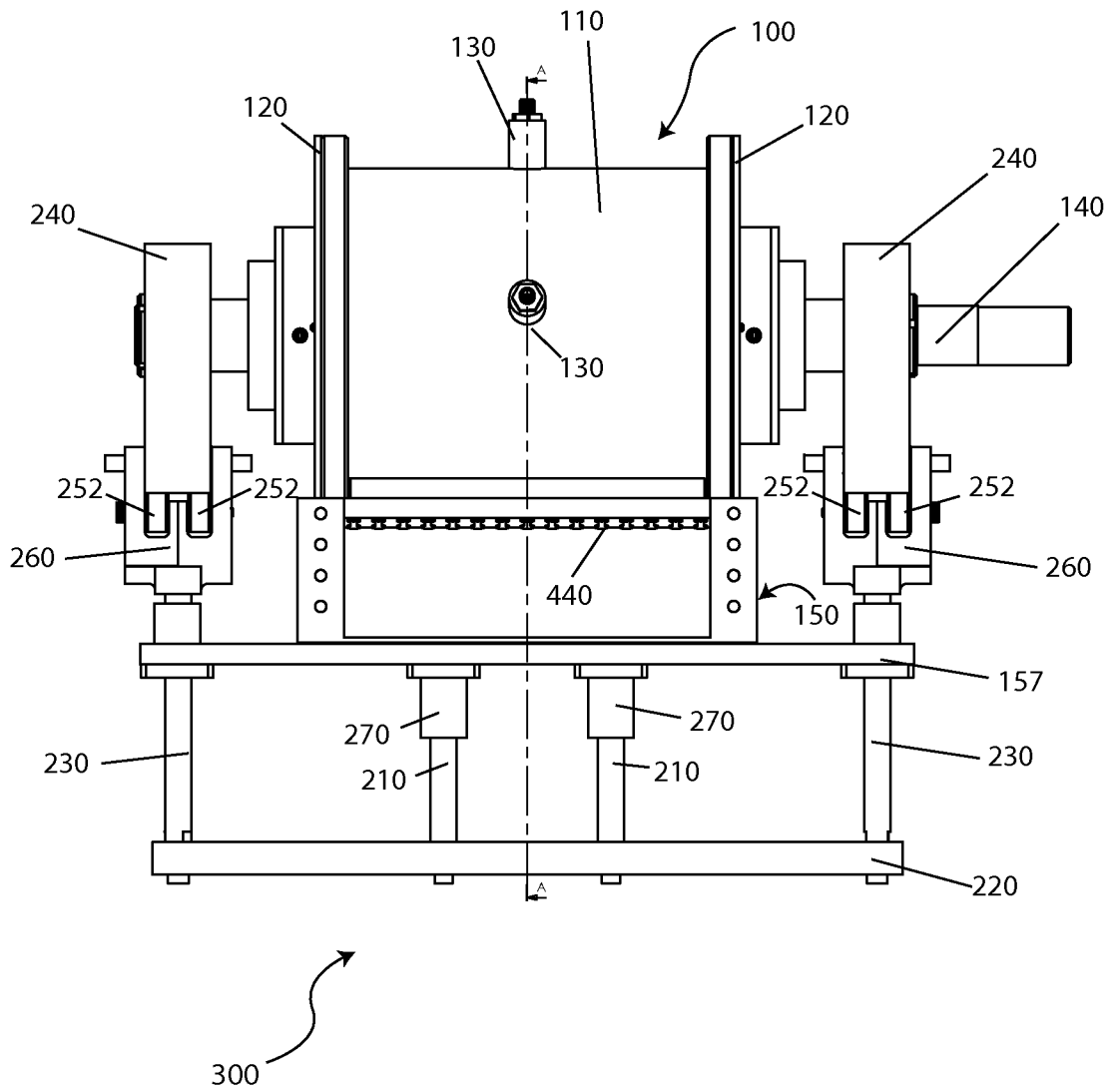
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Fig. 10



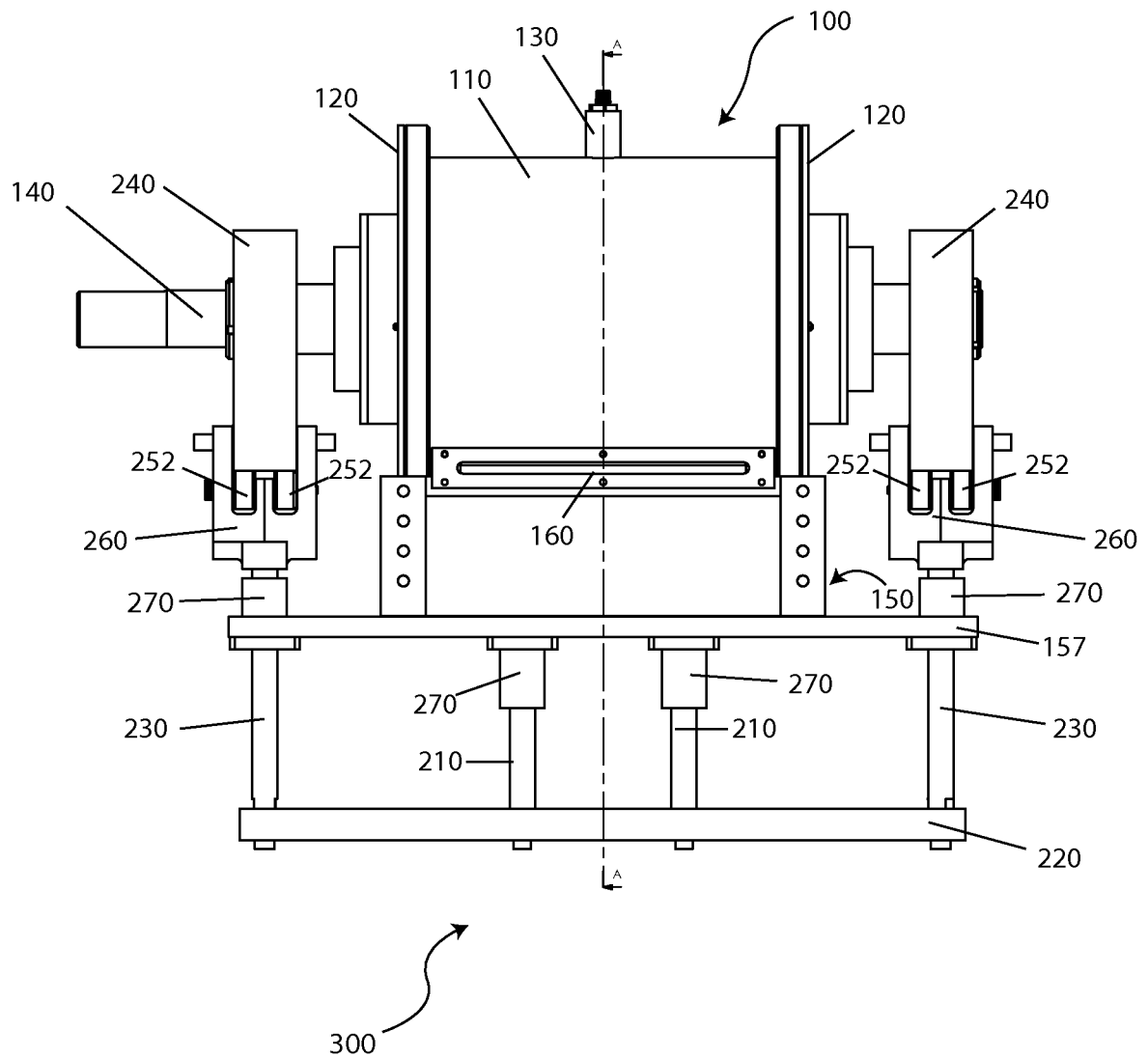
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Fig. 11



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Fig. 12



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Fig. 13

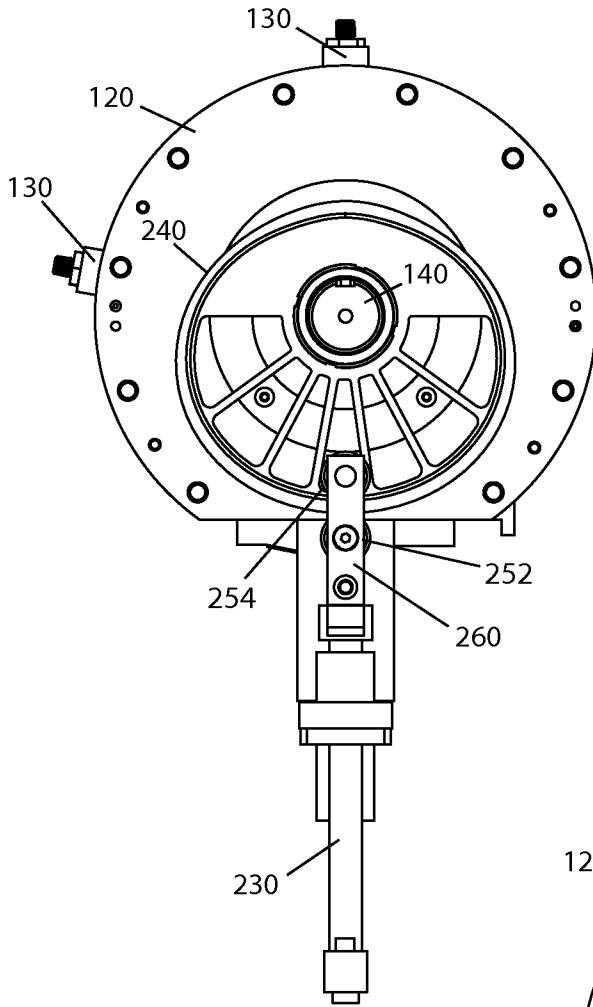
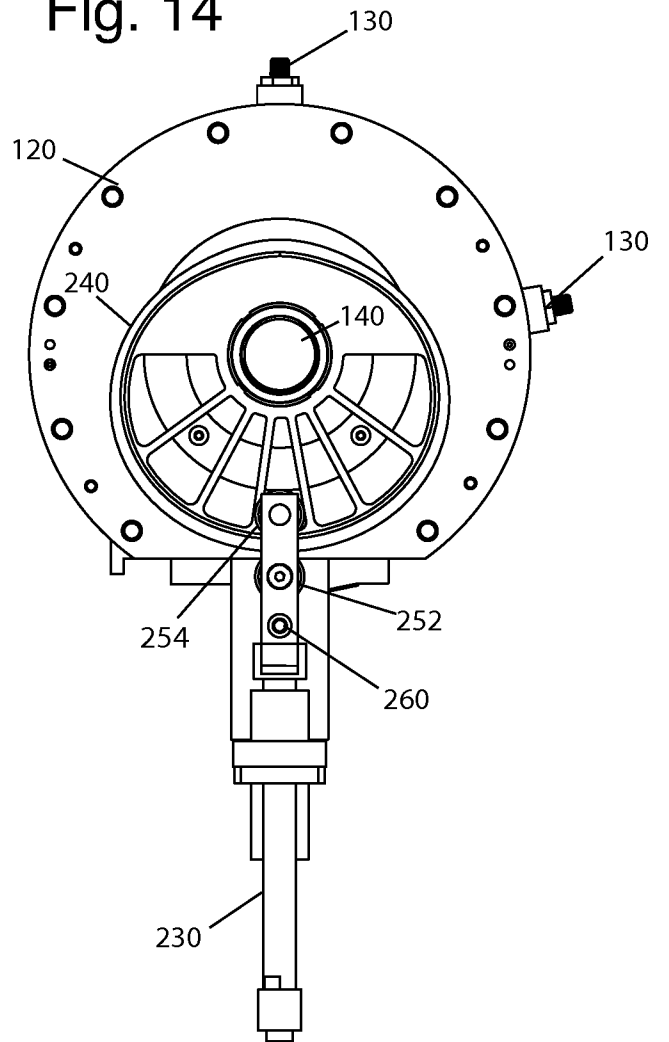


Fig. 14



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Fig. 15

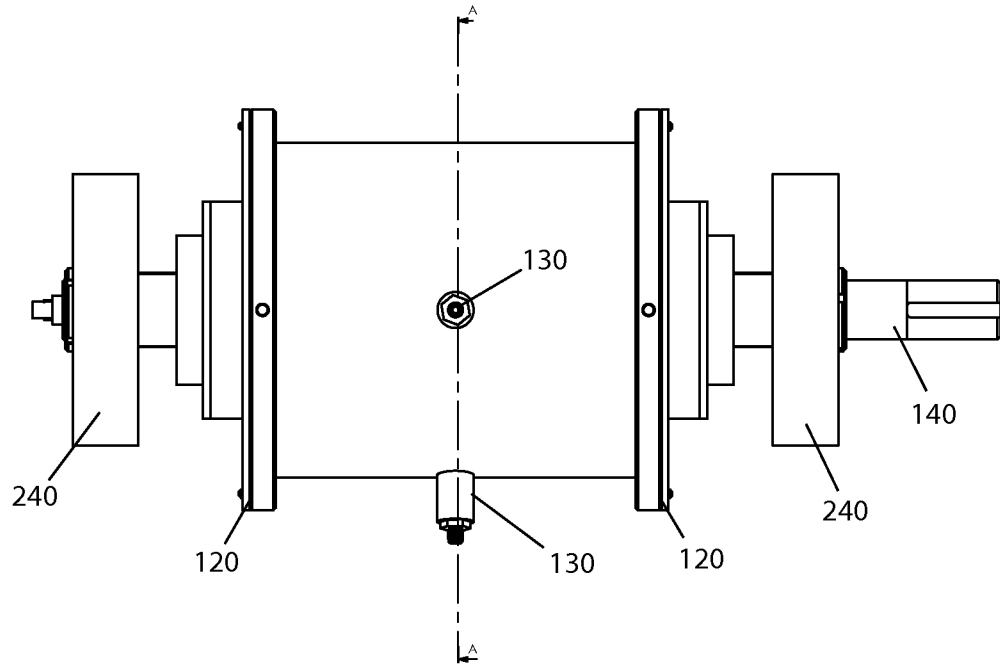
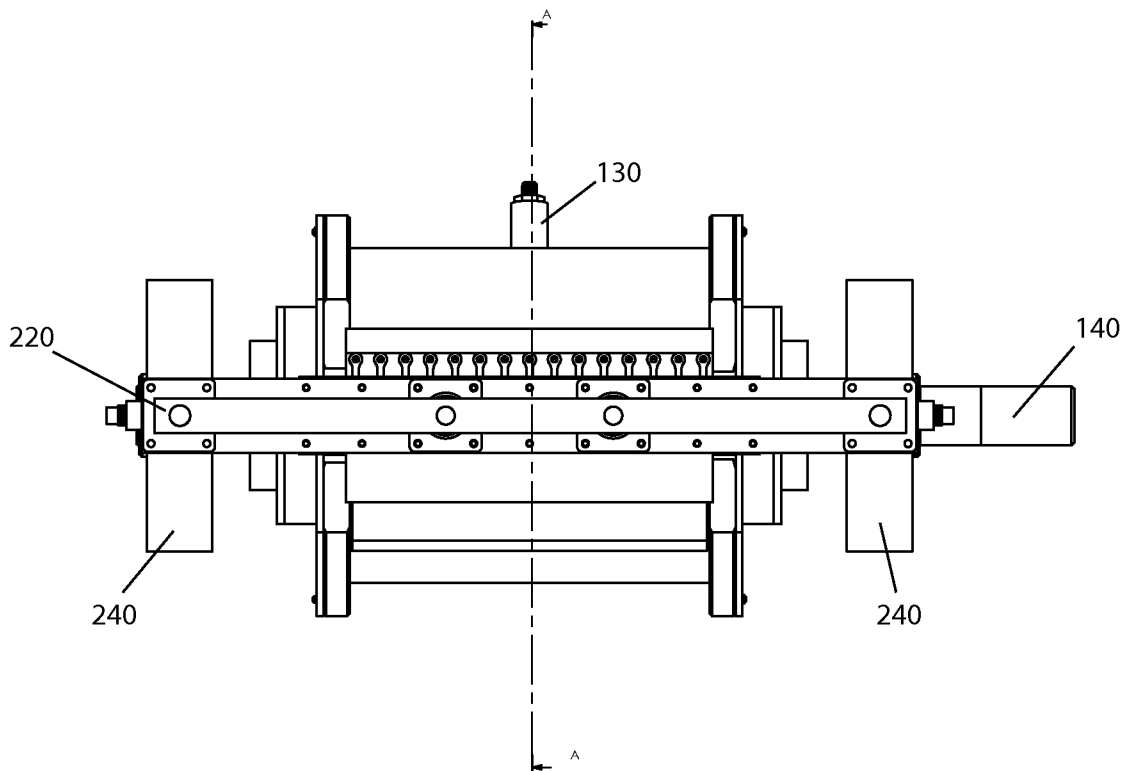


Fig. 16



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Fig. 17

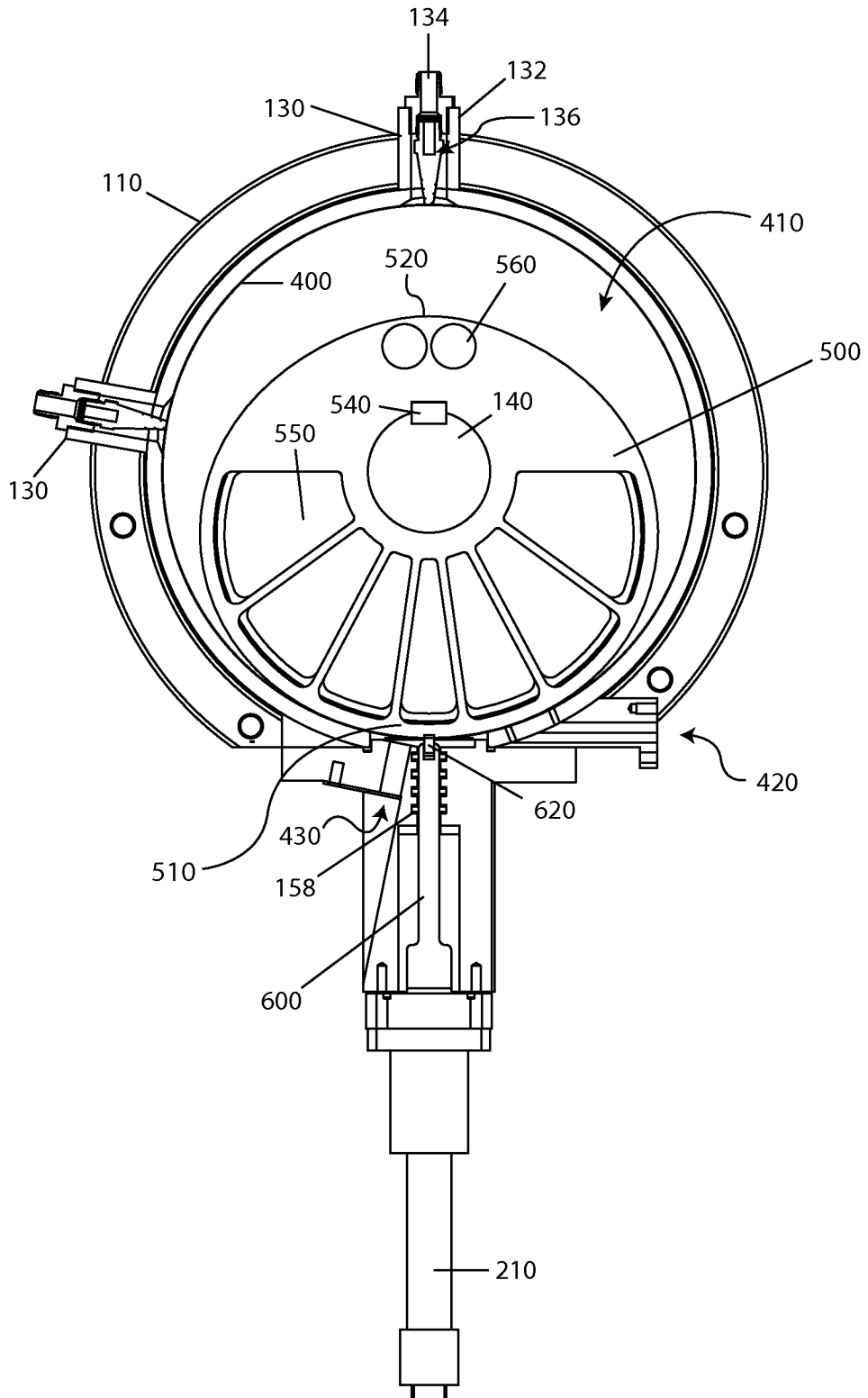


Fig. 18

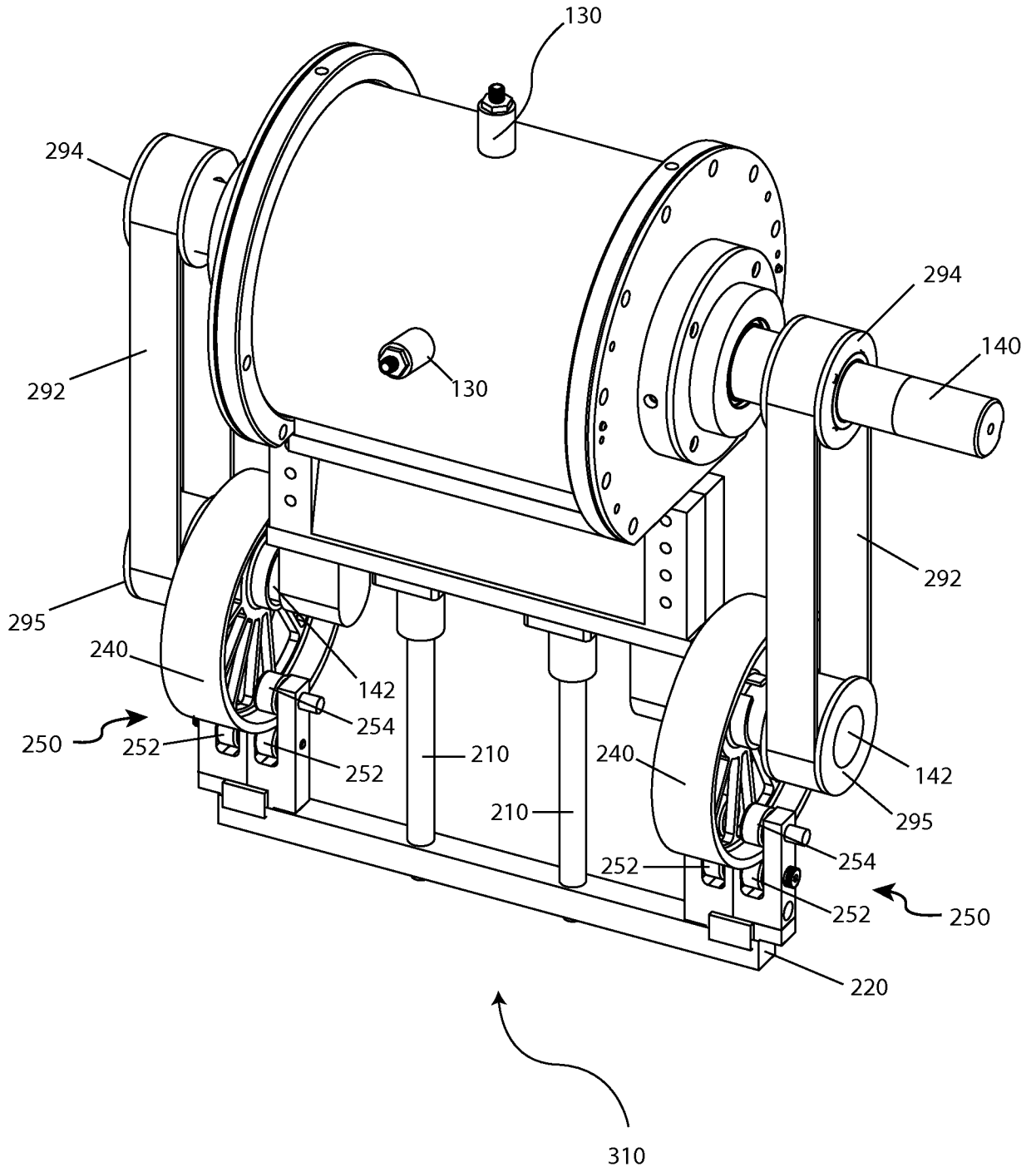


Fig. 20

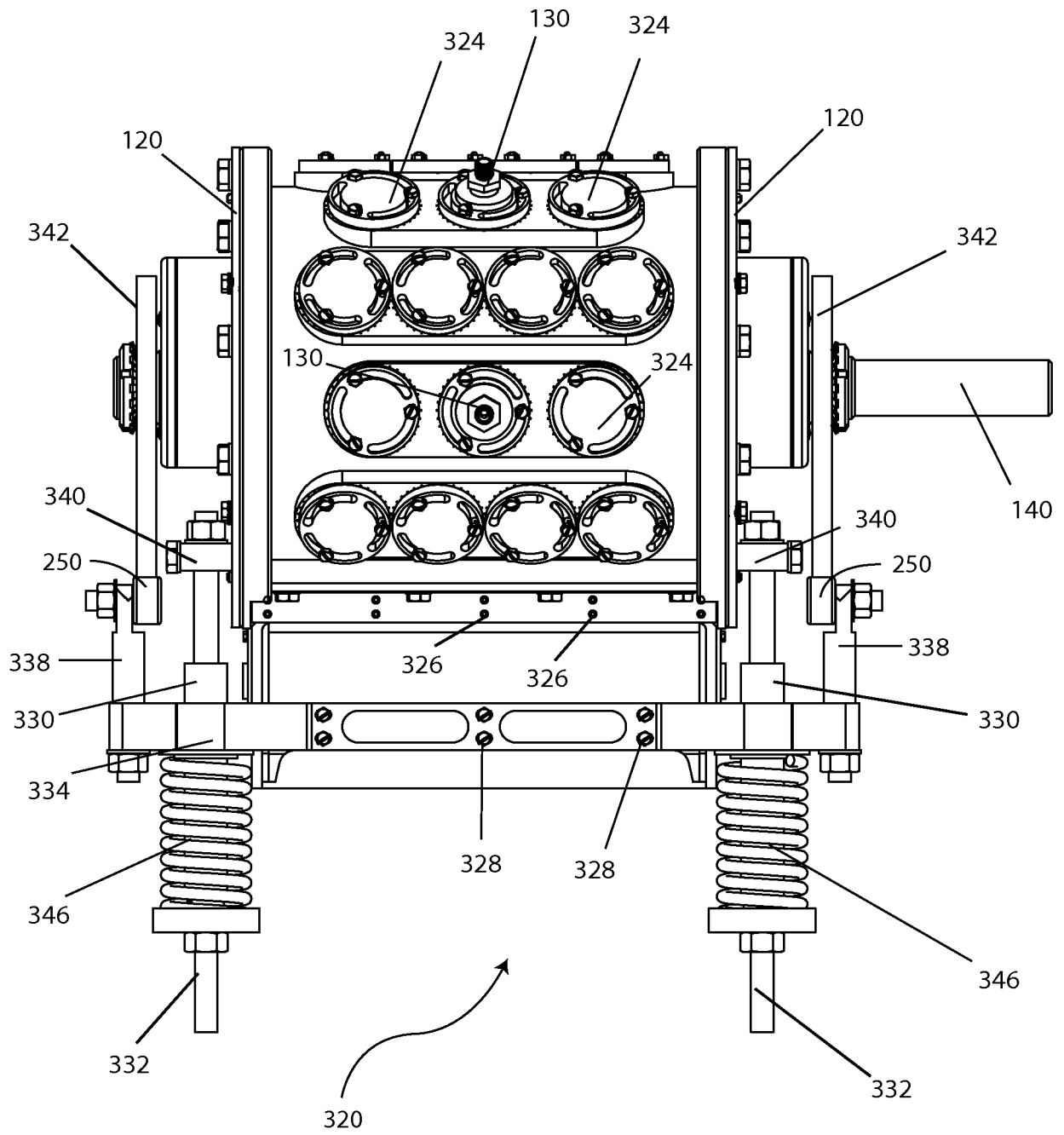


Fig. 21

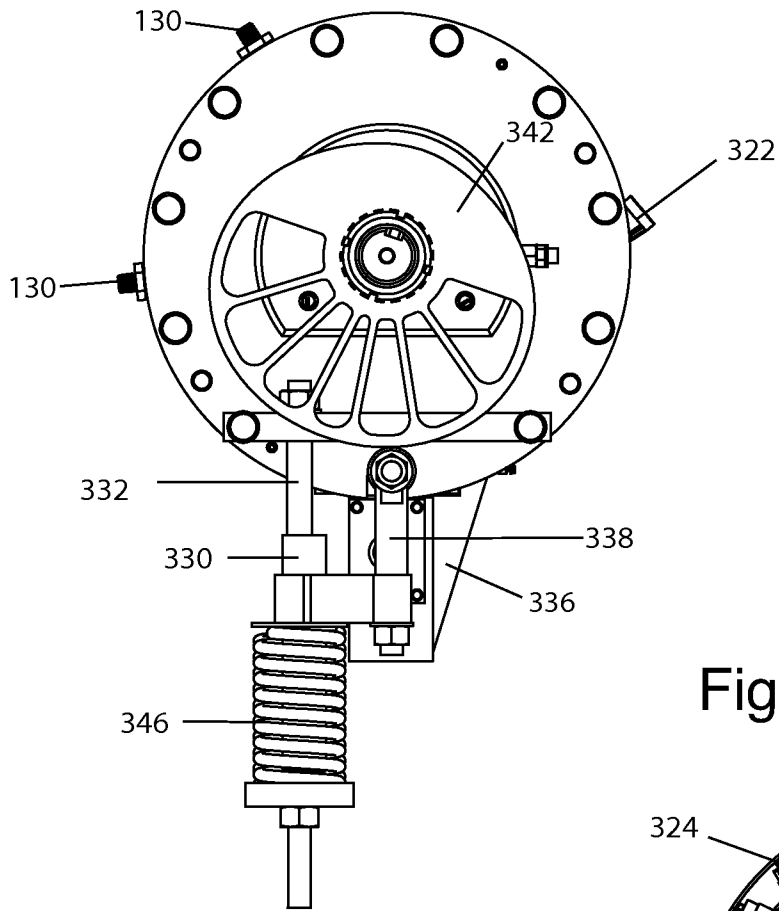
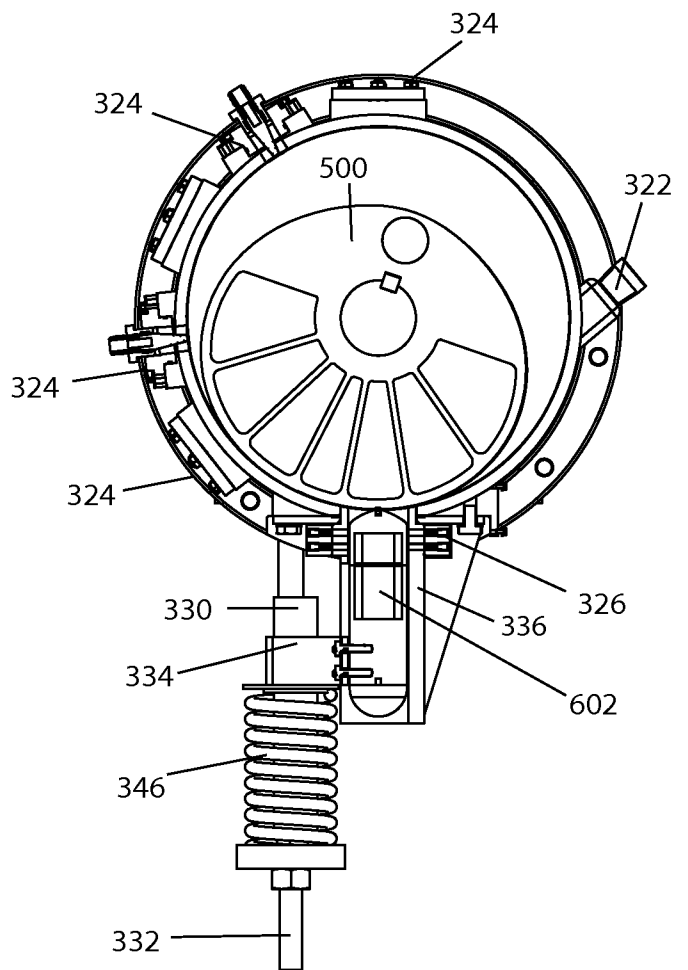
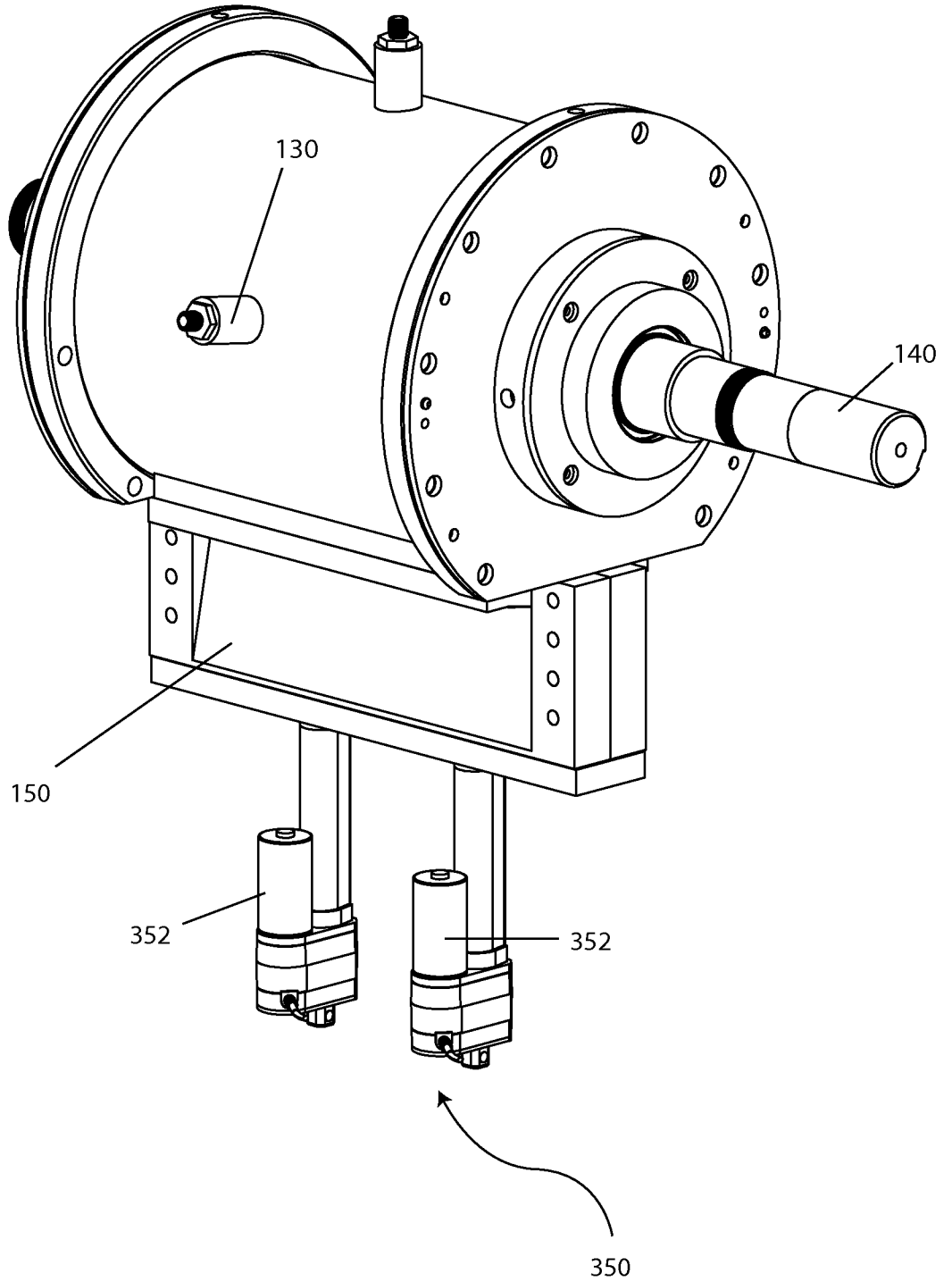


Fig. 22



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Fig. 23



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Fig. 24A

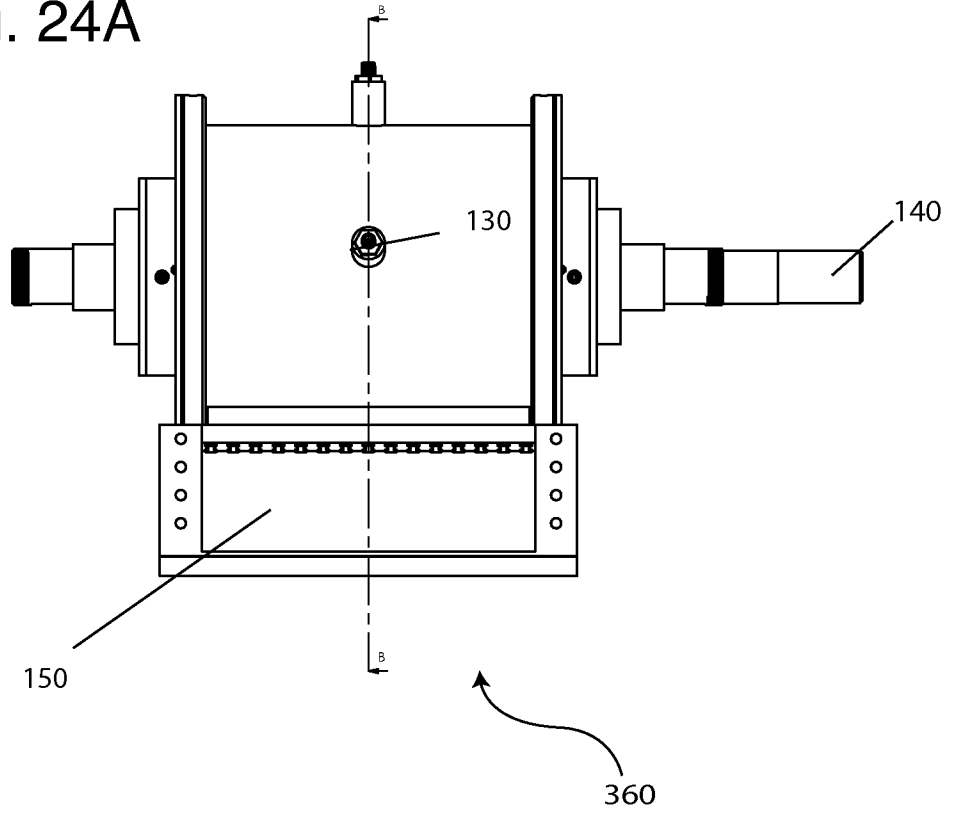
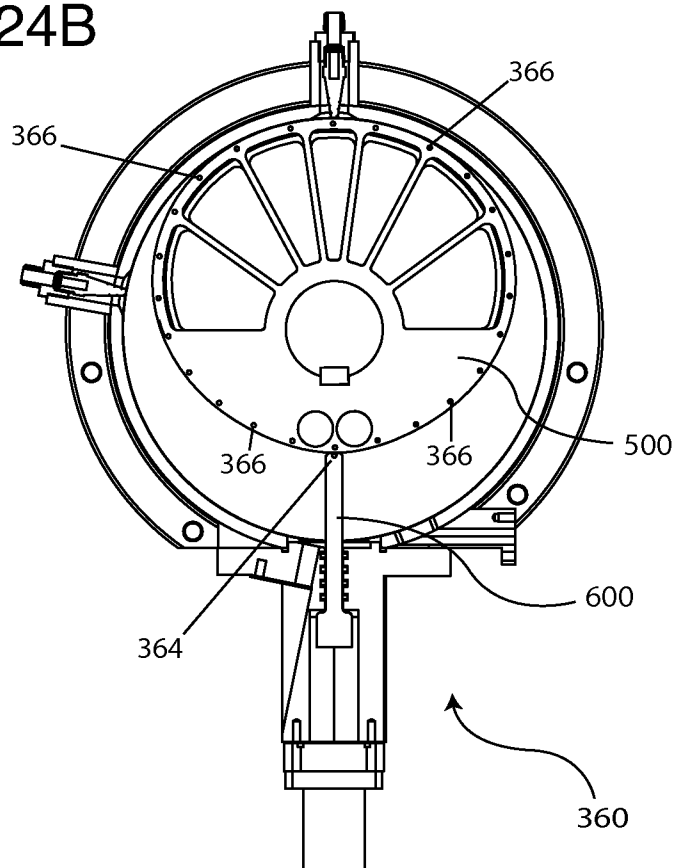


Fig. 24B



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Fig. 25

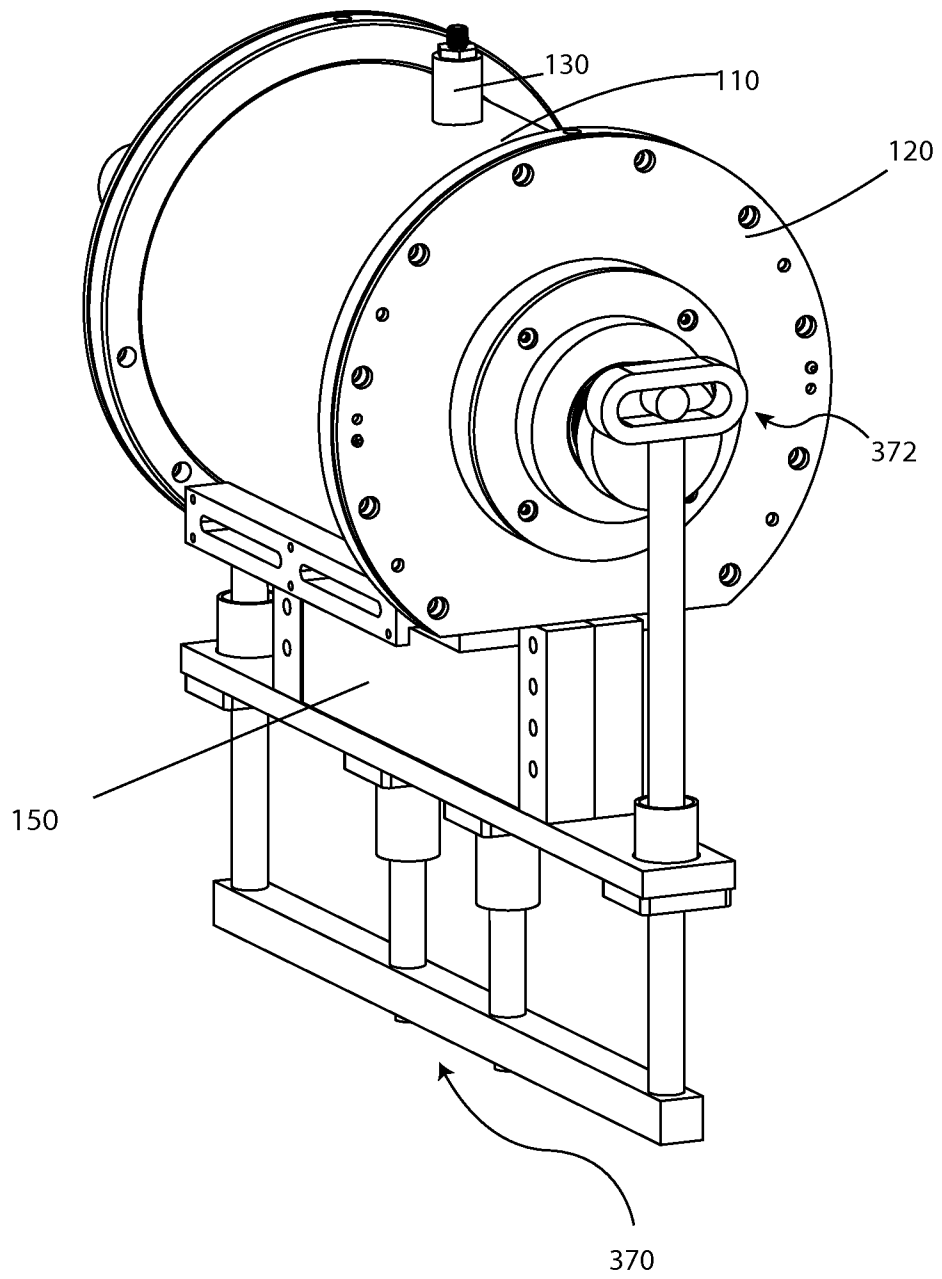


Fig. 26C

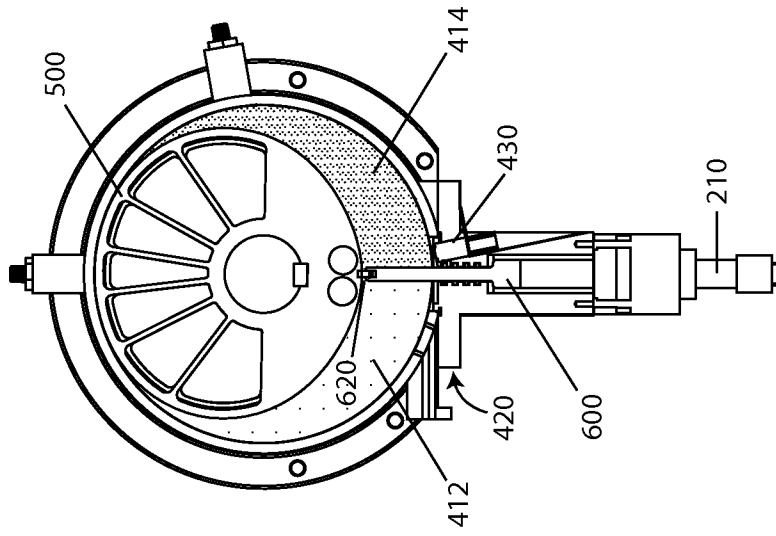


Fig. 26B

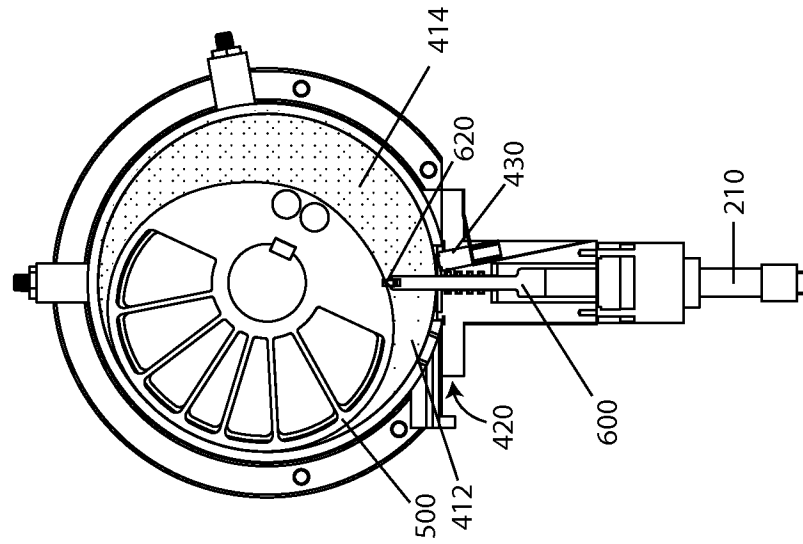


Fig. 26A

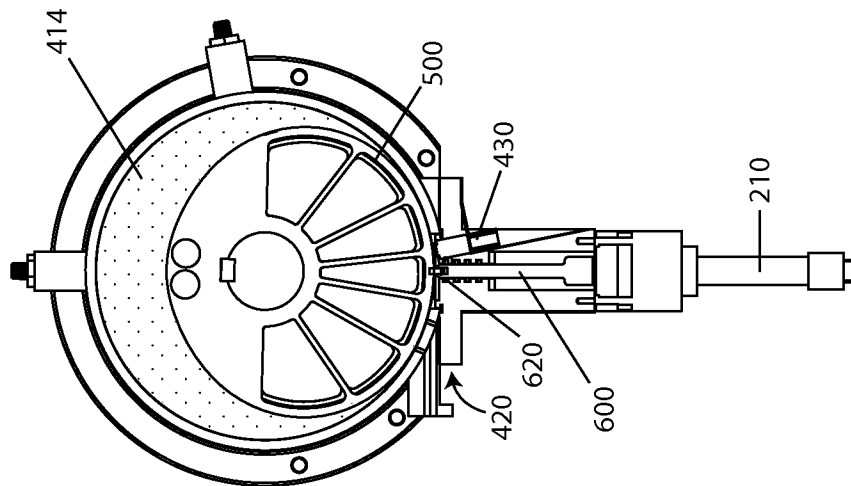


Fig. 26F

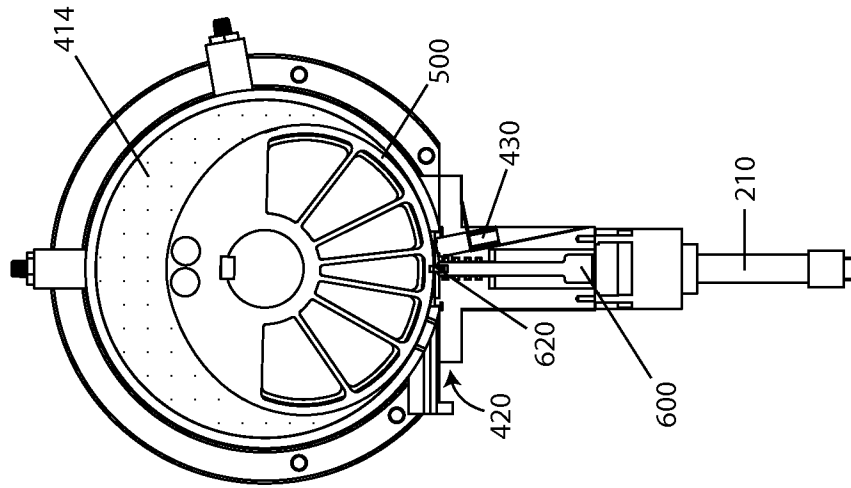


Fig. 26E

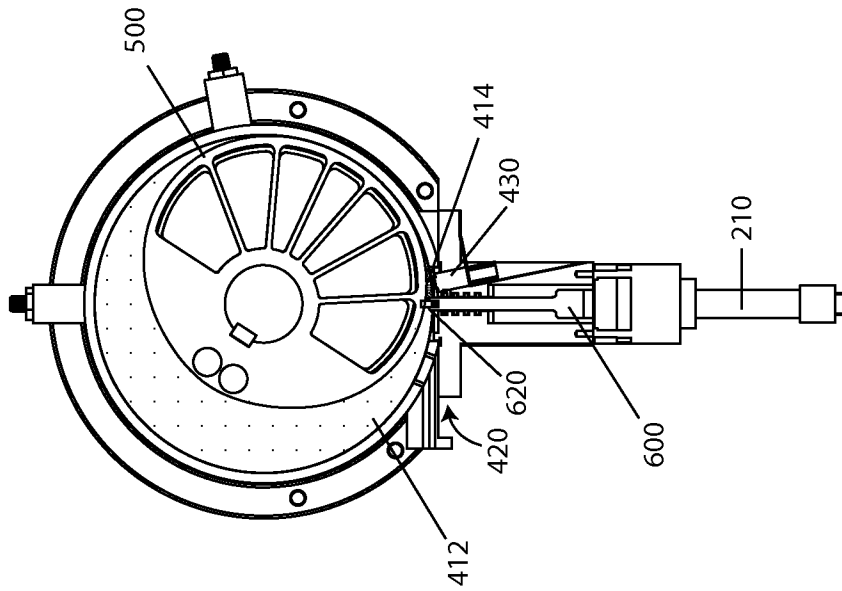


Fig. 26D

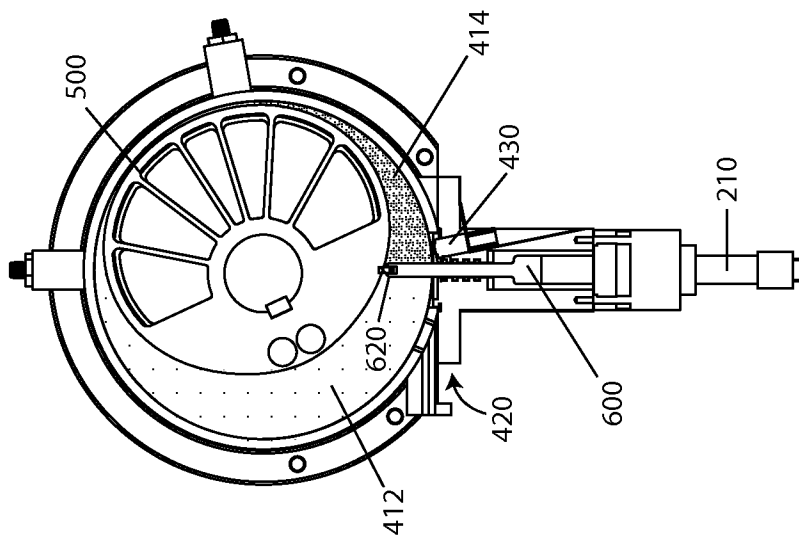


Fig. 27C

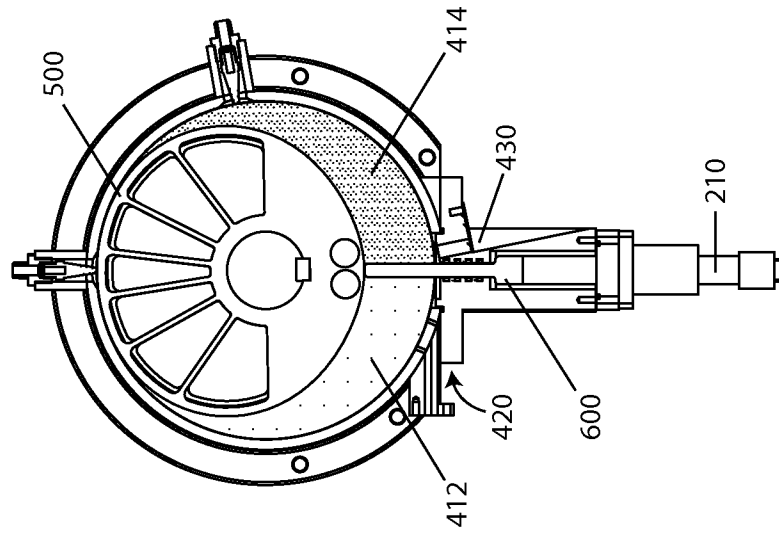


Fig. 27B

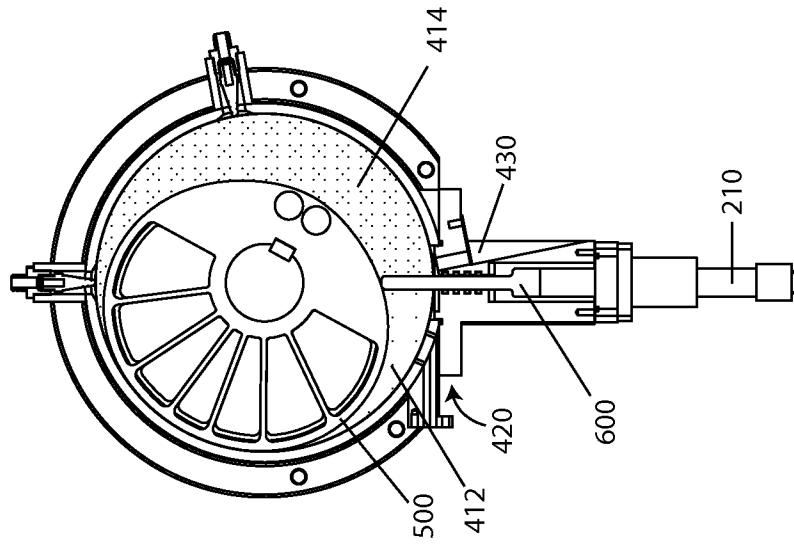
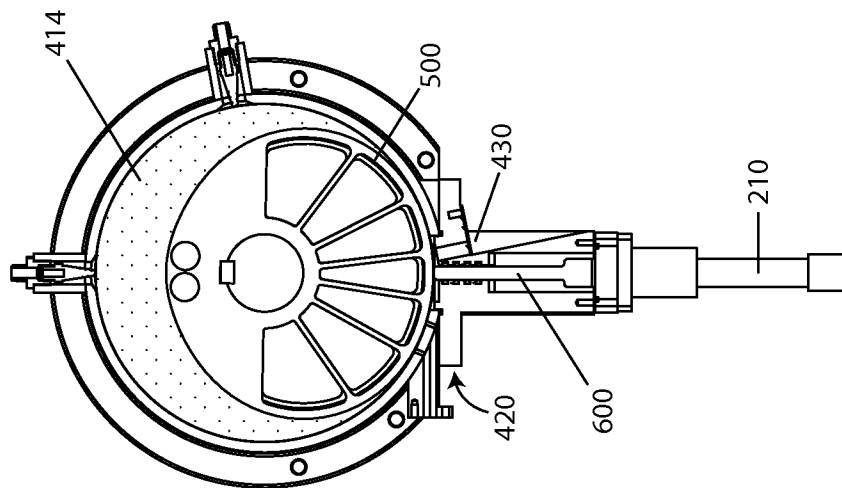


Fig. 27A



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Fig. 27F

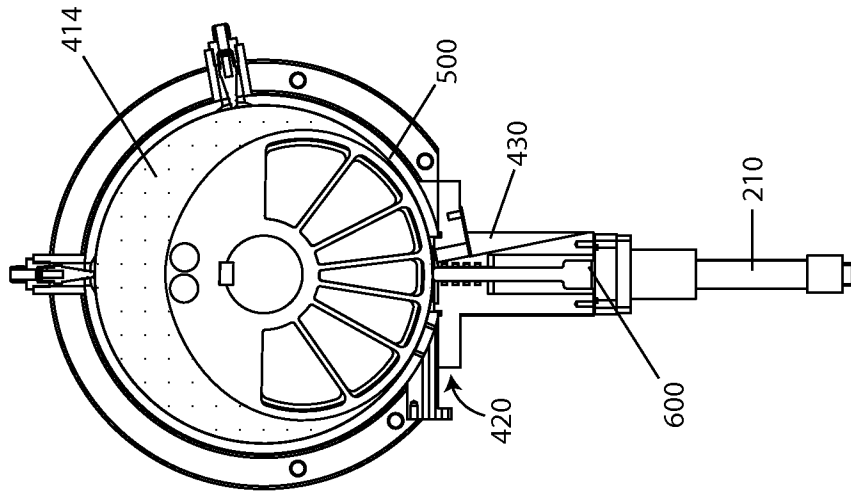


Fig. 27E

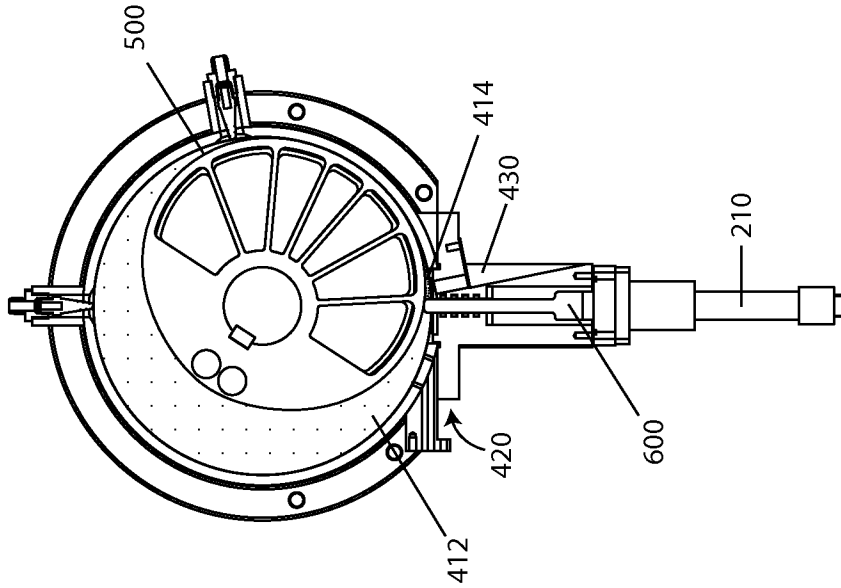
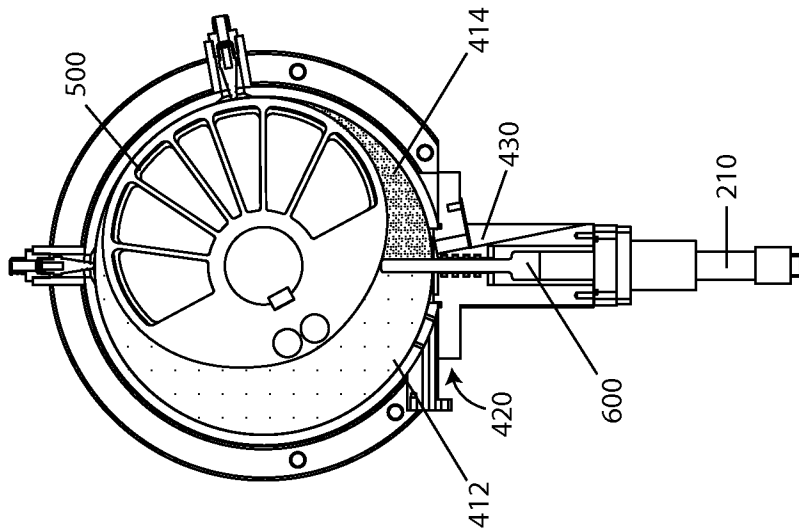
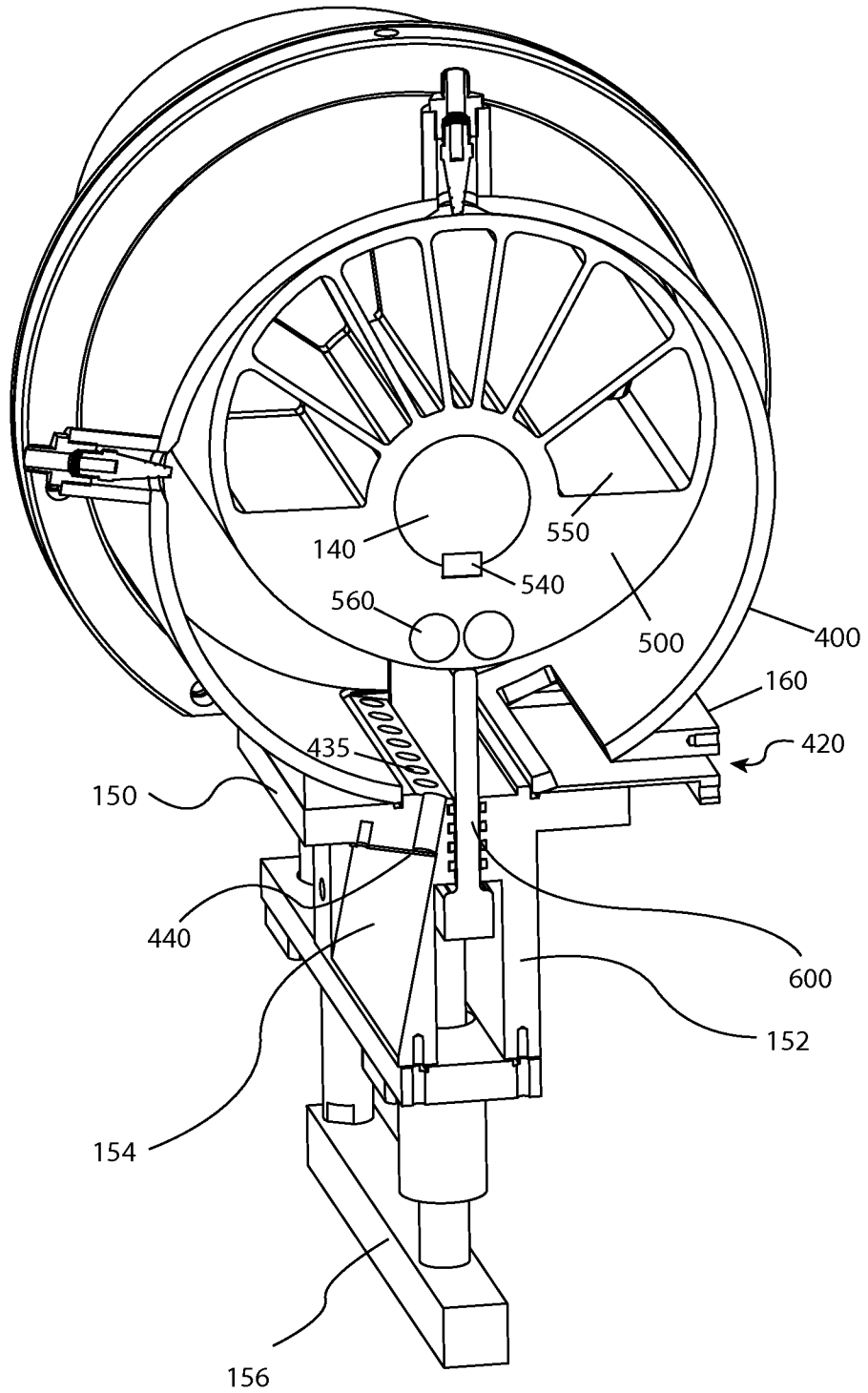


Fig. 27D



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Fig. 28



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Fig. 29

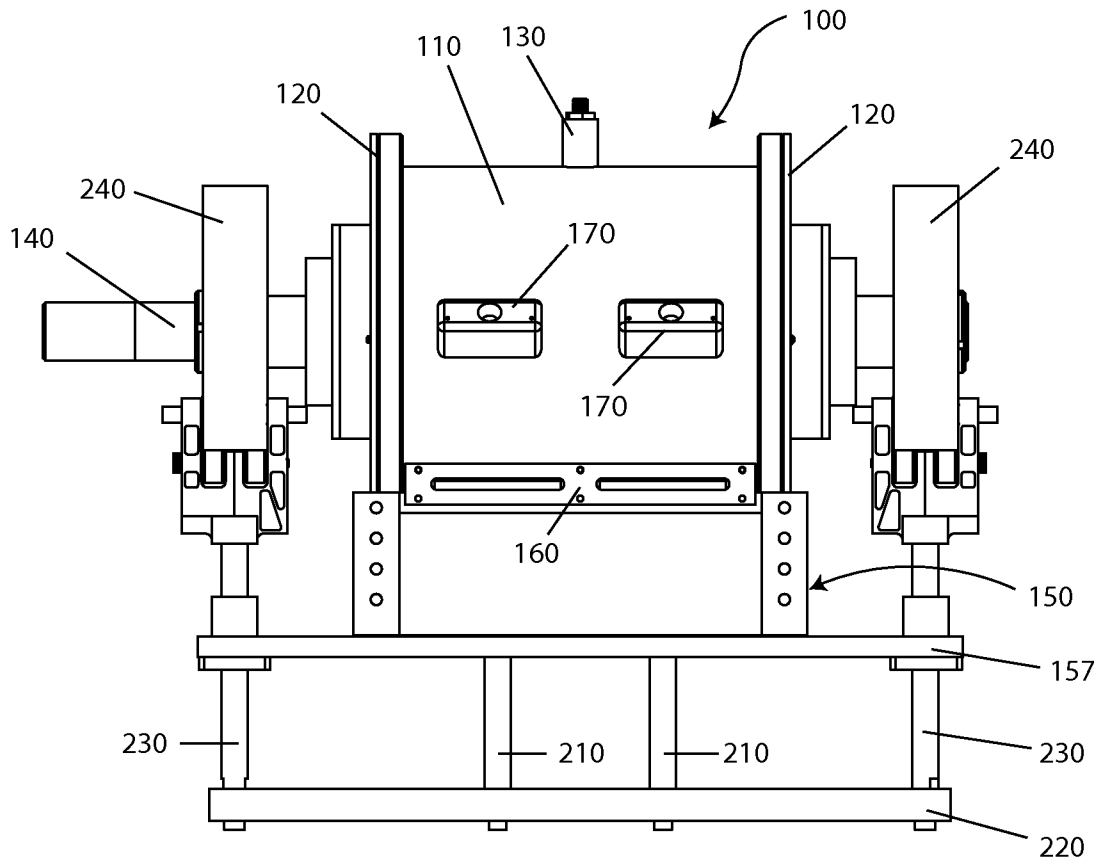


Fig. 30

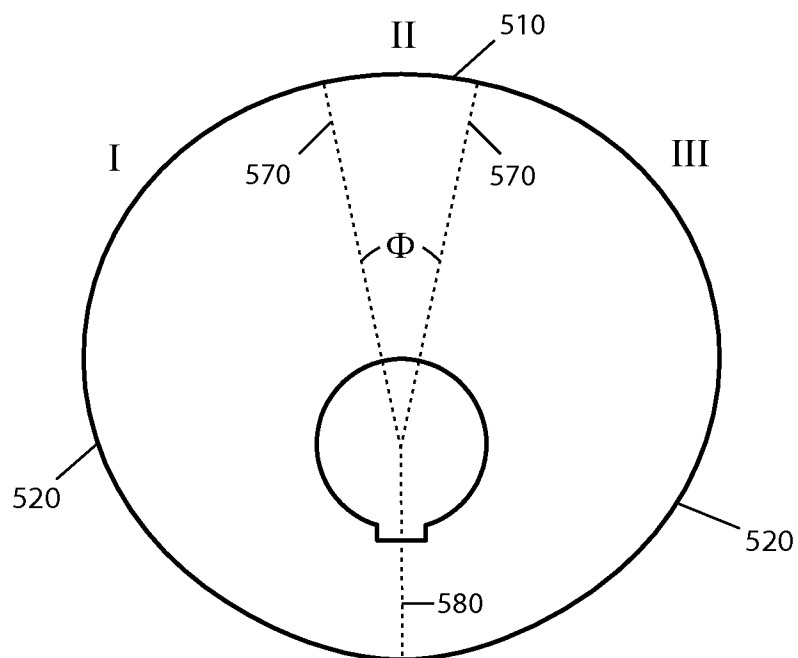


Fig. 31A

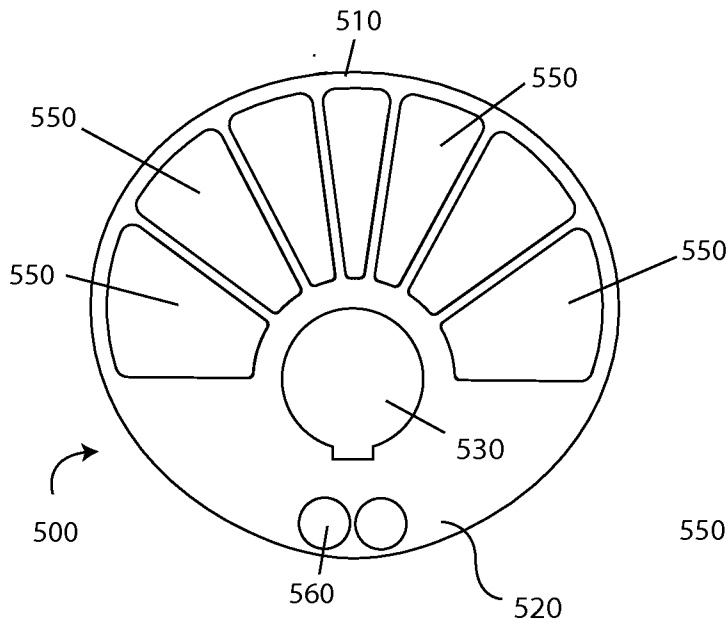


Fig. 31B

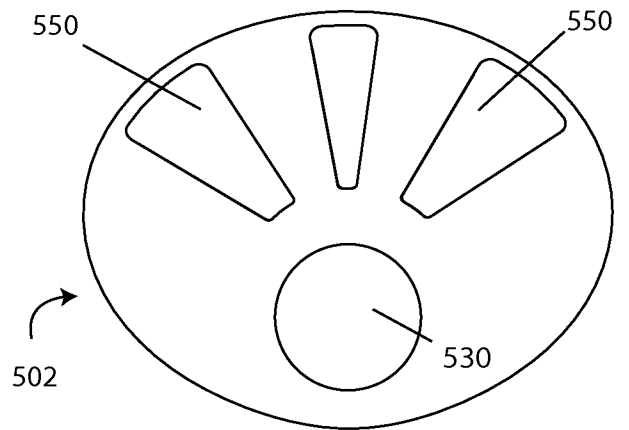


Fig. 31C

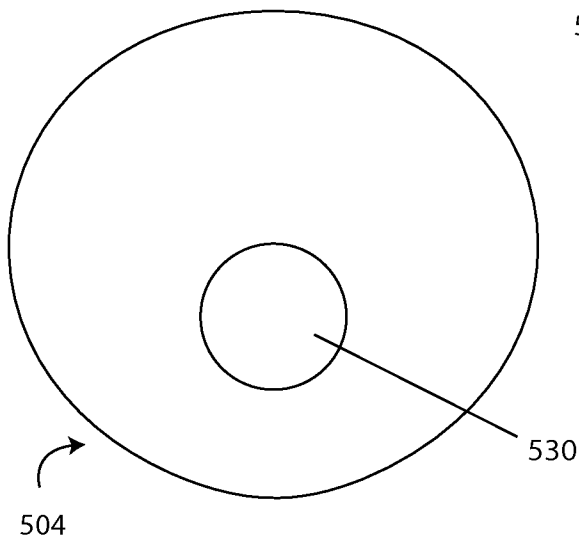
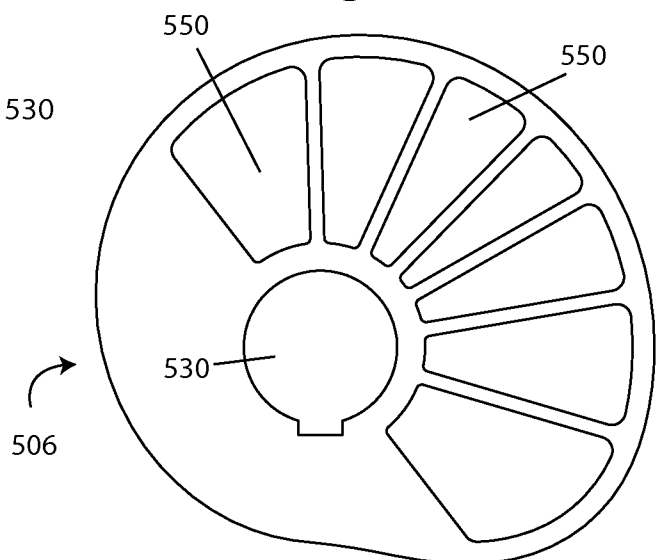


Fig. 31D



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Fig. 32A

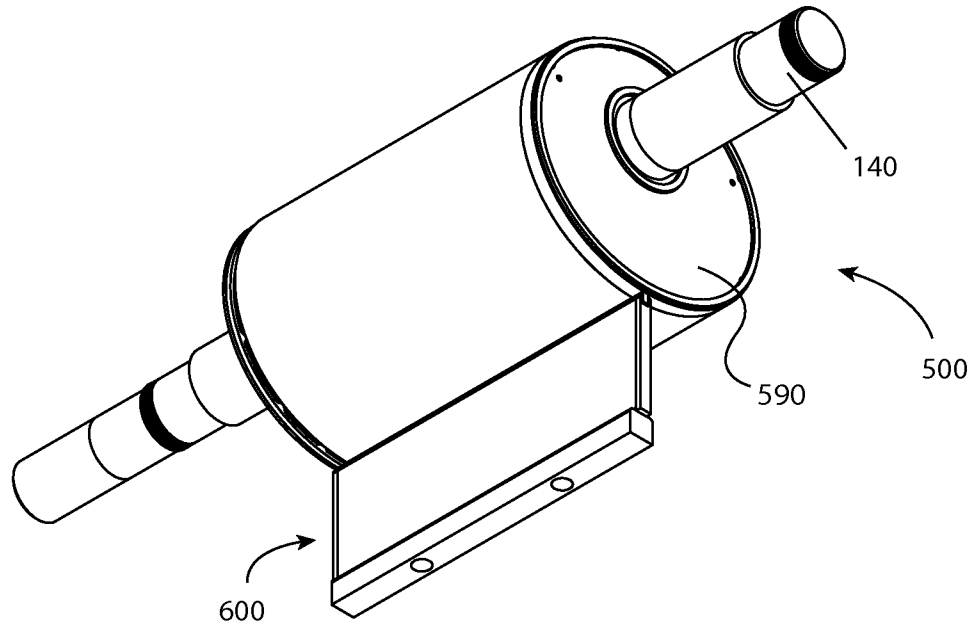
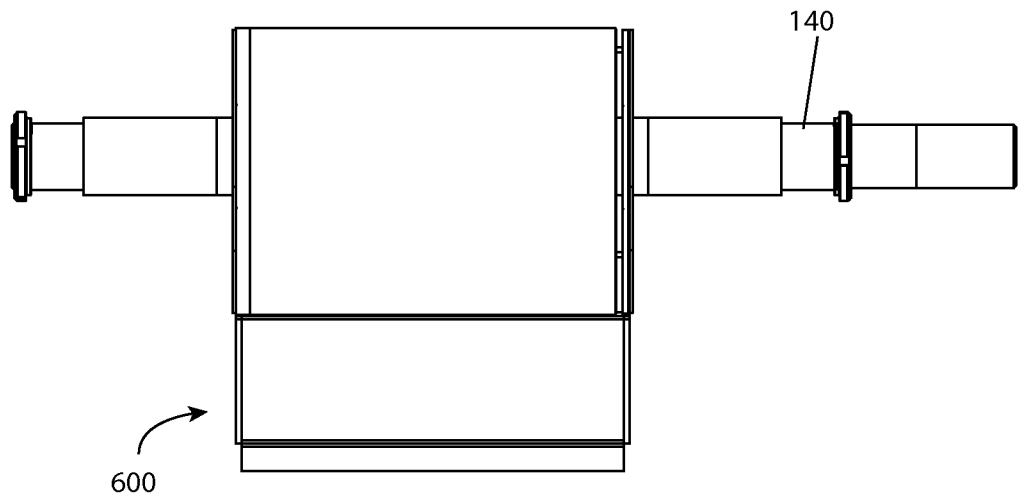
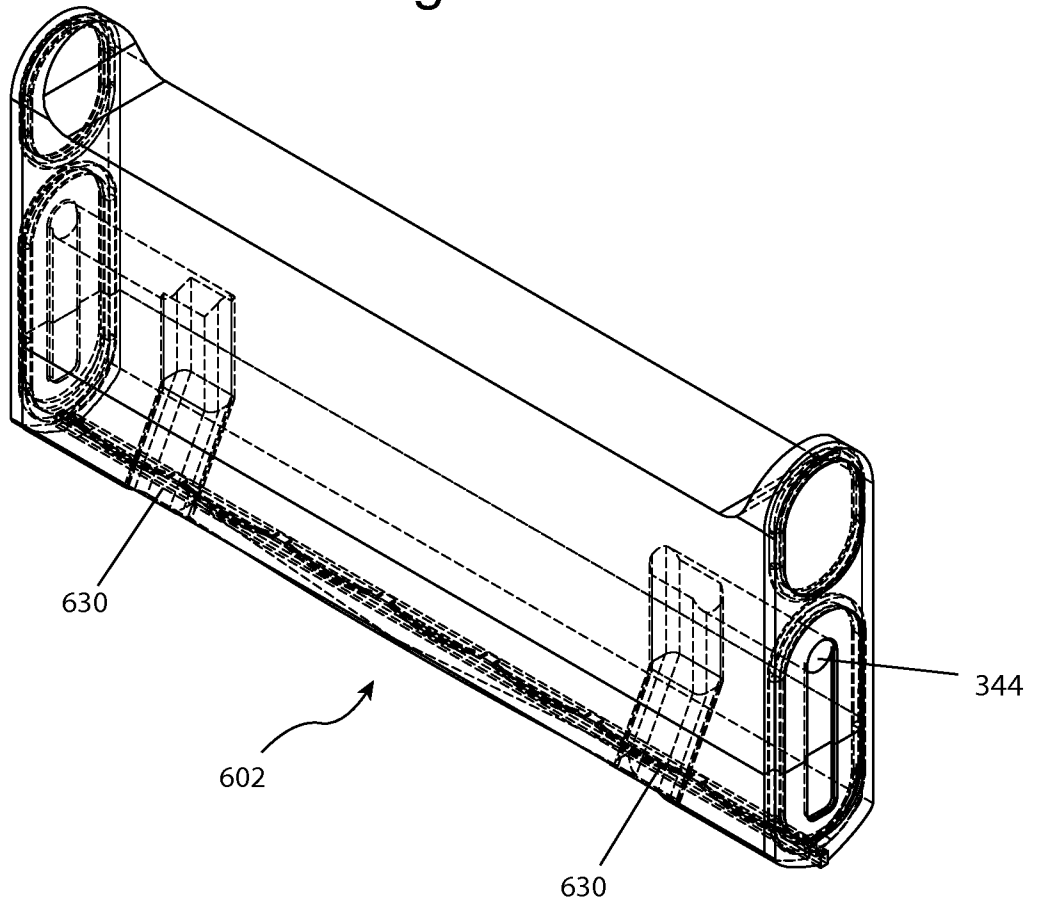


Fig. 32B



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Fig. 33



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Fig. 34A

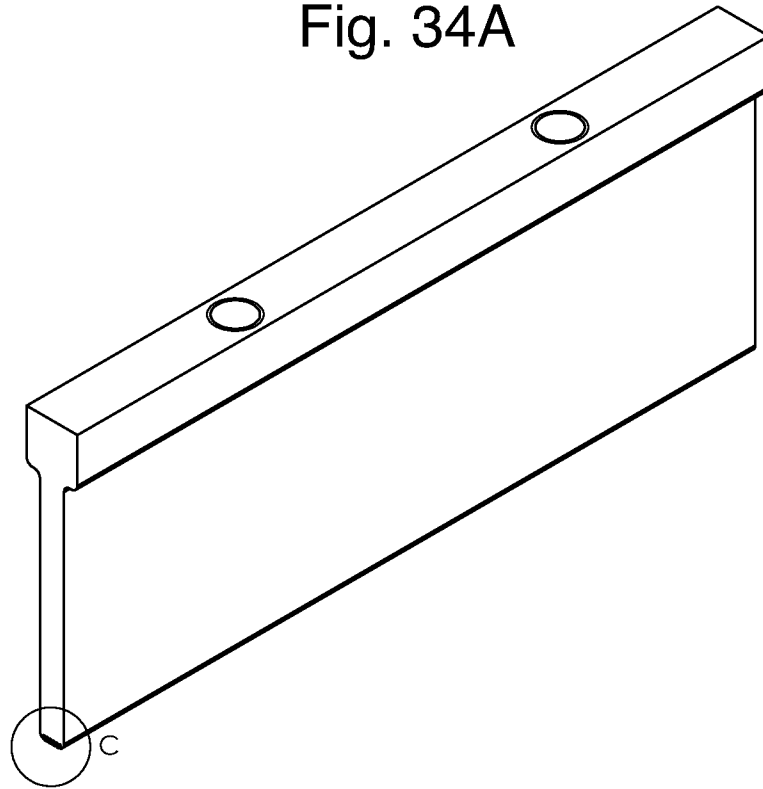


Fig. 34B

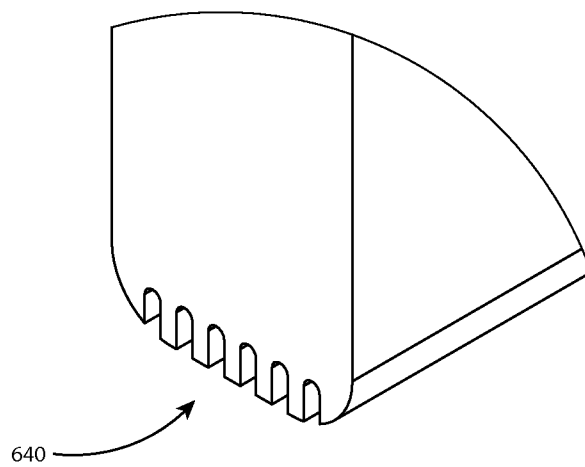


Fig. 35

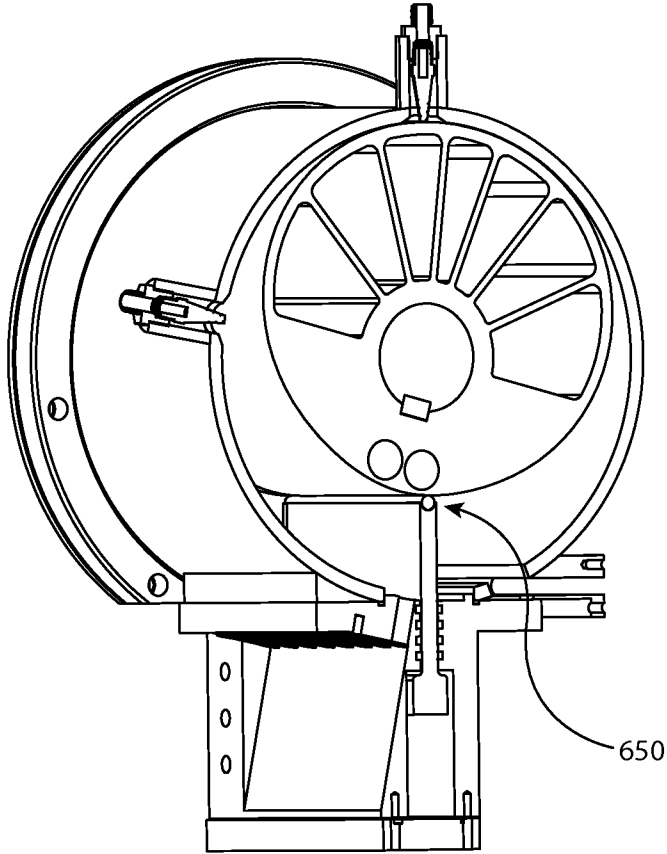


Fig. 36

