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(54) Title: AXIAL PISTON HIGH PRESSURE GAS COMPRESSOR

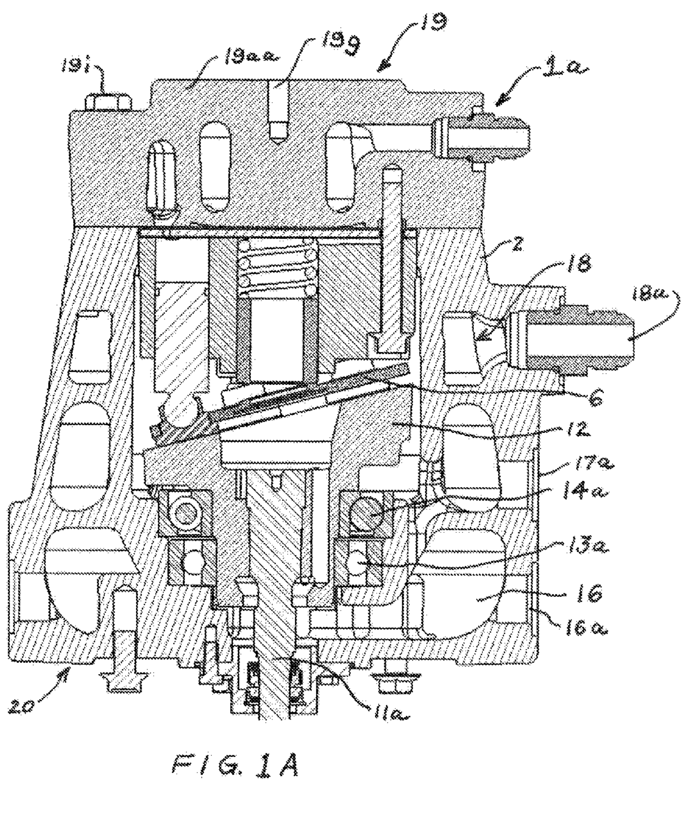

FIG. 1A

(57) Abstract: An axial machine having a wedge that drives a plurality of pistons and a retainer plate for maintaining the slipper shoes in contact with the wedge. A retainer sleeve engages the retainer plate in a continuous rolling engagement as the retainer plate changes its angular orientation with respect to the pistons during operation of the compressor. The retainer sleeve also keeps the retainer plate in position without the need for springs as used in past compressors. The rotating drive shaft has an offset passageway along its shaft to draw oil from the sump along the drive shaft to the wedge to disburse oil on the angular surface of the wedge. A circular valve plate controls the flow of gas into and out of the cylinders.

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TITLE OF THE INVENTION : AXIAL PISTON HIGH PRESSURE GAS COMPRESSOR

INVENTOR : CRAIG W. RIEDIGER

I. CROSS REFERENCE TO RELATED APPLICATION

This application is based on and claims priority of United States provisional patent application 62309531 filed March 17, 2016.

II. FIELD OF THE INVENTION

This invention relates to axial piston machines and more particularly to axial gas compressors that can be operated in the vertical or horizontal position and furthermore have selectable options of an open drive or sealed hermetic drive configurations, all embodied in one oil lubricated axial machine. It is further directed to a gas/liquid separation system in the compressor and axial piston retention ring or plate center positioning means not dependent on a center post but rather centered following the dynamic geometric position of the piston shoes. The invention also discloses an axial piston machine that provides a new oil lubrication system that distributes oil to the machine through the wedge.

III. BACKGROUND AND SUMMARY OF THE INVENTION

Axial piston machines have performed various functions as compressors and pumps and have been driven by electric motors, hydraulic motors, and other mechanical methods in various environments and configurations. The mechanics of the geometry presented by Applicant may be applied to advantage in both pumps and compressors; however preferred embodiments of applicant's invention will hereinafter be primarily descriptive of advantages for compressing working refrigerant fluids in a vapor compression cycle. More specifically the preferred embodiment will be directed to a high-pressure gas compressor for using the natural refrigerant CO_2 as the working gas. There are several primary technology and design hurdles for improving CO_2 gas compressors; these are: (1) higher pressures required, (2) unique properties of CO_2, (3) lubrication of working components due to the first two items, and (4) manufacturability and lack of economy of scale with resultant cost drivers. Importantly, a through shaft would be of great advantage in the flexibility to use a single compressor in many configurations; a boon to achieve economy of scale.

CO_2 is known to be a very effective solvent, and oil tends to dilute in the presence of this gas. Oil loss and dilution results in reduced oil viscosity, and maintaining adequate lubrication film under bearing loads is a crucial consideration for dynamic parts. Ramifications are conditions that affect the durability and operational efficiency of such and similar compressors in a vapor compression system. This is particularly important for thermodynamic cycles of high pressure vapor compression of CO_2 gas refrigerant to a transcritical state to be used for heat pump and/or refrigeration. Separation of the working refrigerant gas from the lubricant (oil) and segregated exclusion from the external vapor compression system circuit and associated components is highly advantageous. A primary reason is because oil (liquid) is known to coat the walls of heat exchangers reducing the heat transfer efficiency of the thermodynamic cycle, and/or oil pooling in undesirable points of a gas circuit which may reduce the oil in the compressor to critically low levels. External means of oil separation and components for the management and return of oil to the compressor do exist for the separation of oil from the working gas. This is conventionally accomplished outside of the compressor in the system; however, it is advantageous for many reasons to separate oil and gas inside the compressor in the process of operation. This neither requires nor precludes the use of external oil management components in a system. The benefits of a low oil output compressor result in downsized or eliminated external oil management system components. It is always advantageous to provide as pure and oil free gas as possible to the system, especially if accomplished economically.

Many conventional compressor designs include an oil sump reservoir and internal pump to assure lubrication accounting for system oil losses while providing an ample oil quantity for circulation to the frictional working components of the machine, as well as establish a return repository for replenishment oil returned back from the system. Most conventional designs allow undesirable and substantial gas mixing enhanced by large internal areas of contact with oil wetted dynamic parts and/or stirred oil foam and/or mist droplets, large surface exposures covered with oil, and free flow into and throughout lubricant containing regions of the machine including the oil reservoir sump area.

It is known in the art that many oil lubricated compressors utilize rolling pistons or crankshaft reciprocating pistons with piston rings or other designs which utilize an internal oil sump reservoir wherein gas/oil exposure and mixing is difficult if not impossible to avoid entirely. This is true of most compressors for applications of the types of cited. For this reason, certain machines (in particular some CO_2 axial, scroll, rolling piston, crankshaft and screw compressors) ignore the issue altogether, allowing prolific oil transport through the entire system counting on the oil entrained gas to lubricate the working parts, and/or require significant and expensive additional oil separation methodology in the system. The downside of this approach is that acceptance of high oil circulation ratio (OCR) dictates acceptance of system and compressor inefficiencies and other undesirable consequences. This cost and design tradeoff may easily lead to related costs of more expensive external oil separation and management components while yielding dubious satisfactory results.

Whether intentional or unintentional, if valve designs are inadequate for liquid refrigerant or oil conditions generated for any reason, liquid through-put manifestations are known to cause detrimental effects to compressors. If excessive liquid transport through valves becomes significant enough, intake valves might oil-can, deform, or fracture, and/or discharge valves might likewise see deformation and/or potential valve backer failure depending on the strength of the backer structure.

Direct piston blow-by gas into oil wetted case areas containing dynamic components results in oil entrainment of the working gas by exposure to oil soaked elements and/or large exposed oil sump region(s). In addition, the route of the intake gas from compressor inlet through to the intake valve should be maintained as oil-free as possible and facilitate oil separation as opposed to enhancing oil entrainment of the working gas. This route is largely overlooked in regard to oil separation and temperature control of the working gas, reducing the ultimate

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efficiency of powering the working gas in and out of the compressor without compromising the lubrication of working parts. An ideal configuration would segregate the intake gas from the internal oil containing and wetted regions of the compressor prior to the intake/compression cycle, and before passing through an intake valve, all the while facilitating the separation of entrained oil in the process. An ideal compressor would accomplish these tasks in either a horizontal or vertical orientation.

Therefore, piston blow-by gas into the compressor's oil bearing regions should be minimized and quickly evacuated limiting undue exposure to the lubricant. In addition, return intake gas should not necessarily be directed directly into or through oil rich internal areas of a compressor as a main gas passage to the intake valve, as this significantly enhances oil entrainment of the gas. Should oil separation methods employed in the external circuit fail, or superheat of the return refrigerant be insufficient, liquid (oil and/or refrigerant) may result in high OCR or liquid slugs to the compressor inlet port. Compressor failure or damage may result as liquid oil and/or liquid refrigerant produces a hydraulic manifestation which effectively does not allow normal gas compression because of the liquid medium state of the compound. The resultant pumped liquid, which is considered incompressible, imparts slamming stress forces to thin reed valve components which may fracture or otherwise deform. For these reasons, it is vastly preferable for internal means of gas/oil segregation, and gas/oil separation, to occur within a compressor machine between inlet gas port and prior to entry through an intake valve into a compression chamber.

Oil lubricated compressors (excluding self lubricated or sealed lubricated components) must rely on any combination of four means to provide liquid lubrication to working frictional contacting parts, and assure that oil is supplied and replenished adequately when in operation. These means are: (1) an oil pump, (2) splash lube, or (3) oil mist circulation designed to supply lubrication to moving parts and (4) immersed running operation. Operational design of oil lubricated compressors is specifically dictated by gravity as a first and primary consideration.

For vertical oriented axially motors, pumps, compressors, and other shaft driven devices, pumping lubricant to high areas from an oil reservoir/sump generally requires an oil pump as a conventional option to oil immersed operation which can be inefficient. Active positive displacement pumps add expense and weight and other design and maintenance logistics burdens. However, it is known in the art that a centrifugal pump requiring no additional parts may be designed starting in the center end of a shaft and line-boring a hole off axis toward the outer diameter and exiting the shaft at some axial distant vertical point. Spinning such a shaft in a vertical orientation is known to centrifugally lift oil from a sump area.

However, this centrifugal pumping concept has not been applied to improved effect in the design of axial piston wedge driven compressors. An axial compressor with its pistons pointed up in a vertical position inherently determines that the driest area of such a wobble plate compressor is the center spacial area above the wedge. The oil sump is below the wedge which in effect "hides" the pistons and slipper shoes and internal piston retention support mechanisms from straightforward splash lube operation. Oil can be distributed around the wedge perimeter but a solution has not been found for splash lubricating the inner surface of the wedge and maintaining a continuous even hydrostatic film while the spinning wedge is flinging oil off of those very surfaces in the opposite direction. Compounding the problem is the fact that the wedge is sloped and therefore oil drains off the surface quickly when stopped.

Relying on splash lubrication routed around and over the top of the wedge cannot be counted on to coat the wedge sufficiently and consistently for dry starting and long term durability. The reasonable conventional alternative would be to lubricate from the shaft. It should be pointed out that using a shaft for centrifugal lubrication requires holes of alignment exiting the shaft with the number of holes and the location of holes and the size of the holes adequately engineered to assure all working parts are lubricated sufficiently. This is a sizeable and sometimes impossible challenge when multiple areas must receive the quantity of oil required at the proper locations but physical limitations of shaft oil distribution prohibit it.

Applicant's invention embodies an improved centrifugal wedge that acts as a pump. The wedge structure allows even distribution of hydrodynamic oil over the wedge surface while providing adequate splash lubrication necessary internally above the wedge lubricating the innermost frictional mechanisms. These are the innermost bottom piston surfaces as well as the combined retainer mechanisms. The wedge simultaneously lifts and splashes the oil on its outer perimeter, splash lubricating the wedge and piston bottom outer surfaces.

Gravity induced bearing loads and oil pooling must be considered. The determinations of the internal/external compressor structures and component orientations, horizontal or vertical conventionally yields a final configuration to be used only in its singular design orientation. The packaging of final equipment largely depends on the selected compressor and its physical dimensions.

As an example, hermetic scroll compressors are known to have a relatively small cylindrical footprint requiring a vertical orientation. This condensed footprint with a taller profile defines a vertical minimum space limit for installed equipment which is determined largely by the length of the motor and compressor vertically stacked and coupled within one hermetic "shell". However, the scroll compressor will not function properly or fail if operated in a horizontal position. The upshot is that taller compressors with vertical orientations may not qualify for use in the design of limited headroom, low profile packaged equipment designed for tight vertical spaces such as interstitial spaces such as above ceilings. Conversely, taller compressors such as scroll compressors are conducive for application in equipment designed where a small footprint is desired and floor space is premium and vertical space is adequate.

As a converse example, crankshaft reciprocating piston compressors conventionally couple with an electric motor in a horizontal orientation and usually employ oil sumps often with oil pumps or splash lube methods providing lubrication to frictional components. This orientation is better suited for low profile, larger footprint equipment packages. Many examples of differing compressors have been designed for use exclusively in either a horizontal, or conversely a vertical operational orientation. In short, an oil lubricated gas compressor is needed that will operate in either a horizontal or vertical application without compromising lubrication of the machine and improve the segregation and separation of lube oil and the working refrigerant.

As has been inferred above, the type of drive system with the consideration of refrigerant containment is extremely important, and even critical for most applications, especially those using CO_2. Driving a compressor with an open mechanical shaft such as provided by a combustion engine is not easily accomplished as a totally sealed hermetic machine. This would require the engine to be included in a single sealed case, or shell, along with the compressor. Therefore, practicality requires a compressor open drive shaft with an adequate rotary shaft seal for sealing the working gas within the compressor. For a horizontal orientation, generally this would most easily be achieved with the compressor interior lubrication system arranged so as to function properly in this exclusive arrangement.

In a vertical orientation, a motor or engine may be theoretically employed above or below the compressor. However, many conventional piston compressors have a high-pressure head and/or intake and exhaust manifolds adjacent to and blocking piston cylinders. In conventional axial piston machines, this head/manifold region with precision internal valve components does not allow a thru-shaft penetration. The conventional design of a vertically mounted axial compressor which is to be used above the drive motor or engine is commonly dictated largely by the compressor head and manifolding in relation to the driving end of the compressor drive shaft. The compressor being the highest uppermost component allows the pistons and valving to be above any oil reservoir, and the compressor drive shaft usually points downward for connection to a prime mover. This is quite acceptable for use with an open drive compressor requiring but few compromises, and further allows an oil sump in the compressor exclusive of and outside of the driving mechanics of the compressor. This of course assumes a compressor rotary shaft seal of adequate design to hold the gas pressures and withstand the temperatures and mechanically generated seal friction caused by shaft rotation and sealing elements. A hermetic or semi-hermetic (fully sealed) application with the motor below the compressor would eliminate the rotary shaft seal requirement because gas is sealed within the entire compressor/motor assembly case or shell. However, now the oil within the compressor seeks the lowest point which is the interior motor shell housing the stator/rotor components. Dry motor operation would require an oil seal (internal) between the compressor/motor which might be expected or designed to leak, thus requiring an oil return motor/sump oil scavenge method up to a separate compressor sump. A hermetic or semi-hermetic immersed oil motor configuration might be an alternate consideration for a bottom mounted motor; however, several technological and cost hurdles exist for efficient operation. To summarize the described vertical configuration: a bottom mounted electric motor coupled with a top mounted compressor which are both contained within a single hermetic or semi-hermetic shell is not a simplistic configuration from a design point of view. The exception for this orientation is an open drive compressor above the motor wherein the gas and oil is contained within the compressor mounted above, and the motor is free to operate normally in ambient conditions below.

An alternative embodiment of vertical orientation of an axial compressor/motor combination is the compressor on the bottom with the motor above the compressor. However, and as previously referred, the head arrangement of most compressors does not allow thru penetration of a drive shaft. Therefore, to consider this orientation, most axial compressors must necessarily be inverted orienting the high-pressure head, manifold, and piston valving located at the bottom of the stacked arrangement. The compressor drive shaft usually exits the opposing end of the compressor case and would point up to couple with the motor. In this position, during compressor idle/off conditions there is a risk that oil migration will leak past the pistons collecting and pooling in valve areas and manifolds. After idle shut-down, compressor start up risks damage to sensitive valving due to oil liquid slugging and hydraulic canning caused by oil which has settled in undesirable areas. Even if this oil is expelled from these regions without incident, the effect is likely to require collection and return of this oil to the compressor. To add to this hurdle, there is no compact or simple provision for an oil sump located within the compressor at the lowest point, inferring an oil flooded compressor arrangement; not an ideal scenario.

To summarize, there is a need for an improved oil lubricated compressor that allows thru-shaft access at either or both the high-pressure head/manifold area and the lower (sump) area of the compressor for use in a vertical orientation. In addition, the same compressor could be used in a horizontal orientation. The same compressor would be capable of coupling with various open drive and hermetic drive configurations including double-ended drive shaft for stacking and plural arrangements of compressors, motors, gas expansion engines or other embodiments. Combined elements of Applicant's invention provide improvements of compressor performance, durability, and cost because of compressor design improvements, packaging flexibility, manufacturability, and economy of scale.

Another problem addressed by Applicant's invention is related to the means for center positioning the piston retainer plate in an axial machine. In the past, a fixed position spherical ball nose segment, post, and spring assembly imparts both necessary force to counteract suction piston forces as well as fixing a centering position of the wobbling piston retaining plate. Centering a piston retainer plate in this way is intended to prevent radial misalignment assuring that wobbling piston slipper skirts do not interfere with respective but oversize retainer bore holes in the retainer plate. Applicant's invention provides a means of centering the retainer plate using the dynamic geometric position of the piston slipper shoes and slipper skirts on the wedge face plane as a centering mechanism.

Applicant's invention provides a means for allowing vertical or horizontal operation and selectable options of preferred open drive or sealed hermetic drive configurations, all embodied in a single oil lubricated axial machine. The preferred embodiment illustrates an axial wobble-plate multi-cylinder compressor allowing either horizontal or vertical orientation incorporating combined improvements including but not limited to, means of: superior lubrication oil/gas segregation in vertical or horizontal orientation, oil/gas segregation/separation in either orientation, oil distribution to frictional surfaces in either orientation, through shaft and load bearing allowing vertical or horizontal stacking and plural arrangements of compressors/motors, flexible adaption accepting open drive or hermetic drive configurations, and a new means for centering the piston retainer plate using the piston slipper shoes and slipper skirts.

III. BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a left side cross sectional view of the inventive compressor with the shaft extending through the head of the compressor.

Fig. 1A is a left side cross sectional view of an alternate embodiment of the inventive compressor in which the shaft does not extend through the head of the compressor.

Fig. 2A is a top plan view of the cylinder block

Fig. 2B is an enlarged view of the encircled area A of Fig. 2A

Fig. 3A is a top plan view of the retainer plate.

Fig. 3B is a cross sectional view taken along line 3B-3B of the retainer plate of Fig. 3A.

Fig. 3C is an end view of the retainer plate of Fig. 3A.

Fig. 3D is an enlarged sectional view of detail area B encircled in Fig. 3B

Fig. 4A is a top plan view of the retainer sleeve.

Fig. 4B is front elevation view of the retainer sleeve of Fig. 4A.

Fig. 4C is a cross sectional view taken along line A-A of Fig. 4A of the retainer sleeve.

Fig. 4D is an enlarged view of the detail area B encircled in Fig. 4C illustrating the retainer sleeve nose.

Fig. 5A is a top plan view of the assembled retainer plate and retainer sleeve.

Fig. 5B is a front elevation view of the assembled retainer plate and retainer sleeve of Fig. 5A.

Fig. 5C is a cross sectional view taken along line 5C-5C of Fig. 5A.

Fig. 6 A is an illustrative plan view taken perpendicular to the sloped plane of an axial piston retainer plate illustrating conventional center positioning methods in which retainer plate bore hole clearances allows function with orbiting piston slipper skirts.

Fig. 6 AA is an illustrative plan view taken perpendicular to the sloped plane of the axial piston retainer plate showing a novel positioning method which uses contact of orbiting piston slipper skirts within their respective retainer plate bore holes.

Fig. 6B is an illustrative plan view taken perpendicular to the sloped plane of the axial piston retainer plate showing the engineered location and sizing of piston slipper skirts and retainer plate bore holes which enable the novel retainer plate positioning method.

FIG. 6C is an illustrative plan view taken perpendicular to the sloped plane of the axial piston retainer plate showing piston slipper skirt and retainer plate bore hole positioning in (3) example wedge slope directions.

Fig. 7A is a front elevation view of the wedge.

Fig. 7B is side elevation view of the wedge.

Fig. 7C is a cross sectional view taken along line 7C-7C of Fig. 7B.

Fig. 7D is a cross section view taken along line 7D-7D of Fig. 7C.

Fig. 8A is a left side elevation view of the compressor housing.

Fig. 8B is a front elevation view of the compressor housing.

Fig. 8C is a top plan view of the compressor housing.

Fig. 9A is a bottom view of the head with front elevation to the right

Fig. 9B is an elevation cross sectional view taken along line 9B-9B of Fig. 9A.

Fig. 9C is an enlarged view of the encircled area C in Fig. 9B.

Fig. 9D is an enlarged view of the encircled area D in Fig. 9C.

Fig. 9E is a front elevation view of the head.

Fig. 9F is a top view of the head in Fig 9E.

Fig. 9G is a plan cross sectional view taken across line 9G-9G of Fig. 9E.

Fig. 10A is a top plan view of the port plate.

Fig. 10B is an end view of the port plate in Fig. 10A

Fig. 11A is a top plan view of the suction reed valve plate and the suction valves.

Fig. 11B is an enlarged view of the encircled area A in Fig. 11A illustrating the suction valve configuration.

Fig. 11C is an end view of suction valve plate in Fig.11A.

Fig. 12A a top plan view of the discharge reed valve plate configuration with discharge valves.

Fig. 12B is an end view of discharge valve in Fig. 12A.

Fig. 12 C is a plan view of the valve assembly looking at the suction valve stacked upon the port plate stacked upon the discharge valve

Fig. 13 is a perspective sectional view of the housing gas and oil separation configuration with function notes.

Fig. 14A illustrates a vertical open drive with a dry (no oil) bottom motor operation.

Fig. 14B illustrates a vertical hermetic drive with bottom motor oil (wet) or dry operation.

Fig. 14C illustrates a vertical open drive with dry (no oil) top mount motor operation.

Fig. 14D illustrates a vertical hermetic center mount upsized single motor deployed with top and/or bottom mounted compressor and alternate machine.

Fig. 14E illustrates vertical hermetic center mounted dual compressors with top and/or bottom mounted motors and/or alternate machine(s), which also allows convenient compressor staging.

Fig. 14F illustrates horizontal open drive with dry (no oil) motor operation.

Fig 14G illustrates horizontal hermetic drive with center mounted dual compressors and downsized end-mounted motors.

Fig. 14H illustrates horizontal hermetic drive with center mount upsized single motor operation. for multiple compressor operation, which also allows convenient compressor staging.

Fig. 15 is an external isometric view of an alternate embodiment of a semi-hermetic compressor.

Fig. 16 is a cross sectional view of the alternate embodiment of Fig. 15.

Fig. 17 is an enlarged cross section view of the retainer sleeve engaging the retainer plate to push the piston-shoe assembly down during the suction stroke.

Fig. 18 is a further enlarged cross section view through valves during suction stroke.

Fig. 19 is a similar cross section view through valves during discharge stroke.

Fig. 20 is a horizontal cross section view through the suction port to illustrate liquid separation from suction gas.

Fig. 21 is a perspective view of the ring-shaped discharge valve also called the omni-ring discharge valve.

IV. DETAILED DESCRIPTION OF THE FIRST EMBODIMENT

Turning first to Fig. 1, there is illustrated an inventive axial piston machine which can be a compressor, pump or engine but for simplicity will be referred to herein as a gas compressor 1. Fig. 1 shows interior structure and components of compressor 1 which has a case or housing 2, a head end 19, and a base mount end 20. Within the housing structure 2 is a cylinder block 3 which contains at least 3 pistons/ring assemblies (piston 4) in respective piston cylinder bores 3a. Cylinder block 3 is a sealed fit to, or may be integral with, housing 2 located in central housing cavity 2a. The preferred embodiment shows a sealed fit of cylinder block 3 to housing 2 and assembled as shown bolted securely with cylinder bolts 9 and cylinder holes 9a in a sealed arrangement of conventional means with head 19a. Each piston 4 incorporates piston ball 4a which is partially encompassed (ball swage fit) by a slipper skirt 5a of piston slipper shoe 5 which are illustrated as one-piece part. Shown as a conventional piston and shoe ball/socket swaged arrangement, it should be noted that in an alternate and reversed configuration the ball might be integral with the slipper shoe and the socket within the piston base. Rotating wedge 12 is affixed to, or alternatively integral as one piece with, shaft 11 which is driven by a rotational force supplied by an electric or hydraulic motor or other mechanical means such as illustrated in Figs. 14A-14H. Wobble plate or wedge 12 presents a smooth, flat low friction angled face surface which piston slipper shoe 5 must follow in a sliding function wobbling about piston ball 4a as wedge 12 rotates. Mechanical rotation upstrokes piston 4 from bottom dead center within its respective bore 3a performing a pumping or compression stroke of a fluid through 180° rotation to top dead center. At head end 19, discharge and suction valving is accomplished by valve assembly 10.

Shaft 11 extends into or through head 19a centered radially at head end 19 by bearing 22 while rotating wedge 12 is supported radially and axially by bearings 13 and 14 which may be combined as one bearing assembly. Shaft 11 may be driven or drive from either the base end 20 or the head end 19 as illustrated by using an open drive coupling method 38 and protective shroud 40, separate but attachable to shaft 11 and base end 20 respectively. Alternative integrated motor/compressor sealed hermetic embodiment does not require a shaft seal between the motor and compressor. Double-ended shaft open drive method using seals 15 and 15a, illustrate seal cartridges for sealing shaft 11 at housing base end 20 and head end 19.

Fig. 1A shows an alternate embodiment of a compressor 1, which shows the versatility of the invention for economy of scale cost reduction. Rotating wedge 12 is supported radially and axially exclusively by engineered bearings 13a and 14a which may be combined as one bearing assembly allowing a shortened, single ended shaft 11a which does not extend into or through head end 19. Engineered bearing support means does not require an opposing end of shaft 11a to be incorporated or otherwise supported. Head19aa (Fig 1A) illustrates a lifting ring pilot hole 19g in lieu of head 19 (Fig 1) bore hole machining for shaft 11 and seal 15a accommodations. Head bolt holes 19h and head bolts 19i thread into housing 2 at 19k (Fig. 8C). Either machined configuration of head 19a or 19aa may be used with compressor embodiment 1A without compromising performance. This describes the primary difference in embodiments of compressor 1 and compressor 1a in Fig, 1 and Fig. 1A respectively; the following description will refer to compressor operation as compressor 1/1a unless otherwise specified.

Retainer plate 6 of applicant's invention Fig.1/1a counteracts piston 4 inertial and suction downstroke forces by applying equal or higher force to piston slipper shoes 5. This force is exerted by retainer sleeve 7 backed by retainer sleeve spring 8. Retainer plate 6 assures piston slipper shoe 5 fully and evenly contacts wedge 12 completely through suction downstroke by capturing wobbling piston slipper shoe 5 as slipper skirt 5a protrudes through bore hole 6c of retainer plate 6, thus applying said pull force on piston 4 to slipper shoe 5 which slides on wedge 12 face angle when shaft 11/ 11a rotates. The retainer plate 6 is illustrated in Figs. 3A-3D.

In addition to prior art complex linkages which are wear prone and expensive to fabricate, time proven axial piston pump downstroke retention methods are exampled by Hugelman U.S. 7,794,212. Conventionally, a spherical ball nose segment, post, and spring assembly performs the dual function of imparting the necessary force to counteract the piston suction forces previously described, as well as perform fixed center positioning of an axial piston retainer plate.

Such prior art methods of center positioning a piston retainer plate prevent its general radial misalignment by fixing and holding its operational centered position from a measured machined center point of the retainer plate itself. Final part fabrications all hold fixed and established center point locations and must have engineered manufacturing tolerances allowing deviation from theoretical true center, particularly when combined with fixed centerline fabrications of mating assemblies. This fact is further exacerbated by non-mated and separately fabricated axial centerlines of cylinder blocks and axial piston groupings. The upshot of these fabrication and assembly inaccuracies requires establishment of acceptable manufacturing, assembly, and operational dimensional tolerances and clearances. Consideration must also be given to the fact that as pistons reciprocate in their cylinder bores piston slipper shoes/skirts wobble about a fixed piston axis and potentially rotate while a wedge surface slides beneath. Therefore, it is conventionally essential that retainer plate bore holes be made excessively large allowing clearance with piston slipper shoe skirts to avoid binding and galling with respective inside edges of retainer plate bore holes to operate without detrimental effects. This additionally requires slipper shoe 5 base diameters to be made larger due to larger bore hole 6c diameters.

Applicant's invention provides a means of operationally center positioning the retainer plate 6 which is not centrally fixed in or by the center of the machine or by the center of the retainer plate itself. As seen in Fig. 3A-3D the retainer plate has a central opening 6b through which the shaft 11 passes. In the past, a ball nose centers the retainer plate and is fixed in the machine. Applied as a centering mechanism, the novel method of piston retention utilizes the dynamic geometric position of piston slipper skirts 5a which remain perpendicular to the wobbling face plane of wedge 12 in a predictable track. If desired, this configuration allows open space in the center of the axial machine allowing drive shaft 11 extension through head 19 end of compressor 1 such as illustrated in Fig. 1.

Figs. 6A, 6AA, 6B and 6C illustrate with exaggerated wedge slope the geometric tracking of the arrangement whereby piston(s) slipper skirt 5a defines a centering position of retainer plate 6 and respective bore holes 6c so as to achieve dynamic center positioning of retention plate 6 at any given position of wedge 12 rotation. These illustrations are viewed perpendicular to the plane of retainer plate.

Fig 6A illustrates conventional slipper skirts (heavy black lines) which orbit in a circular pattern inside retainer plate bore holes (heavy dashed lines). As shown in 6A and for reasons previously described, prior art axial piston machines maintain clearance between slipper skirts and retainer plate bore holes when centered by conventional means. Retainer plate bore holes are necessarily made excessively large to maintain clearance and avoid interference with slipper skirts. In applicant's invention in illustration 6AA, slipper skirt 5a clearance with retainer plate bore holes 6c has been reduced to substantially zero (minimal clearance) at the outer orbit perimeter of slipper skirts 5a to locate retainer plate 6 by rotational contact with retainer plate bore holes 6c as slipper skirts 5a orbit therein. This allowance is effective because opposing piston slipper skirts 5a inherently center retainer plate 6 both in the direction and perpendicular to the direction of the wedge slope (arrow in center of each illustration).

Fig. 6 B illustrates slipper skirt 5a (heavy black lines) orbiting geometry which defines the slipper skirt diameter and the retainer plate bore hole 6c (heavy dashed lines) pitch circle dimension. By sizing these dimensions for near zero clearance, the outer perimeter of slipper skirt 5a orbit circles are coincident with retainer plate 6 bore holes and slipper skirts 5a are allowed to freely wobble about piston ball 4a and potentially rotate while being in constant contact with retainer plate bore holes 6c. This configuration results in extremely low friction and even wear on components. In response to rotation of sloped wedge 12, slipper shoes 5 and retainer plate 6 wobble in relation to the center axis of compressor 1/1A. However, as viewed perpendicular to the sloped plane of retainer plate 6, the axial centerline (points) of slipper skirts 5a orbit circumscribing small circles (depicted) between larger circles (depicted) having diameters equal to a major circle (not depicted for clarity) and minor pitch circle (slipper skirt 5a depicted) dimensions of the ellipse created by projecting the piston slipper skirt 5a pitch circle onto the plane of the sloped retainer plate.

FIG. 6C shows representative slipper skirt 5a positions in (3) different retainer plate 6 slope directions. Thus, slipper skirt 5a axial piston groupings (5 illustrated) continuously maintain points of contact with respective retainer plate bore holes 6c in opposing directions that hold center position of the retainer plate 6 as it wobbles in response to wedge rotation.

Conventional axial hydraulic pump piston retention methods might be attempted in a compressor for the consideration of allowing a shaft through a head porting area. However, such configurations have major drawbacks if applied within a gas compressor. Traditional ball nose spring force centering methods create a line contact sliding frictional interface while wobbling around on a ball nose as a wedge is rotated. Although found acceptable for hydraulic pump application where lubricating oil submerges the entire mechanism, this method is highly susceptible to friction and wear in a gas or low lubricity liquid environment. Dry starting is a serious problem and can be extremely detrimental in a gas compressor. It is questionable if extraordinary conventional lubrication efforts could make such a configuration practically acceptable using a ball nose/retainer compressor embodiment.

In applicant's invention, sleeve nose 7a (Fig. 4B and 4D) of retainer sleeve 7 contacts retainer plate 6 at retainer plate rolling ring 6a, (Figs. 3A and 3D). This is important in that the contacting interface of sleeve nose 7a and retainer plate 6 at retainer rolling ring 6a can be "tuned to roll" by selecting the angles for the contacting surfaces of rolling ring 6a and sleeve nose 7a, thereby virtually eliminating sliding friction. In this embodiment, retainer sleeve 7 is not restrained as to rotation in cylinder central bore 3d (Fig.2A), but could be by conventional securing means. In the example shown in Fig. 5, the contacting angles of sleeve nose 7a with retainer rolling ring 6a are equally matched and there is no significant force to drive rotation of retainer sleeve 7. This low friction rolling action of retainer plate 6 as it wobbles upon rotation of wedge 12 is particularly important for dry start operational friction and wear reduction. Figs 5B and 5C show this assembled interface. An alternate embodiment applies the entire angle to only one of these parts 7/7a or 6/6a, consequently there would be no side loading applied to the flat part. In other words, side loading (radial force) vectors applied to retainer sleeve 7 to cylinder central bore 3d could be virtually eliminated; however, the sleeve would rotate in cylinder central bore 3d due to the vector forces applied by retainer plate 6. If the overall contact angle is unevenly split between parts 7/7a or 6/6a, combinations of retainer plate 6 side loading forces weighed with rolling contact of retainer sleeve 7 and rotation in its cylindrical bore may be optimized with material and lubrication selection to minimize detrimental wear due to sliding and side-loading friction factors.

It is also notable that area in the center of the machine is the driest area within housing core cavity 2a due to centrifugal force spinning oil from inner to outer radial locations. Therefore, a low friction rolling action of these centralized components is a significant advantage especially for dry starts. Retainer plate 6 must be strong enough to withstand deformation due to cantilever reactive forces applied by motion of piston 4, as well as providing a novel method of retainer ring 6 center positioning. This benefit would apply to both axial piston compressors and pumps.

Lubrication of dynamic frictional components is particularly important when compressing dry and/or solvent gases such as CO2, and gravity is a primary consideration, therefore orientation of compressor 1/1a must be considered. Initial description will describe a vertical orientation of the preferred embodiment shown. Oil lubricated axial devices used as gas compressors are prone to experience dry running in certain conditions, and consideration should be given to where oil pools and drains. Radial centrifugal force of rotational components sling oil from inner to outer circumferential elements, and oil viscosity dilution may occur under certain circumstances. Dry starts exacerbate this manifestation as it takes time for parts to become coated after extended shutdown. Applicant's invention utilizes a variety of these characteristics to advantage in the method of oil distribution to working frictional components. In addition, the separation of mixed entrained gas and oil is a major design consideration of the invention. Oil sump 16 is a cavity located inside housing 2 at base end 20 which is substantially concentric with axial shaft 11/11a. Oil drain ports 16a and 16b are integral with oil sump 16, and are used for exit porting of oil to external cooling means if necessary (not shown), and/or oil sampling, and/or draining compressor 1/1A of lubricant. Oil sump 16 is the source of lubrication oil for the machine, and every attempt has been made to limit working gas exposure to this oil reservoir as well as central housing cavity 2a, the area of rotational and compressing components.

Although conventional centrifugal oil pump means are known to be employed within spinning drive shafts in various vertically oriented machines, applicant's invention is an improvement for upright vertical axial piston compressors. In a vertical position, adequate lubrication of piston slipper shoe 5 at the frictional interface with wedge 12 sloped angle face is a difficult challenge. Applicants invention embodies an improved centrifugal wedge 12 located within the wedge structure that allows even distribution of hydrodynamic oil film over the wedge 12 surface while providing adequate internal splash lubrication necessary above the wedge 12 for lubricating the innermost frictional mechanisms. These are the innermost bottom piston 4 surfaces as well as the piston retainer plate 6 and retainer sleeve 7 interface. The wedge 12 further contains a dry start and slow start oil reservoir in the wedge which distributes stored oil on startup. Additionally, the wedge 12 simultaneously spin lifts and splashes oil around its outer and upper perimeter, splash lubricating piston 4 and slipper shoes 5 bottom outer surfaces.

The new method effectively introduces centrifugal pumping configuration achieved through the spinning wedge 12, which now has the dual function of a traditional wedge and as a pump. Wedge cavity 12c is oil filled in fluid communication with oil sump 16. In operation, oil enters wedge cavity 12c near the axial center of the machine at a reduced diameter 11b of shaft 11/11a, which may or may not exhibit an impeller surface configuration. Upon operational rpm of shaft 11/11a, oil is centrifugally spun upward from wedge cavity 12c whose outer wall is at a greater radial distance from spinning shaft 11/11a. Oil continues to rise up through at least one wedge riser cavity 12a whose outer wall is at a greater radial distance and forced up into wedge oil distributor cavity 12b. Wedge riser(s) hole(s) or cavity 12a may also be configured as a combined monolithic structure functional with shaft 11/11a and wedge 12, or a rotational locking interface exampled as deepened spline or keyway channels or other conduits (not shown) providing one or more oil paths upward inside the rotating wedge member. Oil is centrifugally distributed from wedge oil distributor cavity 12b over wedge 12 sloped face supplying abundant oil to slippers and pistons, and providing a hydrodynamic lubrication film on the surface of wedge 12 on which slipper shoe 5 lifts and rides. The walls of oil distributor cavity 12b may be contoured to maximum effect for enhancing oil film continuity over the planar wedge 12 surface, as well as determine the "throw" of excess oil that is spun off and up to the inner mechanisms. Oil channel holes (not shown) in communication with oil distributor cavity 12b may be radially drilled to points exiting the sloped face of wedge 12 in line with or inboard near the center of the circumscribed path of slipper shoes 5 if required for additional slipper shoe 5 lubrication. The bottom of wedge distributor cavity 12b may form one or more reservoir pockets 12d. Figs. 7d and 7c illustrate one or more holes or cavities configured to collect and store oil upon shutdown to achieve a lubricating dry start advantage by reducing time for oil to reach wedge 12 face when rpm is initialized.

Gas/oil separation and gas return within central housing cavity 2a is additionally addressed by applicants' invention. Central housing cavity 2a and all adjacent housing cavities must be maintained at operational suction pressure of the machine and not allowed to build pressure within due to gas blow-by. These communicating cavities must be vented back into suction regions in head 19/19a. It is advantageous to vent this gas immediately to reduce mixing exposure to lubricating oil and particularly advantageous to separate this gas from the oil before venting. Operated in a vertical position, gas blow-by past the piston 4 and or other high pressure leakage into housing core cavity 2a is vented up thru retainer sleeve bore 7b clearance between shaft 11 and retainer sleeve 7 interior bore wall. The space between shaft 11, oil slinger protrusion 11c and retainer sleeve 7 at entrance to retainer sleeve bore 7b may be engineered to optimize the gas/oil separation and venting function. Vented gas passes up through retainer sleeve spring 8 in central cylinder bore 3d, and further passes within a radial cut cylinder vent slot 3c (Fig. 2A) along the head end 19 face of cylinder block 3. Vented gas from vent slot 3c is further routed around cylinder bolt 9, though valve assembly 10 bolt access holes 9b to thread into head 19/19a in hole 9c. Counterbore 19e (Fig. 9c) registers with assembly 10 bolt access holes 9b and continues housing 2 venting path to head suction vent slot 19f (Fig. 9A).

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To achieve oil separation prior to gas venting, shaft oil slinger 11b on shaft 11/11a is a close engineered clearance at entrance to retainer sleeve bore 7b. Shaft rotation of shaft oil slinger 11b acts as a centrifugal oil separator, slinging oil and/or oil foam radially outward to help separate gas from oil and also lubricate the contact interface retainer sleeve nose 7a of retainer sleeve 7 and retainer plate 6 at retainer rolling ring 6a. This is the driest area of the oil wetted core area of the operational machine. Wear compatibility of retainer sleeve 7 and retainer plate 6 may be enhanced by using differing materials and surface finishes optimizing low wear at the contact interface of retainer plate rolling ring 6a. Retainer plate rolling ring 6a may or may not be a separate insert or application of selected material applied or affixed to retainer plate 6.

Suction refrigerant returned from a vapor compression system may benefit from conditioning gas phase and/or oil separation. Refrigerant returning from such a system may be in a cold liquid or quasi liquid state termed liquid slugs and by-passed oil from a compressor into a system may return with the suction gas. As a practical matter, liquid is considered incompressible and functions to hydraulically impact valve components. As such, liquid returning to a compressor is undesirable and may cause detriment to a reed valve in a compressor, but other valve embodiments may suffer as well. An intake valve might see “oil-canning” deformation; a discharge valve might see valve or backer failure. A solution to this problem is provided by Applicants’ invention and is of particular benefit to a CO_2 system.

It is recognized that refrigeration or cold environment heat pump applications present the real possibility of liquid slugging if the fluid is not effectively evaporated (heated). Conversely, excessively warm suction gas may return which is less dense; therefore, mass flow entering cylinder 3a is reduced and volumetric efficiency is reduced. Using a casting core processes or alternate fabrication(s) one can incorporate into housing 2 interior cavities which substantially encircle central housing cavity 2a. Return suction gas from a system enters compressor 1/1a at housing intake port 18a into housing intake manifold 18 and routed generally circumferentially around central housing cavity 2a, expelling through housing suction ports 2b (Fig. 8C) prior to high pressure compression. This routing as illustrated in Fig.13 increases initial suction gas interior surface contact and dwell time having relatively low velocity as the housing intake manifold 18 may be made large in comparison with available suction gas space volume in head 19/19a. This is useful in that by design, working fluid refrigerant may be conditioned before passing through housing suction ports 2b.

To provide an engineered solution, wall 2c (Fig.1) of housing central cavity 2a provides an oil washed thermal bridge, or alternatively an insulating surface which may be utilized to enhance or resist thermal transfer to the working fluid and/or lubricant cooling. Wall 2c is adjacent to housing intake manifold 18, either of which may or may not be insulated by coating, insert, or other application (not shown). In this way, more heat or less heat may conduct through wall 2c to affect desired return refrigerant in housing intake manifold 18 and conditioning the temperature phase state which is derived largely from the application of the compressor.

Housing intake manifold 18 further maintains substantial segregation of the incoming return gas from oil mist and soaked internal working components within central housing cavity 2a providing low velocity and wall contact dwell time helping separate oil from gas. Housing intake manifold 18 may further contain an oil separation media (not shown or specified). Separated oil thus pools at the bottom of housing intake manifold 18 draining down into housing return oil cavity 17 through weep hole(s) 44 in the structural web between intake manifold 18 into housing return oil cavity 17 as illustrated in Fig. 13. As seen in Fig. 13, the housing intake manifold provides a circular path around the housing 2 to further allow the separation of oil from gas. At least one oil port 17a is provided and used for oil filling and/or returned oil from external cooling means if required, and/or oil level control means installed into port if required. Housing return oil cavity 17 is in fluid communication or integral as a single cavity space with oil sump 16 forming a cavity to establish an oil level, and further provides a space for dissipation of gas from entrained oil and oil foam.

Refrigeration suction gas expels from housing suction ports 2b (Fig. 8C) which are registered by conventional means with head intake ports 19b (Fig. 9A) and access head suction manifold 19c (Fig. 9C and 9G) further communicating with suction valve ports 19d (Fig. 9A).

Suction and discharge valving is accomplished via valve assembly 10 (Fig 1/1A and Fig 12C). Valve assembly 10 consists of a port plate 23 (Fig. 10A,), a suction reed valve plate 24 (Fig. 11A), and a discharge reed valve plate 25 (Fig 12A). Port plate 23 clamps circular suction reed valve plate 24 sandwiched and sealed between port plate 23 and cylinder block 3. Suction valves 24a cover and seal port plate 23 and suction valve ports 23a. Suction valves 24a flex open to cylinder 3a allowing refrigerant gas to fill cylinder 3a upon down stroke of pistons 4, but suction valves 24a are limited in travel by suction valve side stops 24b (Fig. 11B) contacting cylinder side stop recesses 3b (Fig. 2B) which are machined in head end 19 of cylinder block 3. Suction reed valve 24a sidestops 24b position gas pressure loads so as to minimize bending at the valve neck clampline. Suction valve sidestops 24b are positioned alongside the suction valve ports 23a so gas loads are reacted with minimal affect on bending at the neck of suction valve 24a near the clampline. Sidestops are in line across the centroid of the ports to best counteract forces and gas loads around the ports. Reaction forces at the tip of a typical valve create undesirable valve back bending which increases fatigue stresses at the clampline where valve neck bending originates. Therefore, a conventional design utilizing a valve stop at the tip is inadequate for high pressure operation as exampled by CO_2 vapor compression. Center holes 23c in port plate 23, and 24d in suction reed valve plate 24, and 25c in discharge reed valve plate 25 each allow optional shaft 11 thru protrusion.

Means for valving discharge of refrigerant gas from piston cylinder 3a clamps discharge reed valve plate 25 (Fig. 12A) sandwiched and sealed between port plate 23 and head 19/19a. Discharge valves 25a cover and seal port plate discharge valve ports 23b (Figs. 10A and 12C). Discharge valves 25a are pressurized to open and discharge gas from cylinder bore 3a upon full compression up-stroke of pistons 4, but are limited in travel by backer cut 19e (Fig. 9D) in head 19/19a. High pressure discharge gas exits discharge port 23b into discharge manifold 19m (Figs. 9C, 9G) from head discharge port holes 19n (Figs. 9A, 9G.)

It is of note that a conventional alternate (not shown) to housing intake port 18a might locate intake return gas porting directly into head 19 at any convenient radial position. This head

19/19a alternative to housing 2 intake port 18a provides a shunt option to using intake manifold 18 directly piping suction gas directly into head 19/19a. This alternative intake porting option provides a combination of thermodynamic temperature, liquid separation, piping and operational orientation options. This may be of advantage in horizontal operation.

Horizontal operation may be obtained by orienting ports 18a, 17a, 16a in a vertical top position. The oil reservoir becomes the lower half section of compressor 1/1a. Oil lubrication now occurs through splash oil distributed by rotating wedge 12 dipping thru the oil bath. Cylinder vent slot 3c and housing suction port 2b remain above the oil level.

Applicant's invention enables prolific economy of scale manufacturing and a multitude of application deployments. Vertical orientation yields an extremely small footprint and horizontal operation reduces necessary headroom. While singular in axial design functionality and fabrication layout, the combination of a double-ended shaft in a single machine operable in either a vertical or horizontal orientation is unlike any known conventional compressor embodiment. These combined orientation and drive benefits vertical or horizontal, hermetic, semi-hermetic, and open drive options are unknown using a single compressor embodiment such as illustrated in Figs. 14 A-H. A motor, shown generally as motor 46, drives the compressor 1, which may be the compressor illustrated in Fig. 1 or Fig. 1A, as applicable for the various arrangements. The various configurations that the invention allows are meant to illustrate the numerous possibilities in which the invention may be utilized.

IV. DETAILED DESCRIPTION OF THE SECOND EMBODIMENT

In the alternate embodiment, illustrated in Figs. 15-20, there is a semi-hermetic high pressure axial piston compressor/pump which will be simply referred to as compressor 100. The compressor has many similarities with the compressor 1 of the first embodiment. However, the compressor 100 has added features for high speed capability and reduced manufacturing complexity in a semi-hermetic configuration as will be described herein.

As shown in Figure 15, the compressor housing 108 of the compressor 100 is connected by bolts 112 to the sump housing 101 and the head 110. Three external parts with two bolted joints for this embodiment versus five external parts and four bolted joints in comparable reciprocating compressors is a manufacturing advantage for this embodiment. There is an electrical power connection 114 where power is brought into the compressor 100 in order to drive it by means of the electric motor. There is also a suction connection 116 and discharge connection 118. The sump housing 101 has an oil drain plug 104 and an oil sight glass 106.

Fig. 16 is a cross sectional view of the compressor 100. A wedge shaft 120 is vertically mounted in the compressor. The bottom of the wedge shaft 120 is supported by a lower bearing 122 mounted in the sump housing 101. There is an upper bearing 124 mounted in compressor housing 108 near the upper end of the wedge shaft 120 that centrally aligns the wedge shaft while it is spinning. The lower bearing 122 may be a ball bearing, or alternatively, a combination of journal and thrust washer bearings to run at high speed, greater than 3600 RPM.

The wedge shaft 120 drives a wedge 121 as previously described in the first embodiment. The wedge shaft 120 is driven by the electric motor having motor stator end windings 128, motor stator laminations 130 and a motor rotor 132 that is shrunk fit onto the wedge shaft 120.

As more clearly illustrated in Fig. 17, valve plate 135 and cylinder block 134 are fixed below the head 110. There is a plurality of pistons 136 in cylinder bores 138. A freely articulating shoe 142 is swaged onto the piston ball 140 of each piston. The shoes with attached pistons are assembled in corresponding bores of retainer plate 146. The flange of each shoe is trapped between the retainer plate and the wedge 121. As the shaft rotates, a shoe is pushed upward by the wedge and shoes on the opposite side of the retainer sleeve 148 are pushed downward by the retainer plate which is a lever pivoting on a fulcrum at contact point 144 of the retainer sleeve. As the wedge 121 rotates and pistons 136 reciprocate axially, the retainer plate146 wobbles, but does not rotate. The retainer sleeve 148 is centered in the cylinder block 134 and is positioned to hold the wobbling retainer plate 146 down so pistons, attached to shoes 142, go through the entire range of motion allowed by the rotating wedge face. The retainer sleeve 148 reacts against significant piston inertia and suction forces on the retainer plate 146. That net force is transmitted through a small area around the contact point 144. The contact point 144 between the retainer plate 146 and the retainer sleeve 148 is equally distant from the centerlines of the retainer plate and the retainer sleeve, geometrically defined by a line bisecting the wedge angle formed by the compressor 100 centerline and a plane through piston balls 140, for rolling contact to avoid mechanical losses and failures from sliding at the contact interface that is inherent in hydraulic pump retention mechanisms. This is illustrated and described in the first embodiment.

In this alternate embodiment, the flat retainer plate 146 is less complicated to manufacture than the conical profile in the first embodiment or the spherical interface in traditional axial piston hydraulic pumps, but with the contact point 144 positioned for rolling contact in both embodiments. The rolling action of the retainer plate 146 on the retainer sleeve 148 is explained in detail in the previous embodiment. The retainer plate is radially restrained by the piston shoes as explained in the first embodiment.

In this alternate embodiment, the controlled length retainer sleeve 148 rigidly maintains a small axial clearance in the stack of parts between the wedge 121 and the valve plate 135 without a spring. Consequently, speed is not limited by the spring force. Elimination of the spring with associated locating features reduces manufacturing complexity. Friction loss at the shoe-to- wedge interface is less under normal operating conditions.

The alternate embodiment also has a modified oil distribution system. As seen in Fig.

16, there is an offset passage 160 extending upward from the bottom of the wedge shaft 120. As oil enters the end of the wedge shaft 120 under the oil level of the oil sump 102, the spinning shaft forces the oil outward and upward along the outer wall of the offset passage 160. Above the small diameter rotor 132 is an upper offset passage 165 in fluid communication with the offset passage 160 and at a greater offset from the center of the shaft 120 to force the oil upward into the centered large diameter receiving cylinder 166. Centrifugal force distributes oil evenly around the inside diameter and up the walls of the receiving cylinder 166 for uniform distribution over the top edge 162, over conical surface 168 and onto the wedge surface for lubricating shoes as they slide over the wedge 121. The oil then flows back through an oil return passage 111 in housing 108 to the oil sump 102 for recycling through the compressor. To provide oil to the wedge 121 at start-up, there is a small start-up reservoir 170 that is located in the floor of the receiving cylinder 166. Oil drains into the start-up reservoir 170 when the compressor is shut off and is distributed to the wedge surface 121 on start-up.

In the first embodiment, as well as prior art reciprocating piston compressors, both reed valves and traditional ring valves in the suction function flex to open under pressure created by fluid flow through the valve and are closed by the flexing force in the valve as fluid flow decreases. At higher speed, the forces and deflections are higher which increases the valve stress, leading to failure, so speed has to be limited, typically 1800 rpm rated speed. Applicant has developed a new suction valve illustrated in Figs. 18 and 19. Cylinder pressure normally holds the suction plate valve 200 closed against the seat in the valve plate 135, Fig. 19. As the piston 136 moves downward, low pressure in the cylinder allows suction gas to push the suction plate valve 200 downward which opens the valve for gas flow. The suction plate valve 200 moves freely in the valve guide bore 202, as seen in Fig. 18, with no axial resistance until it is stopped by the shelf 198 created by the top of the block 134 outside the cylinder bore 138. In this open position, the valve 200 remains flat and is supported all around its outer edge to resist downward gas forces with low valve stress, even at high speed. An additional benefit is that the suction plate valve 200 stays wide open until the piston passes through bottom dead center (zero piston velocity) so the cylinder pressure can nearly equalize with the pressure entering the valve unlike reed valves and traditional ring valves that maintain a pressure drop down to zero flow.

There is a common omni-ring discharge valve 206 illustrated in Figs. 18 and 19 (a single valve part serves all cylinders). The omni-ring discharge valve 206 is radially positioned by a large bore 208 in the valve plate 135. For each cylinder where cylinder pressure is less than the discharge pressure, the discharge pressure holds the omni-ring discharge valve 206 closed flatly against the port seat on the top side of the valve plate 135, Fig. 18. During the compression stroke, the piston 136 moves upward and cylinder pressure overcomes discharge pressure locally at that cylinder port to flex the omni-ring discharge valve 206 open against the bottom of the head 110 above that particular port, Fig. 19. At the end of the compression stroke, the piston stops at top dead center, flow stops, pressures equalize and flexing forces in the omni-ring discharge valve 206 returns the valve to its normal flat shape against the port in the valve plate 135 and the closed omni-ring discharge valve 206 seals as shown in Fig. 18. The manufacturing advantage is that the head 110 limits valve travel without independent backer parts and fasteners used in other devices.

In this embodiment, liquid separation is modified to provide cooling for the semi-hermetic motor. In Fig. 20, a high velocity mixture of liquid and gas enters the offset, horizontal suction connection 116 and flows toward the opposite wall that is curved around the compressor center. Dense liquid flung against the curved wall of the compressor housing 108 collects along the wall, separates from the gas and drops downward due to gravity. Less dense gas may flow all the way around inside the curved wall of the housing, and some may recycle past the suction connection 116. Because the passage 212 gradually narrows along this flow path, most of the gas flow is squeezed between and over the end winding wires, which cools the end windings 128, as the gas is pushed upward to the suction gas passage 109 in compressor housing 108 as seen in Fig. 16. Cast features in the housing provide this separation and cooling functionality without added parts or incremental machining cost.

In this alternate embodiment, the compressor 100 is dynamically balanced. There is a couple created by axial piston inertia forces offset from the compressor centerline. This couple is about a horizontal axis through the wedge. There is another couple about this same axis from offset centrifugal forces on mass distributed non-symmetrically along the wedge shaft. By design, these couples offset each other for dynamic balance. This enables low vibration, high speed operation without vibration dampers or other added parts.

In this embodiment, the electrical box 105 in Figure 15 is integrated into the cast compressor housing 108 to eliminate a joint between a loose electrical box and the housing typical in traditional reciprocating compressors. Likewise, gas and oil passages 212 and 111 are integrated into the compressor housing casting to reduce manufacturing complexity.

Features of rolling retainer interface, solid retainer sleeve, non-flexing plate valve for suction and dynamic balance enable the technical breakthrough of high capacity from speed above traditional 4-pole motor speed (1800 rpm at 60 hz.). Elements of this embodiment have been tested at 2-pole motor speed (3600 rpm at 60 hz) and with all the elements of this embodiment, higher speed is achievable.

The flat retainer plate, solid retainer sleeve, oil distribution system, non-flexing plate valves for suction, omni-ring discharge valve and features integrated into castings reduce part count and complexity relative to traditional concepts in piston compressors/pumps.

V. CLAIMS

What is claimed is:

1. An axial compressor comprising:

a centrally mounted drive shaft;

a wedge mounted in the axial compressor, the wedge having a planar angular surface;

a plurality of slipper shoes placed on the planar angular surface of the wedge;

a plurality of cylinders disposed in a cylinder barrel;

a piston disposed in each of the cylinders, each piston having one end operatively connected to the slipper shoe and the other end slidably received in the cylinder;

drive means for causing the wedge to drivingly engage the pistons thereby causing the pistons to reciprocate;

a retainer plate engaging the slipper shoes for maintaining the slipper shoes in contact with the wedge;

a retainer sleeve mounted in axial parallel alignment with the drive shaft, the retainer sleeve having a top and bottom, the bottom of the retainer sleeve engaging the retainer plate and restraining the retainer plate from movement parallel to the axis of the drive shaft, the retainer plate securing the slipper shoes beneath the retainer plate to maintain the slipper shoes in contact with the wedge angular surface.

2. The axial compressor of claim 1 and further comprising a contoured surface on the bottom of the sleeve, and a complementary receiving surface on the retainer plate engaging the contoured surface on the bottom of the sleeve whereby the two surfaces engage each other in continuous rolling engagement as the retainer plate changes its angular orientation with respect to the pistons during operation of the compressor.

3. The compressor of claim 1 wherein the top of the retainer sleeve engages a valve plate for securing the retainer sleeve between the valve plate and the retainer plate to prevent axial movement of the retainer sleeve without the use of a spring.

4. An axial compressor comprising:

a centrally mounted drive shaft having a top and a bottom;

a wedge mounted in the axial compressor, the wedge having a planar angular surface;

a plurality of slipper shoes placed on the planar angular surface of the wedge;

a plurality of cylinders disposed in a cylinder barrel;

a piston disposed in each of the cylinders, each piston having one end operatively connected to the slipper shoe and the other end slidably received in the cylinder;

drive means for causing the wedge to drivingly engage the pistons thereby causing the pistons to reciprocate;

a retainer plate engaging the slipper shoes for maintaining the slipper shoes in contact with the wedge;

a retainer sleeve mounted in axial parallel alignment with the drive shaft, the retainer sleeve having a top and bottom, the bottom of the retainer sleeve engaging the retainer plate and restraining the retainer plate from movement parallel to the axis of the drive shaft, the retainer plate securing the slipper shoes beneath the retainer plate to maintain the slipper shoes in contact with the wedge angular surface;

an oil sump at the bottom the compressor, the bottom of the drive shaft disposed in the oil sump;

at least one offset oil passageway offset a first distance from the center line of the drive shaft and extending upward from the bottom of the drive shaft, the rotation of the drive shaft forcing the oil from the sump upward in the oil passageway for distribution onto the wedge surface for lubricating the slipper shoes as they slide over the wedge.

5. The axial compressor of claim 4 wherein the oil passageway comprises a lower passageway and an upper passageway in fluid communication with each other, the lower passageway at a first offset distance from the center line of the drive shaft and the upper passageway at a second offset distance from the center line of the drive shaft that is greater than the first offset distance of the lower passageway to force oil into a receiving compartment for distribution onto the wedge surface.

6. An axial compressor comprising:

a centrally mounted drive shaft;

a wedge mounted in the axial compressor, the wedge having a planar angular surface;

a plurality of slipper shoes placed on the planar angular surface of the wedge;

a plurality of cylinders disposed in a cylinder barrel;

a piston disposed in each of the cylinders, each piston having one end operatively connected to the slipper shoe and the other end slidably received in the cylinder;

drive means for causing the wedge to drivingly engage the pistons thereby causing the pistons to reciprocate;

a retainer plate engaging the slipper shoes for maintaining the slipper shoes in contact with the wedge;

a retainer sleeve mounted in axial parallel alignment with the drive shaft, the retainer sleeve restraining the retainer plate from movement parallel to the axis of the drive shaft, the retainer plate securing the slipper shoes beneath the retainer plate to maintain the slipper shoes in contact with the wedge angular surface;

a valve plate having a top and bottom surface;

a circular suction valve on the bottom surface and a ring-shaped discharge valve on the top surface;

the suction plate valve pulled downward during a suction stroke to open the suction plate valve thereby allowing gas to flow into the cylinder;

the suction plate valve pushed upward during a discharge stroke to close the suction plate valve thereby restricting gas from flowing out through the suction plate valve;

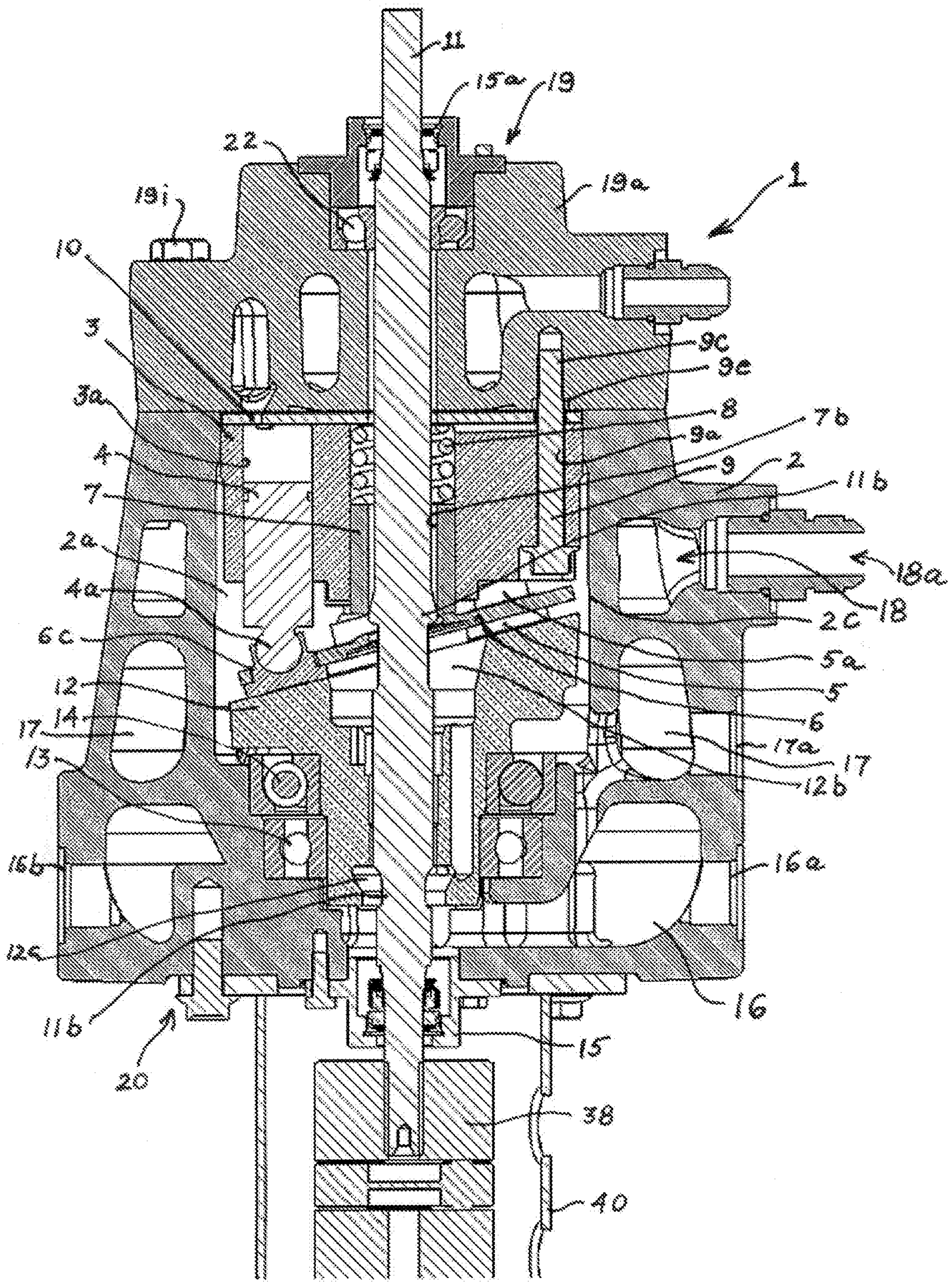
the ring-shaped discharge valve pulled downward during the suction stroke to close the ring-shaped discharge valve thereby restricting gas from flowing into the cylinder through the ring-shaped discharge valve;

the ring-shaped discharge valve flexing upward during the discharge stroke thereby allowing gas to flow out of the cylinder through the ring-shaped discharge valve.

7. The axial compressor of claim 6 and further comprising a suction guide bore for receiving the suction valve which is free to axially move in the suction guide bore, and

a discharge guide bore for receiving the ring-shaped discharge valve which is free to axially move in the discharge guide bore;

8. The axial compressor of claim 6 and further comprising a supporting shelf on the cylinder block surrounding a perimeter portion of the suction plate valve for supporting the outer edge of the suction plate valve during the suction stroke.


11
15a
19
22
19a
1
19i
10
3
9c
9e
3a
8
7b
9a
4
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2
11b
7
2a
18a
4a
2c
18
6c
5a
5
12
6
14
17
17a
17
13
12b
16b
16a
12c
11b
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15
38
40

F/G. 1

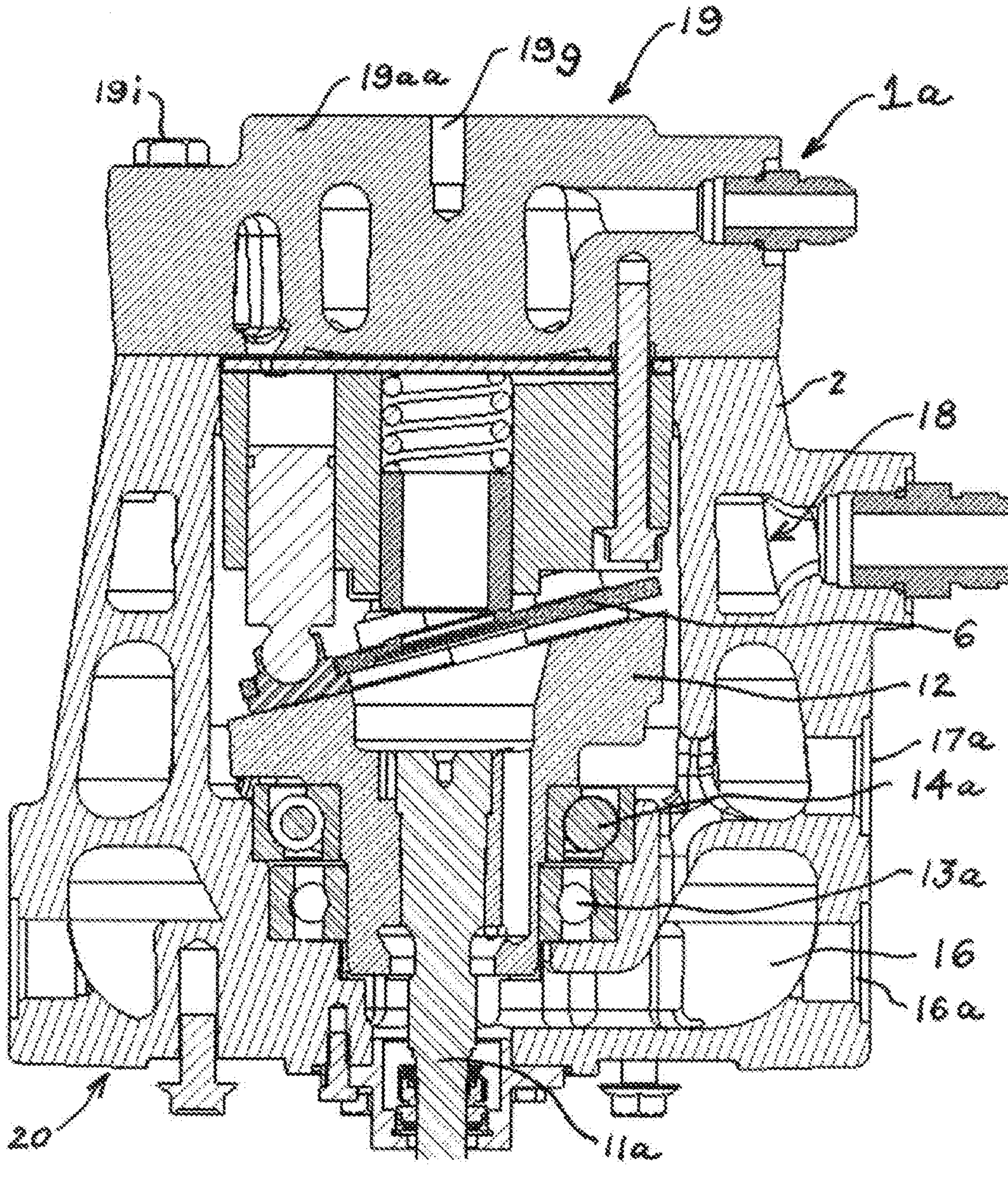

FIG. 1A

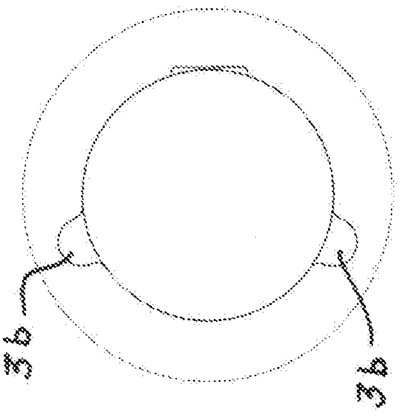

FIG. 2B

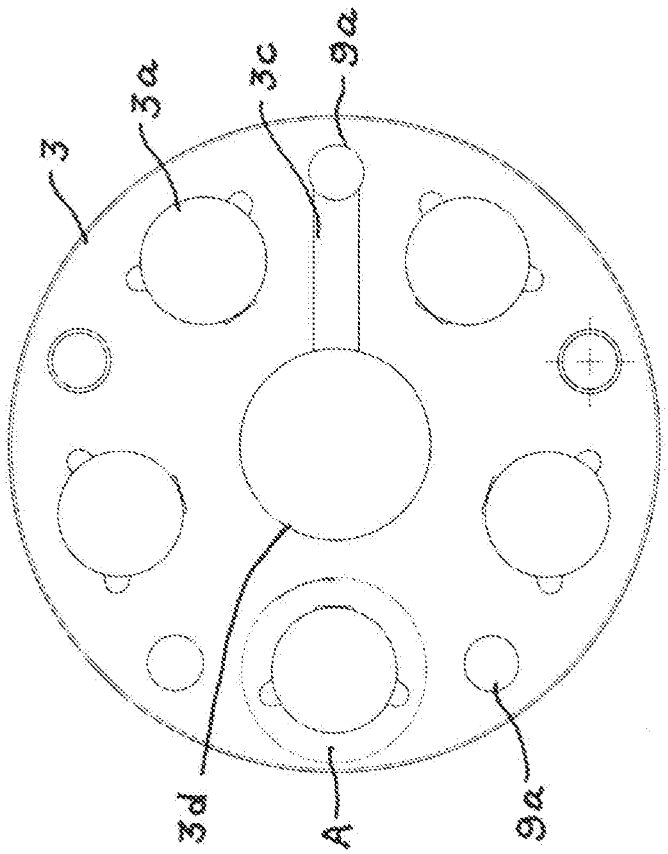

FIG. 2A

FIG 3D

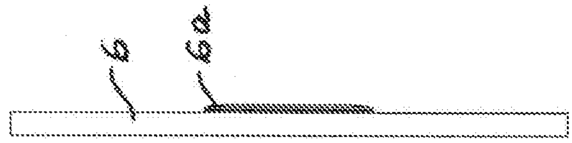

FIG 3C

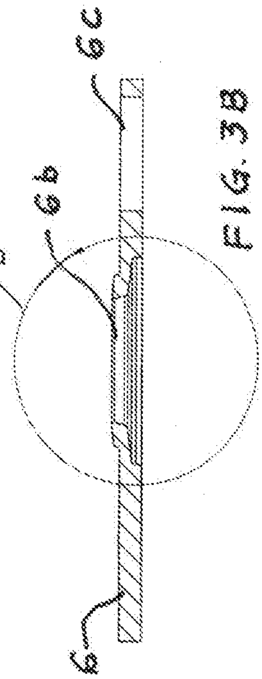

FIG. 3B

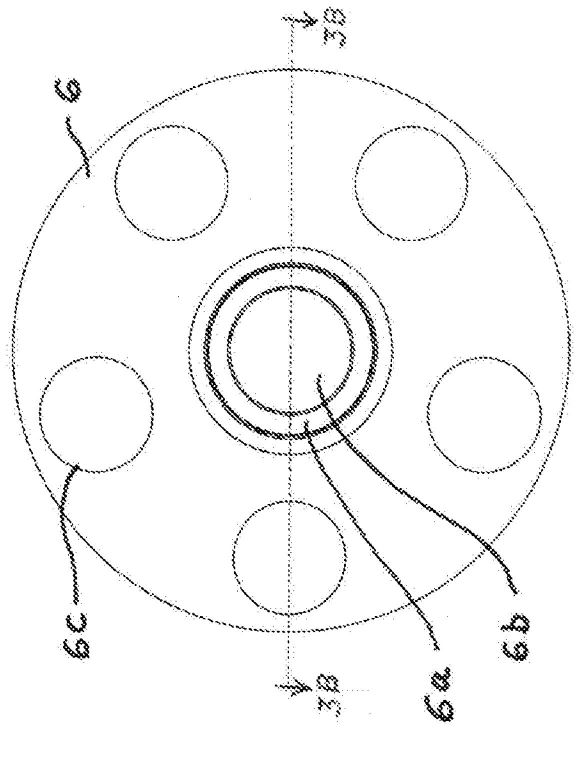

FIG. 3A

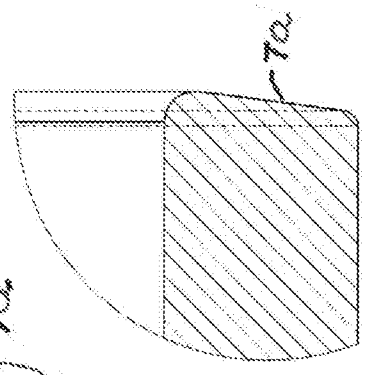

FIG. 4D

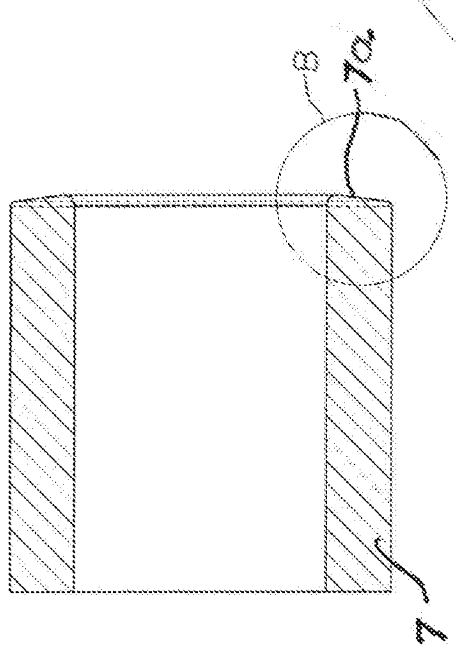

FIG. 4C

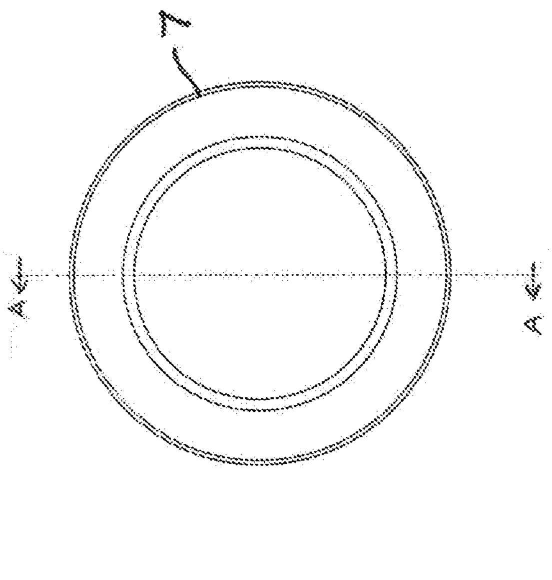

FIG. 4A

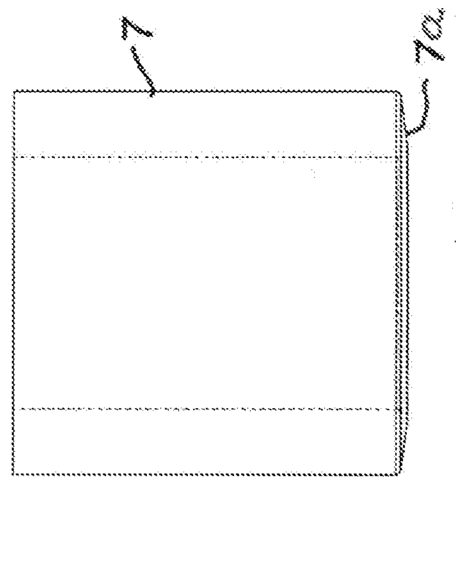

FIG. 4B

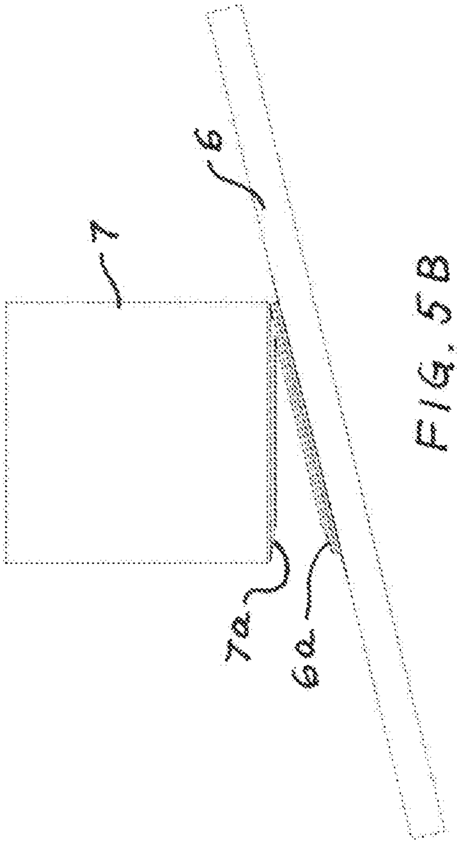

FIG. 5B

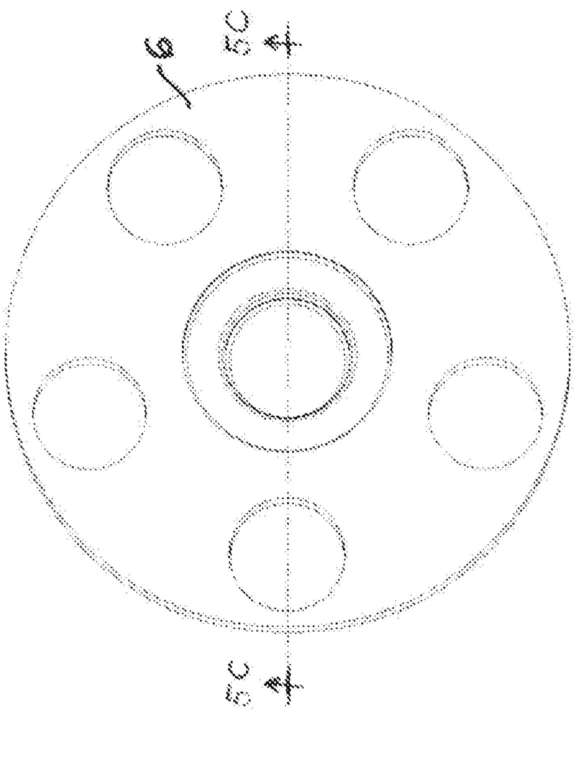

FIG. 5A

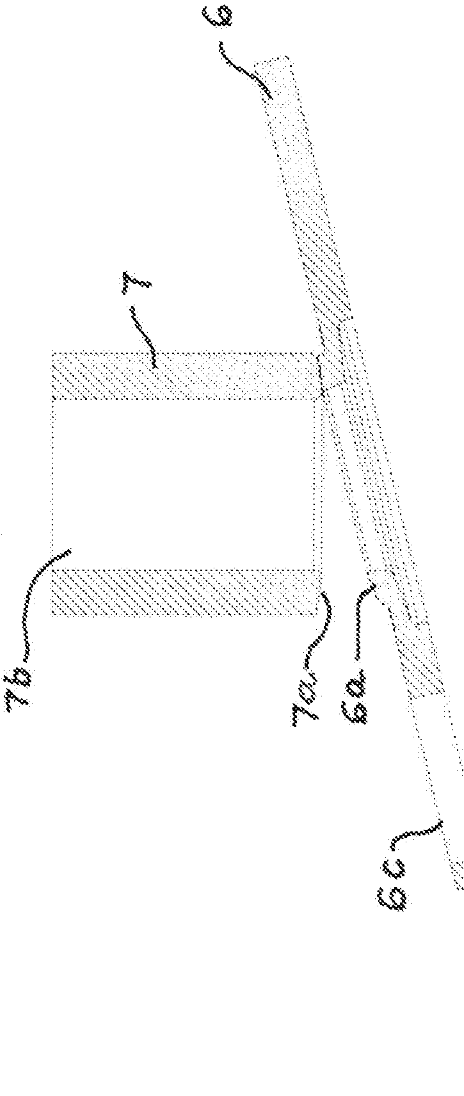

FIG. 5C

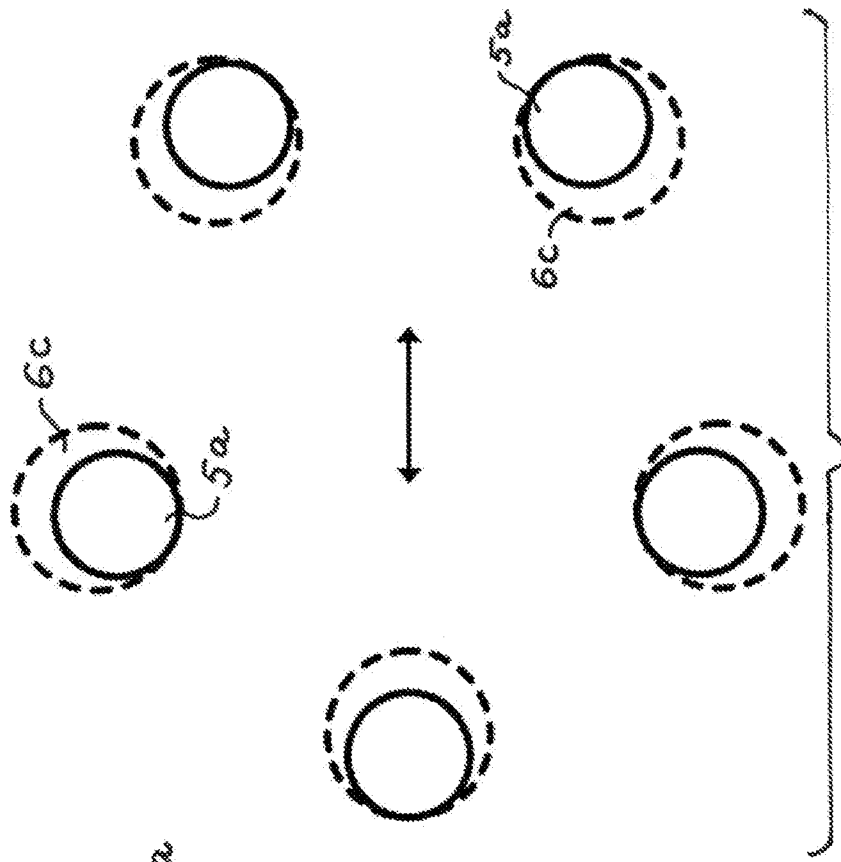

FIG. 6AA

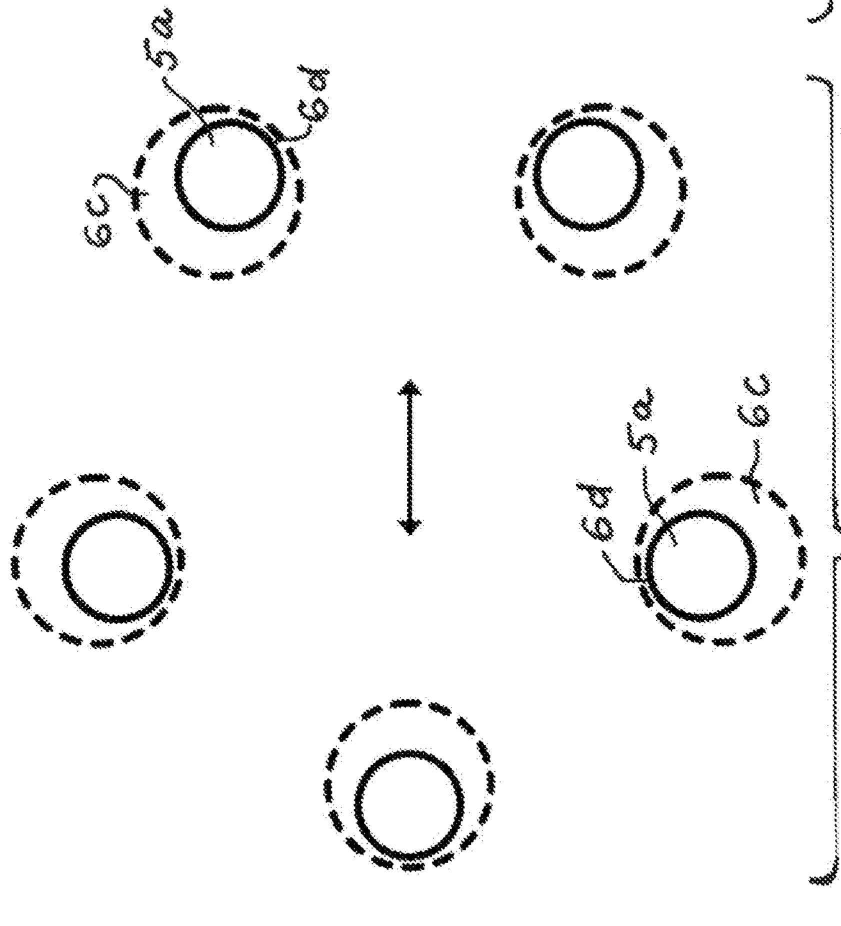

FIG. 6A

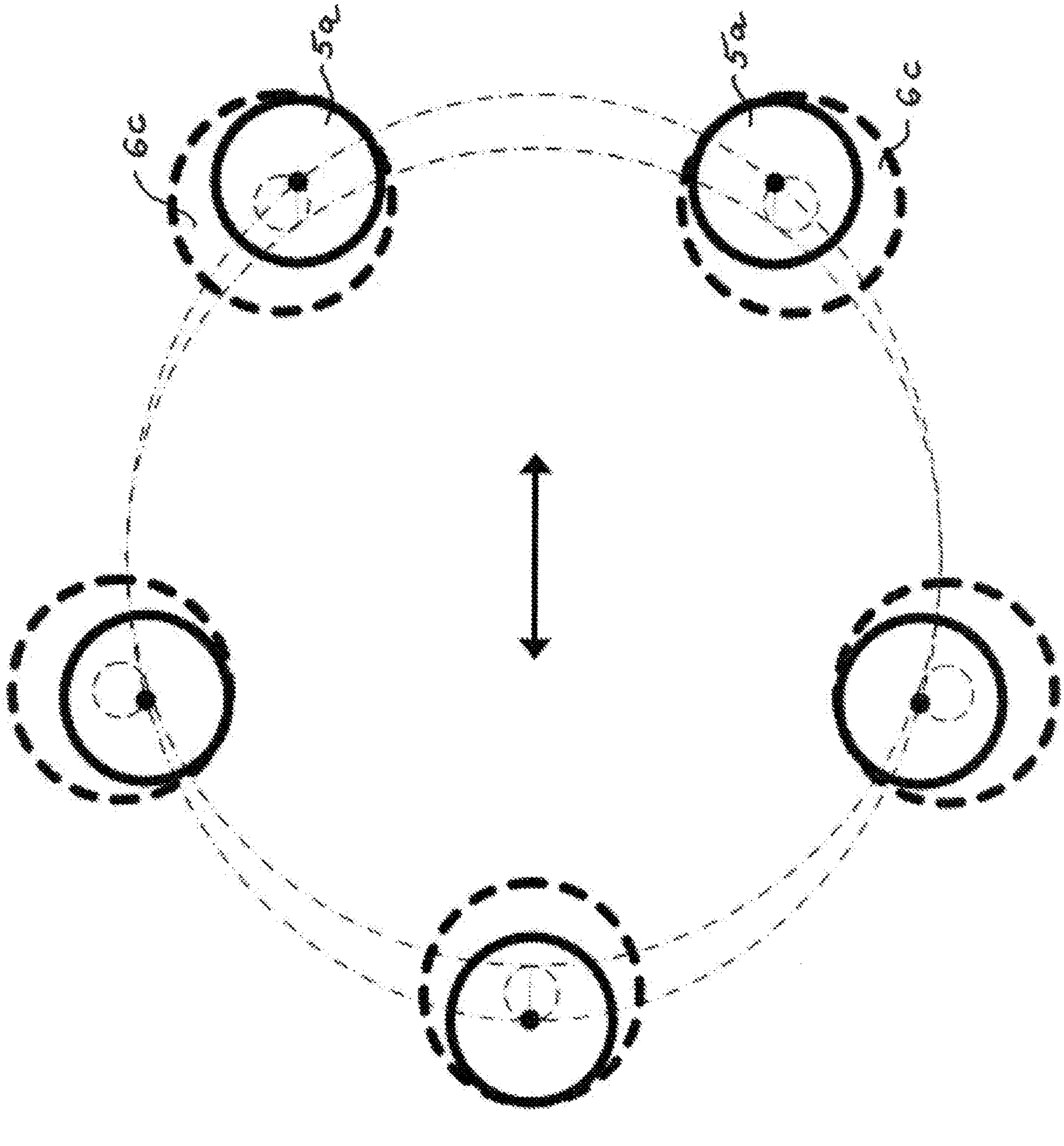

FIG. 6B

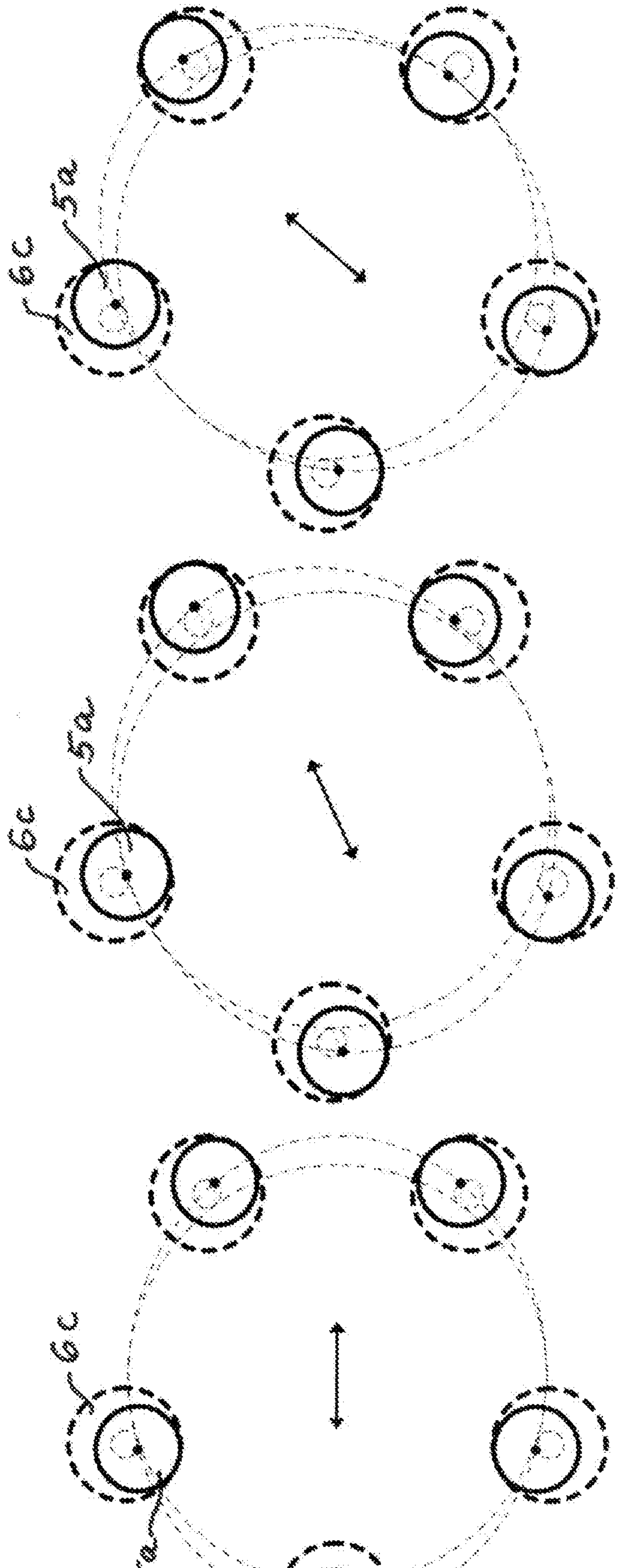

FIG. 6C

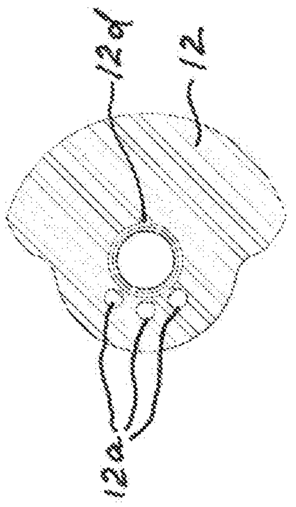

FIG. 7D

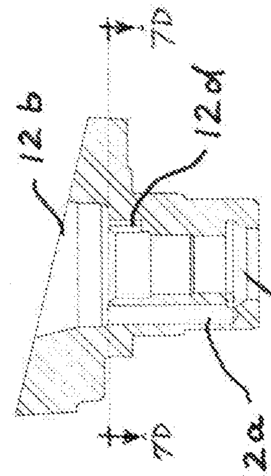

FIG. 7C

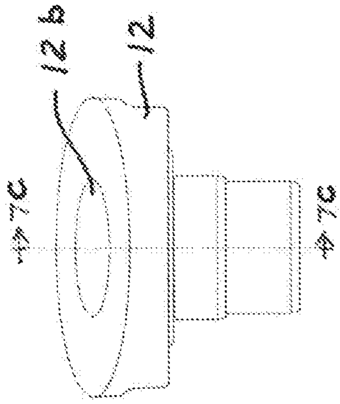

FIG. 7B

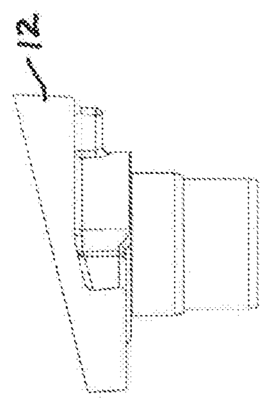

FIG. 7A

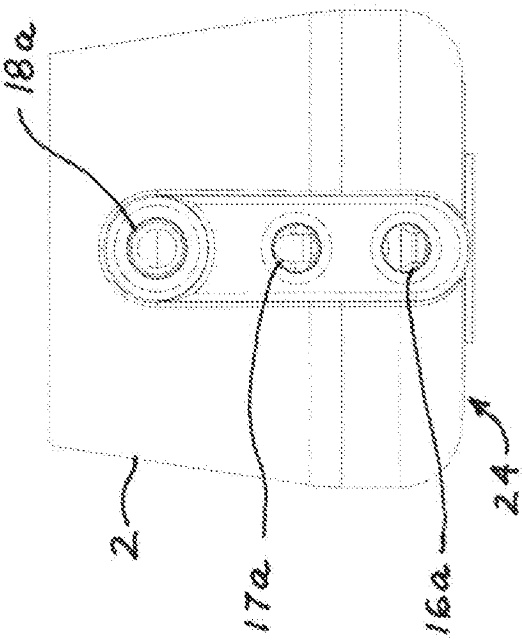

FIG. 8B

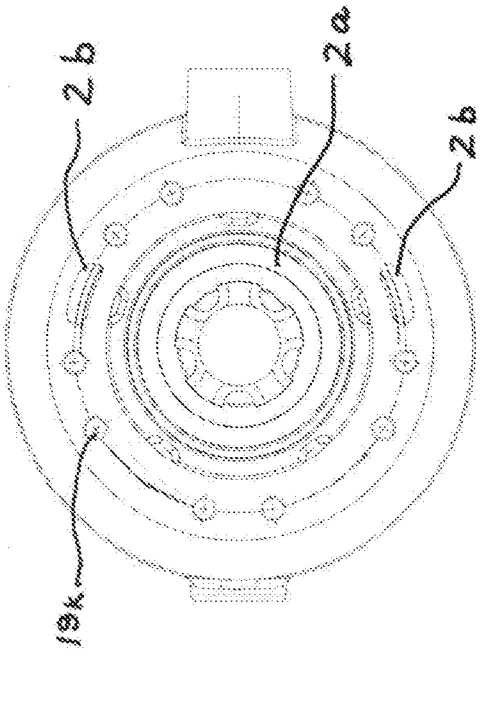

FIG. 8C

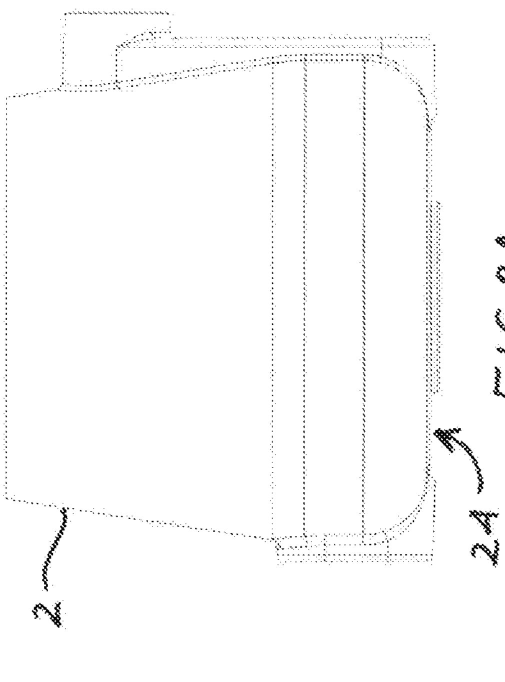

FIG. 8A

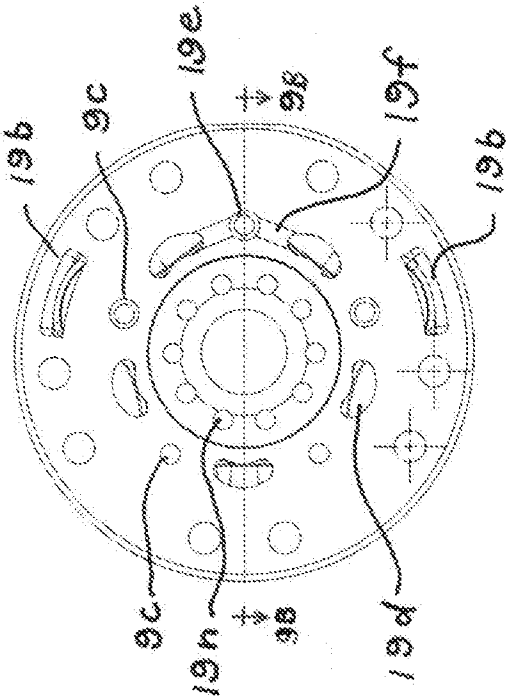

FIG 9A

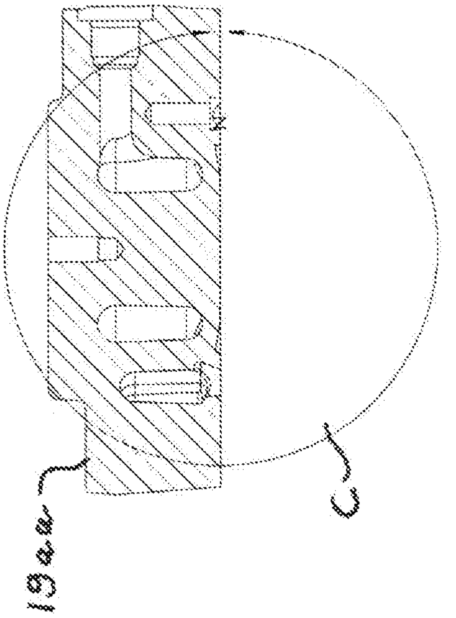

FIG. 9B

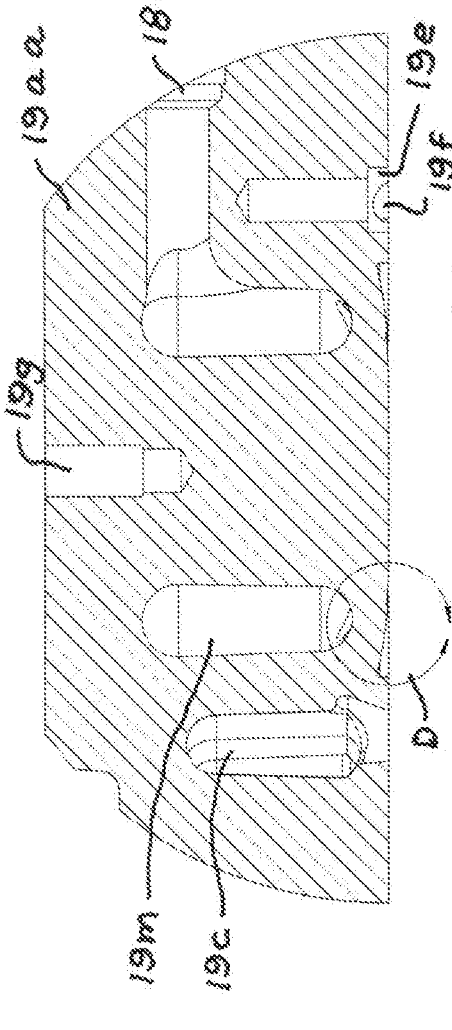

FIG. 9C

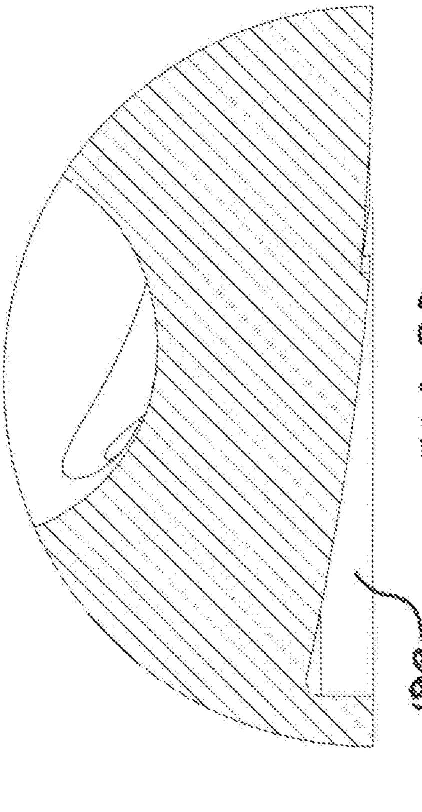

FIG. 9D

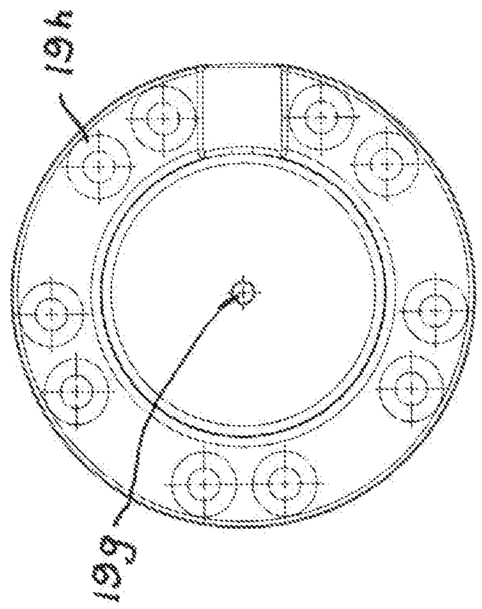

FIG. 9F

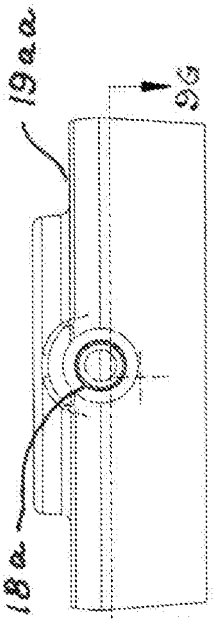

FIG. 9E

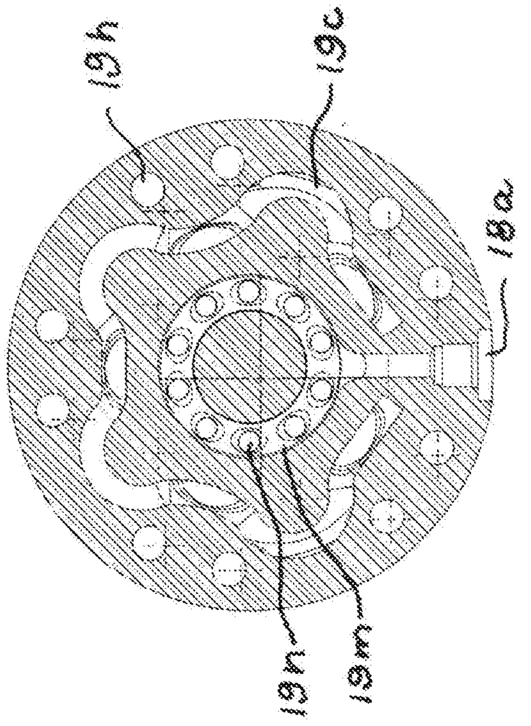

FIG. 9G

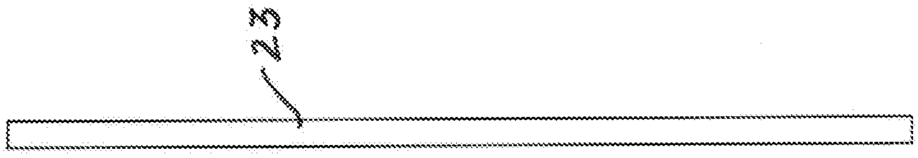

FIG. 10B

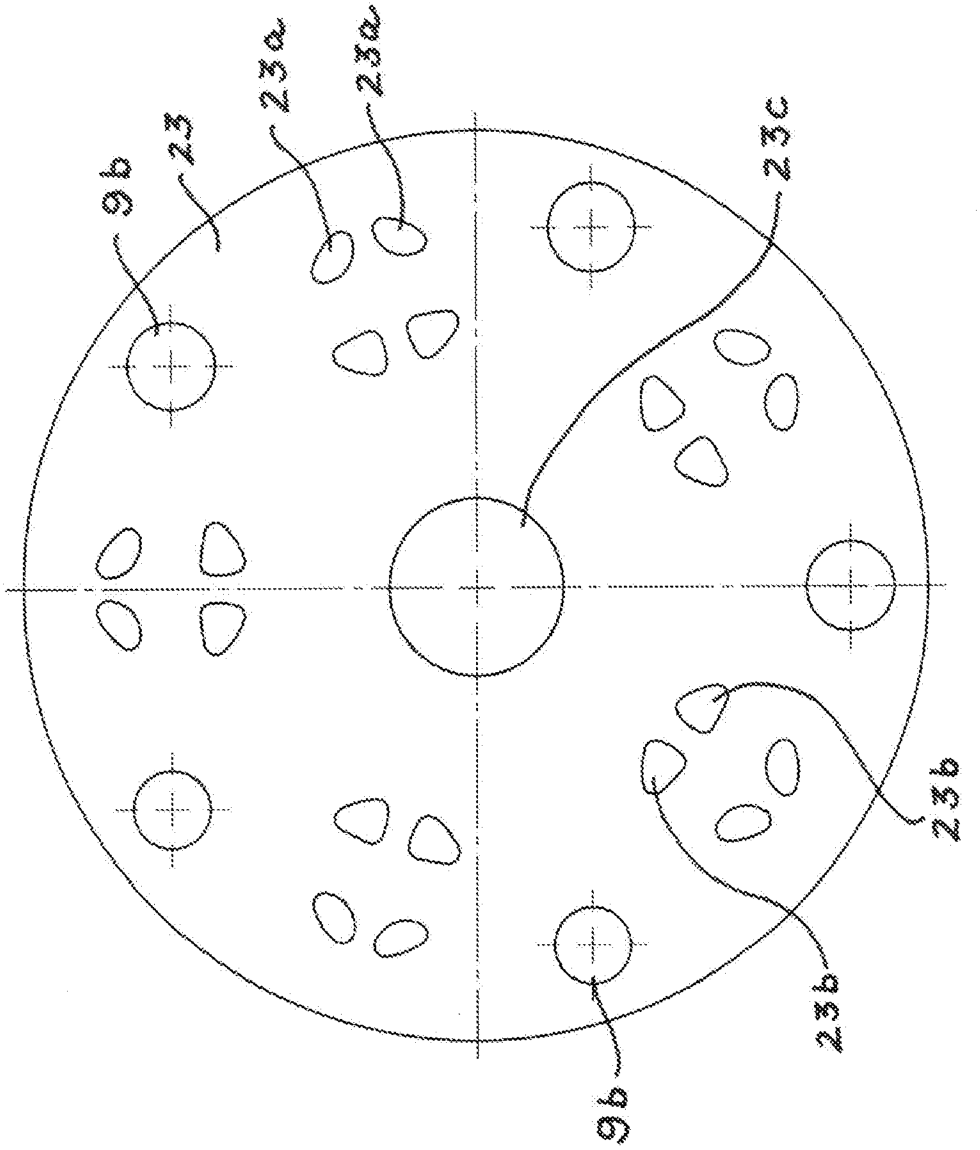

FIG. 10A

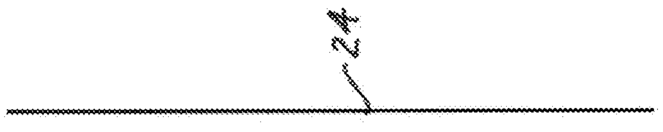

FIG. 11C

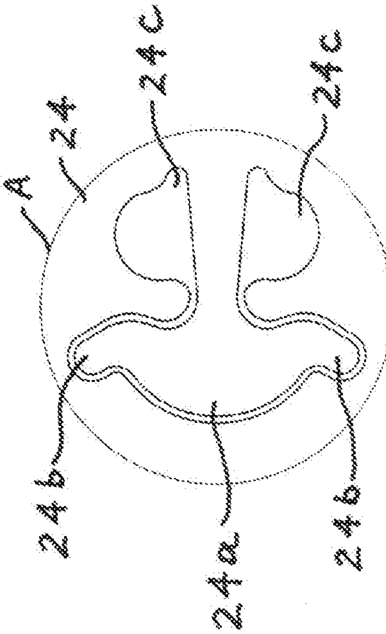

FIG. 11B

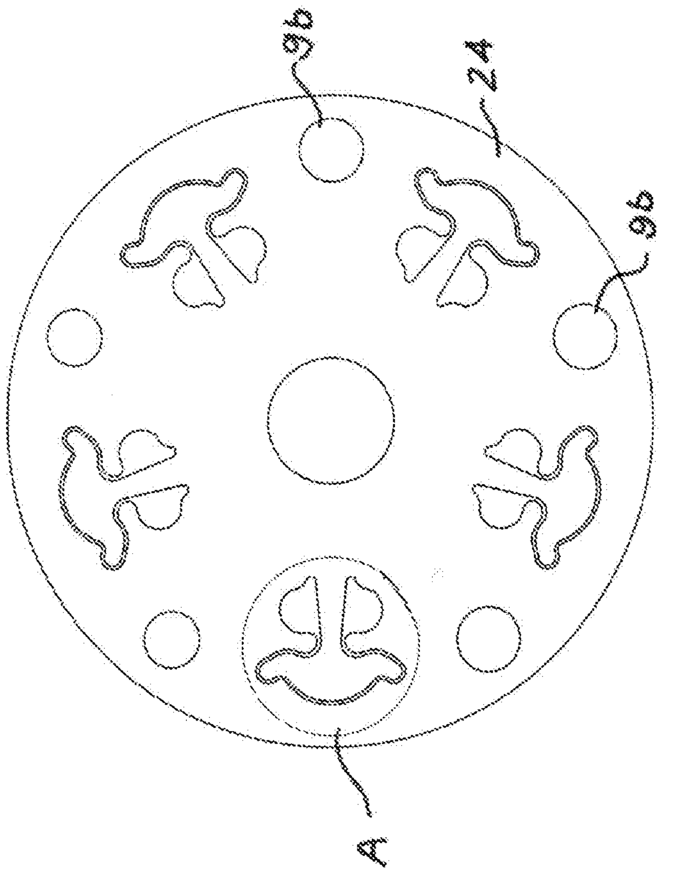

FIG. 11A

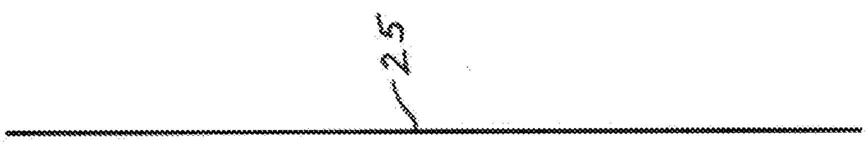

FIG. 12B

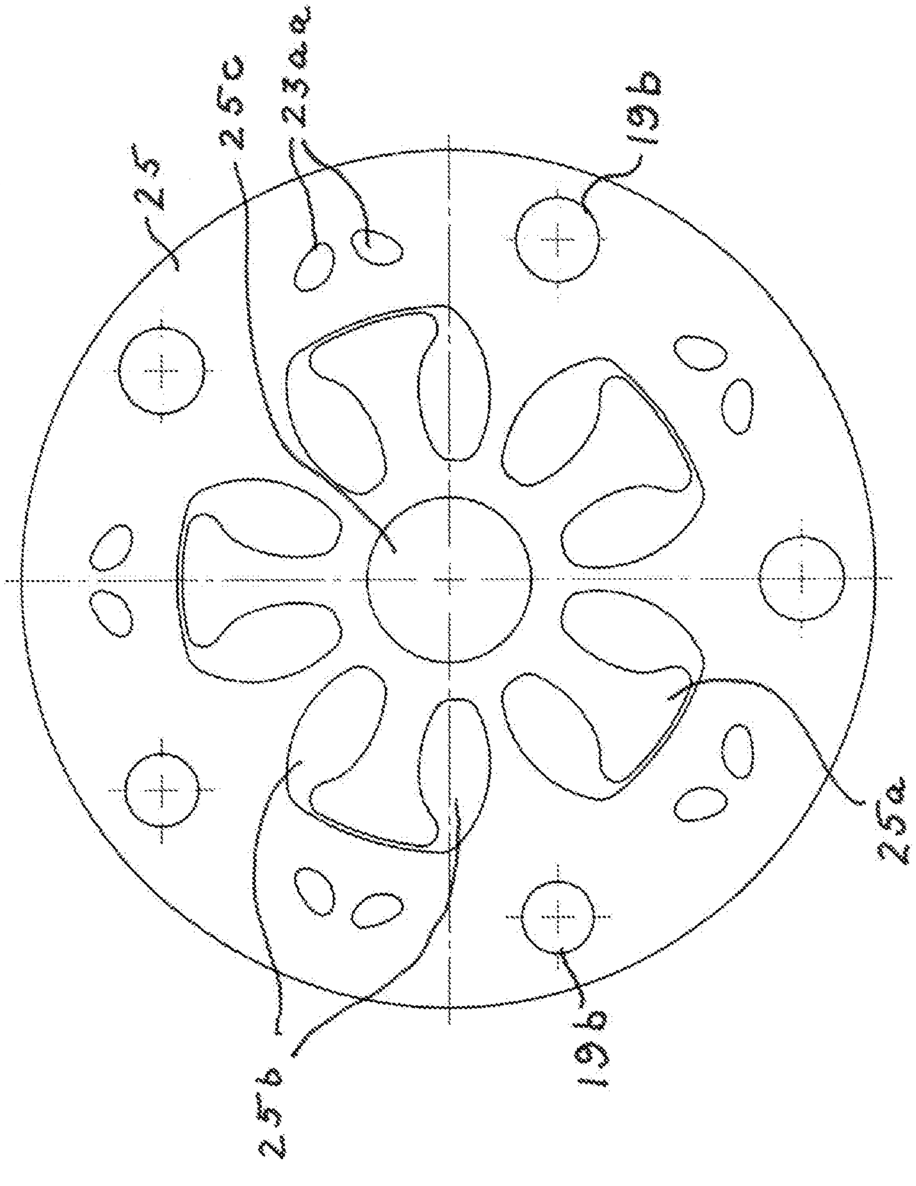

FIG. 12A

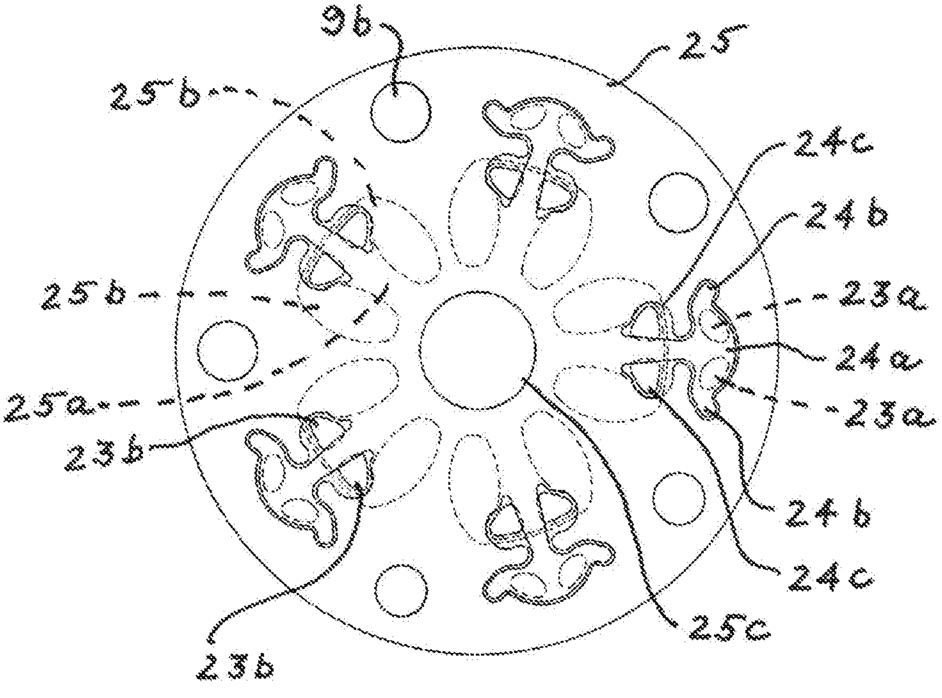

FIG. 12C

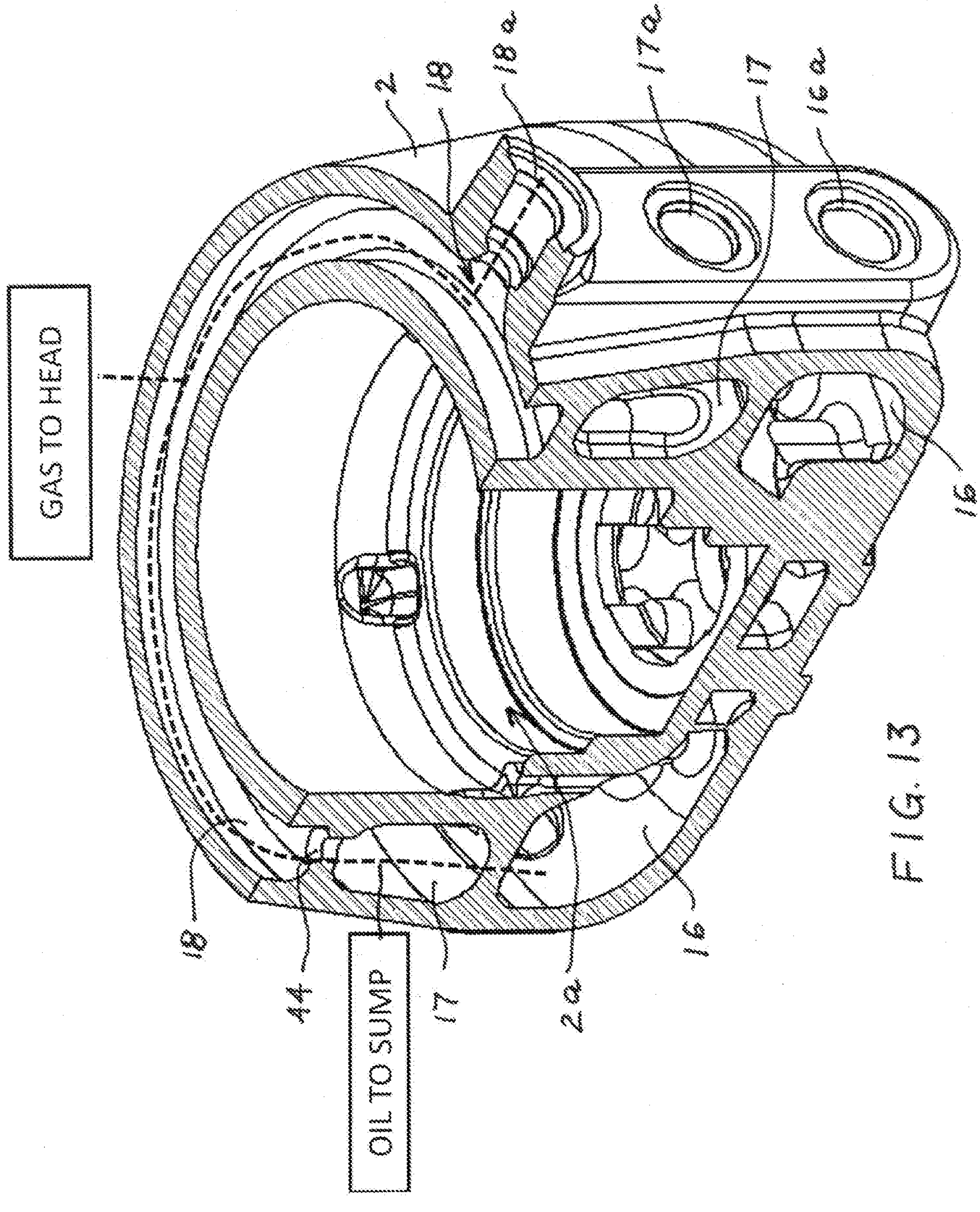

FIG. 13

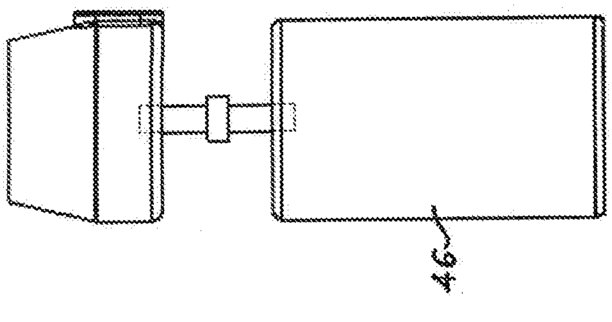

FIG. 14A

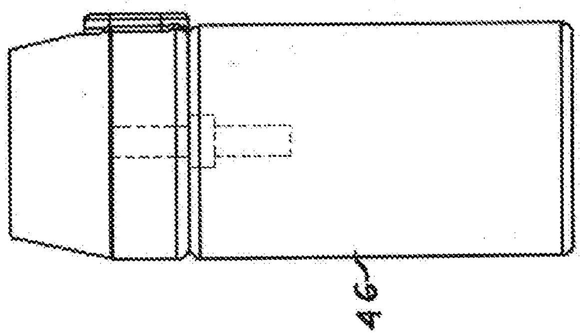

FIG. 14B

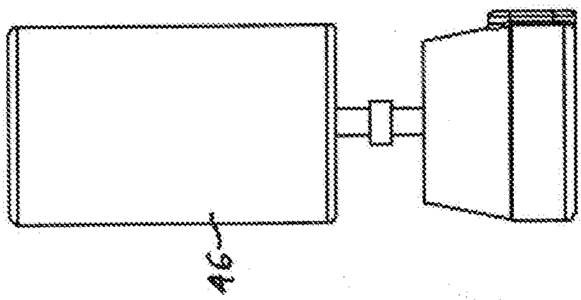

FIG. 14C

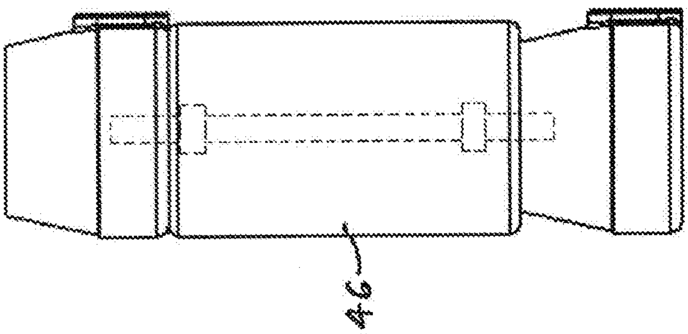

FIG. 14D

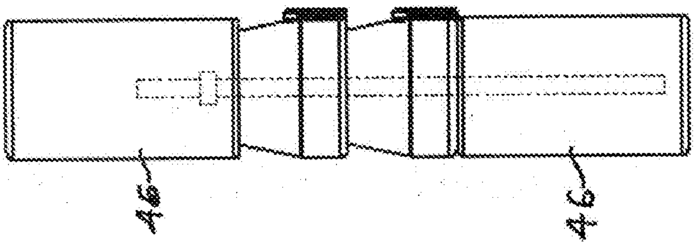

FIG. 14E

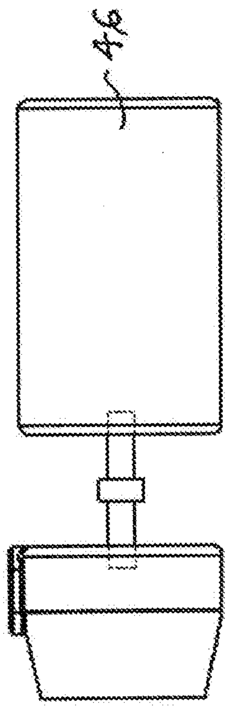

FIG. 14F

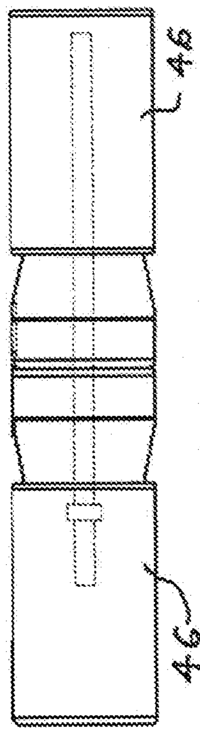

FIG. 14G

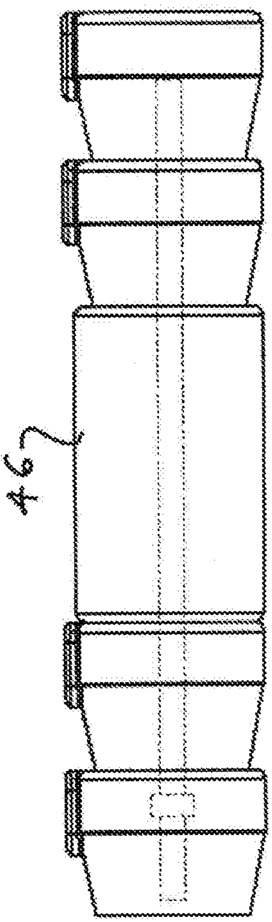

FIG. 14H

Semi-hermetic compressor
external isometric view

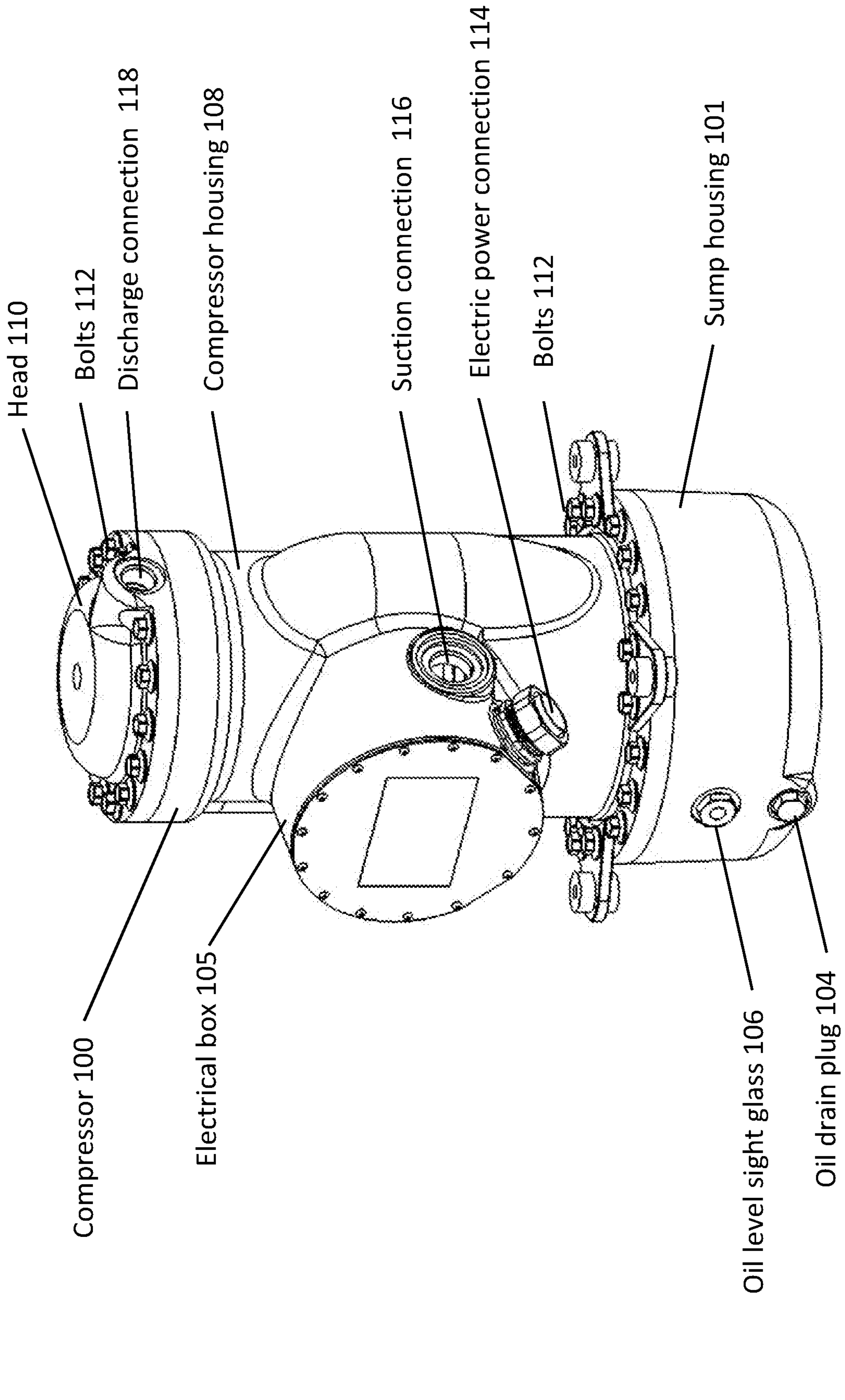

Figure 15

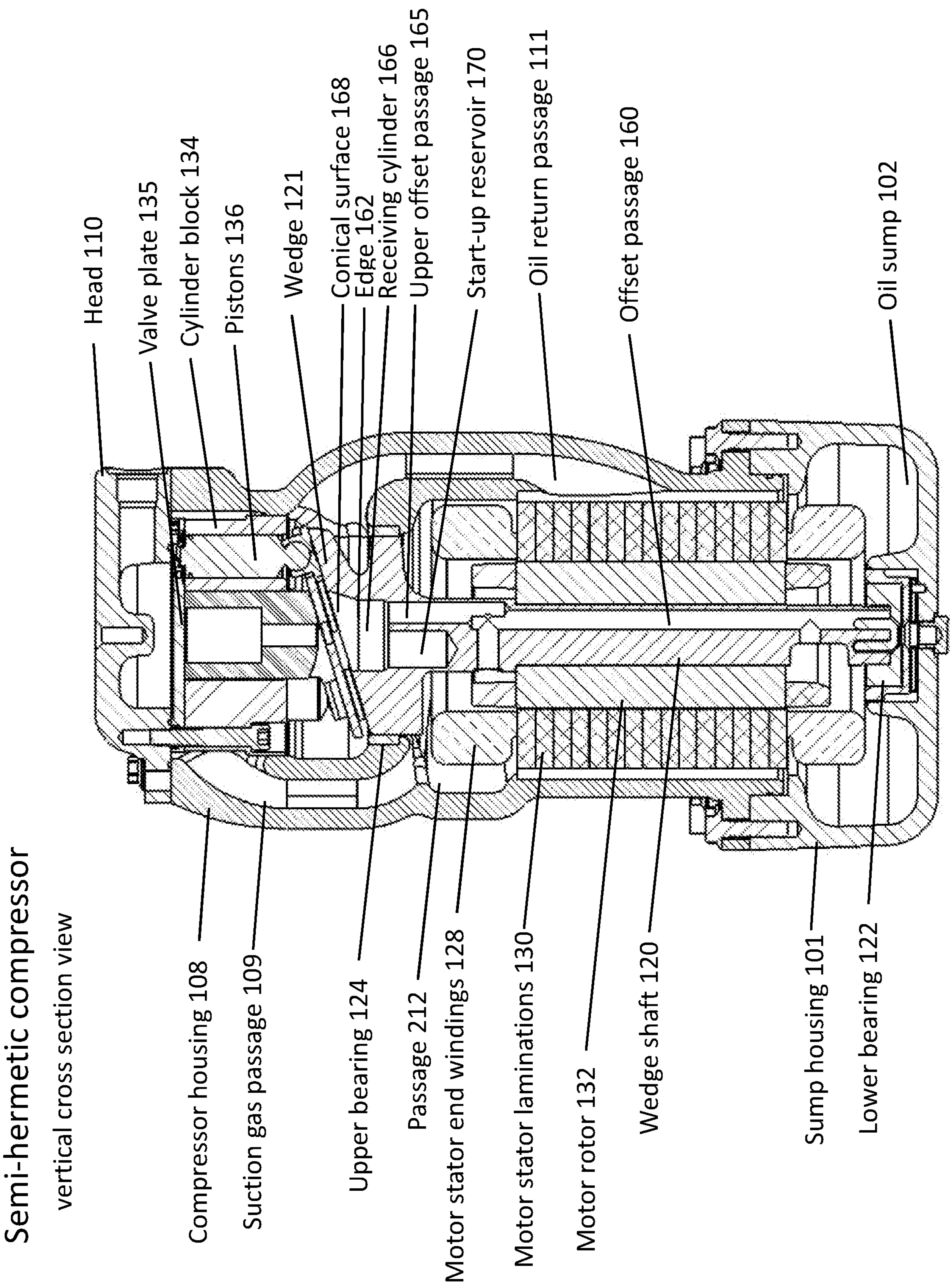
Semi-hermetic compressor
vertical cross section view
Head 110
Valve plate 135
Cylinder block 134
Pistons 136
Wedge 121
Conical surface 168
Edge 162
Receiving cylinder 166
Upper offset passage 165
Start-up reservoir 170
Oil return passage 111
Offset passage 160
Oil sump 102
Compressor housing 108
Suction gas passage 109
Upper bearing 124
Passage 212
Motor stator end windings 128
Motor stator laminations 130
Motor rotor 132
Wedge shaft 120
Sump housing 101
Lower bearing 122

Figure 16

Piston retention
vertical cross section view

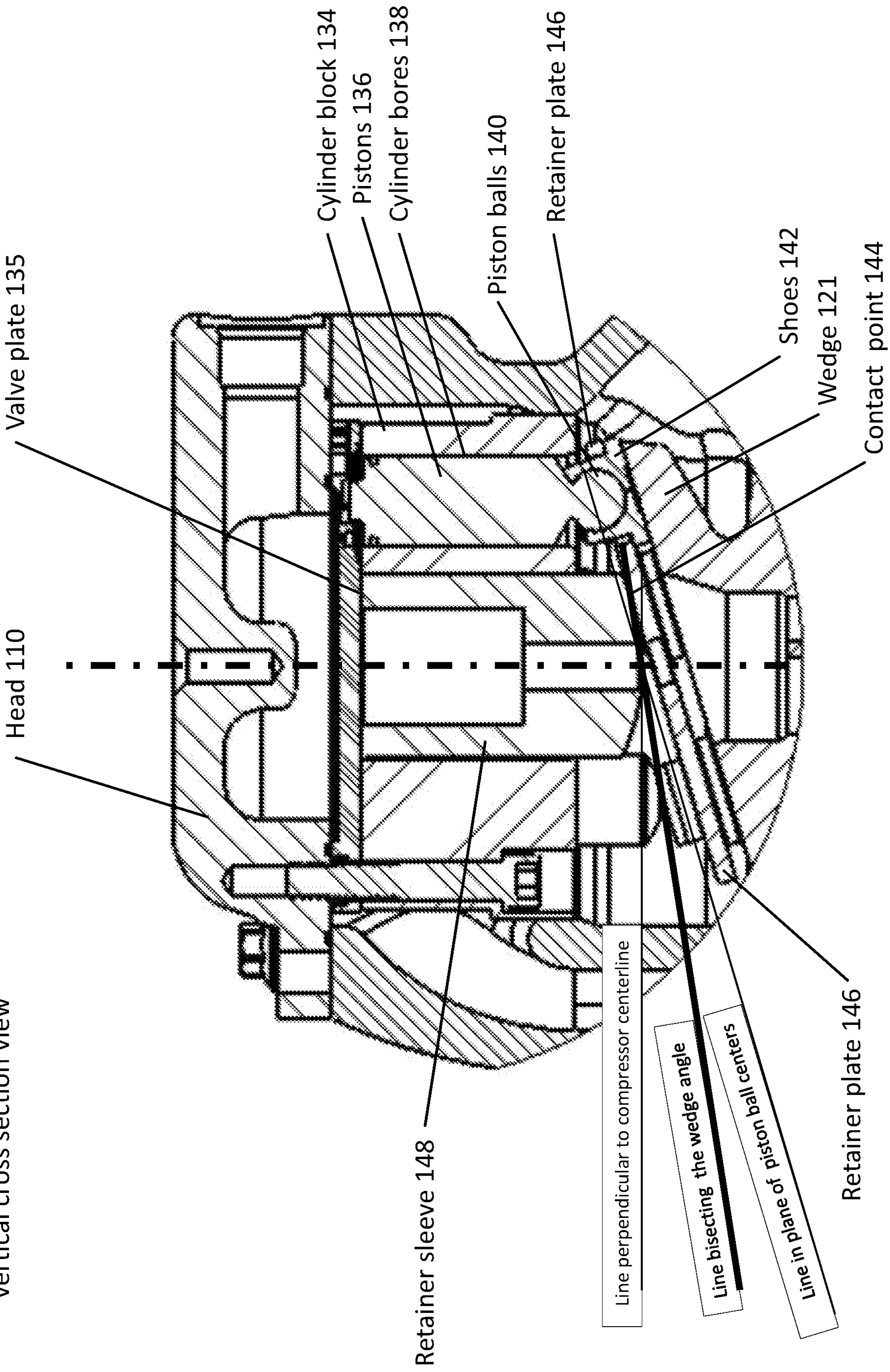

Figure 17

Valves during suction stroke
Vertical cross section through valves

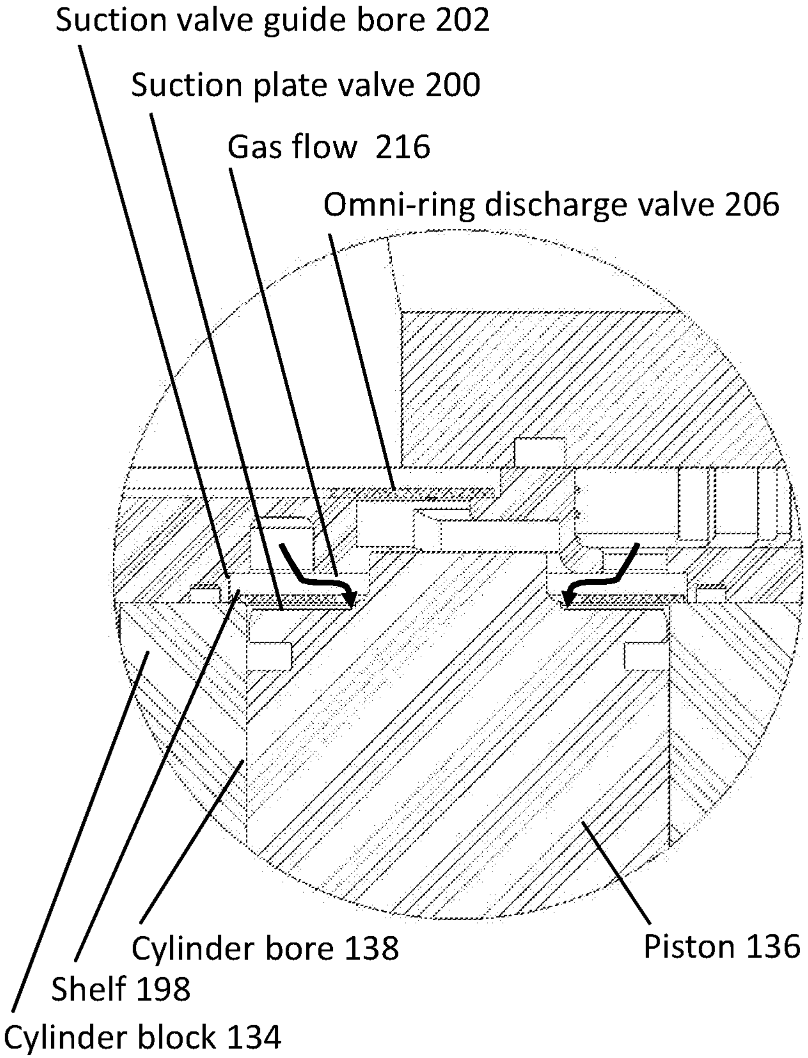

Figure 18

Valves during discharge stroke
Vertical cross section through valves

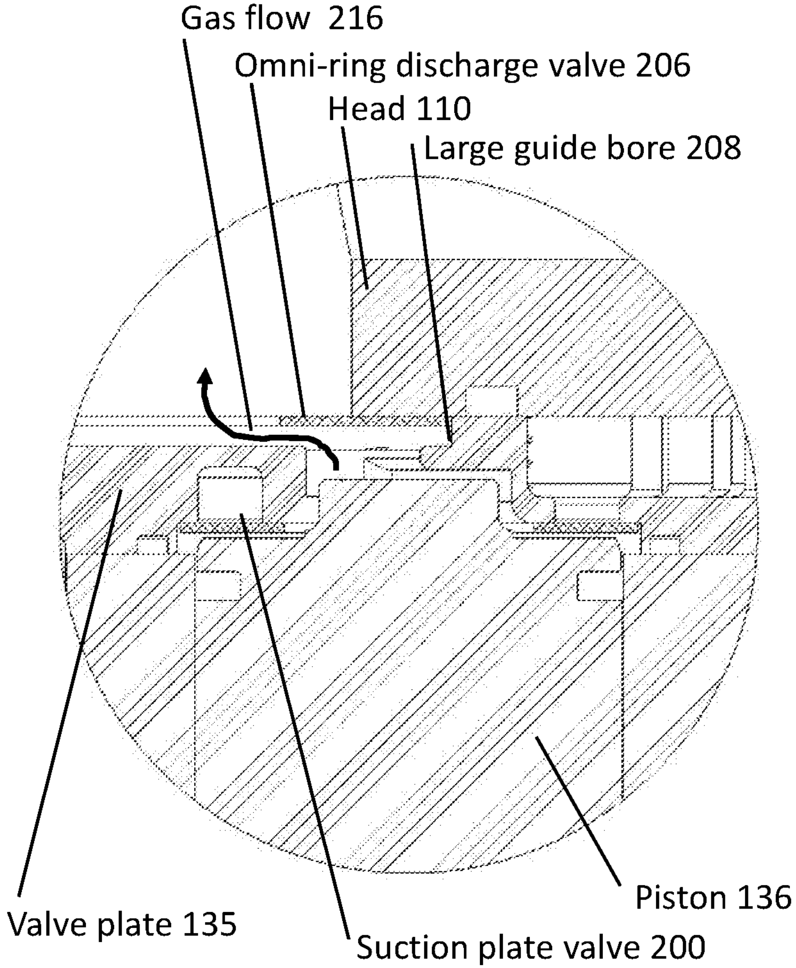

Figure 19

Liquid separation from suction gas flow

horizontal cross section through suction connection

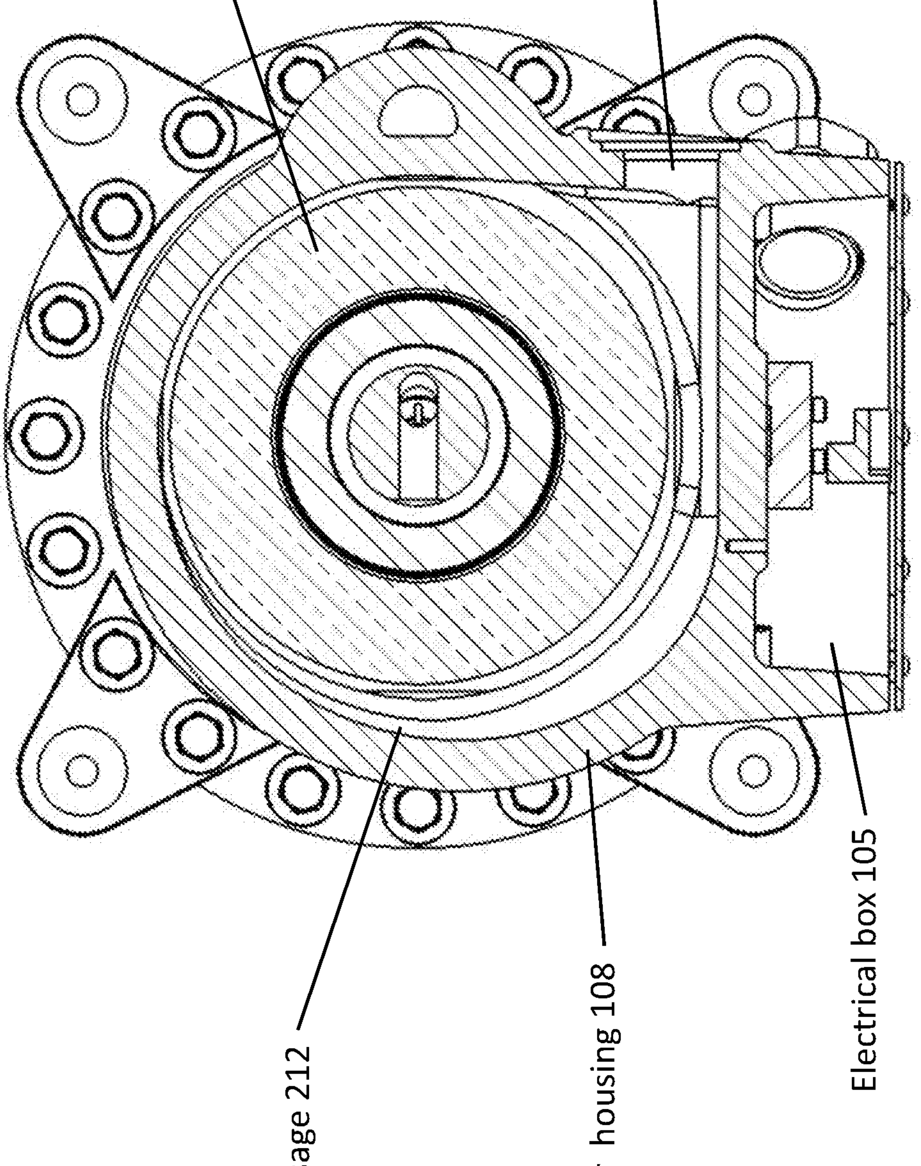

Figure 20

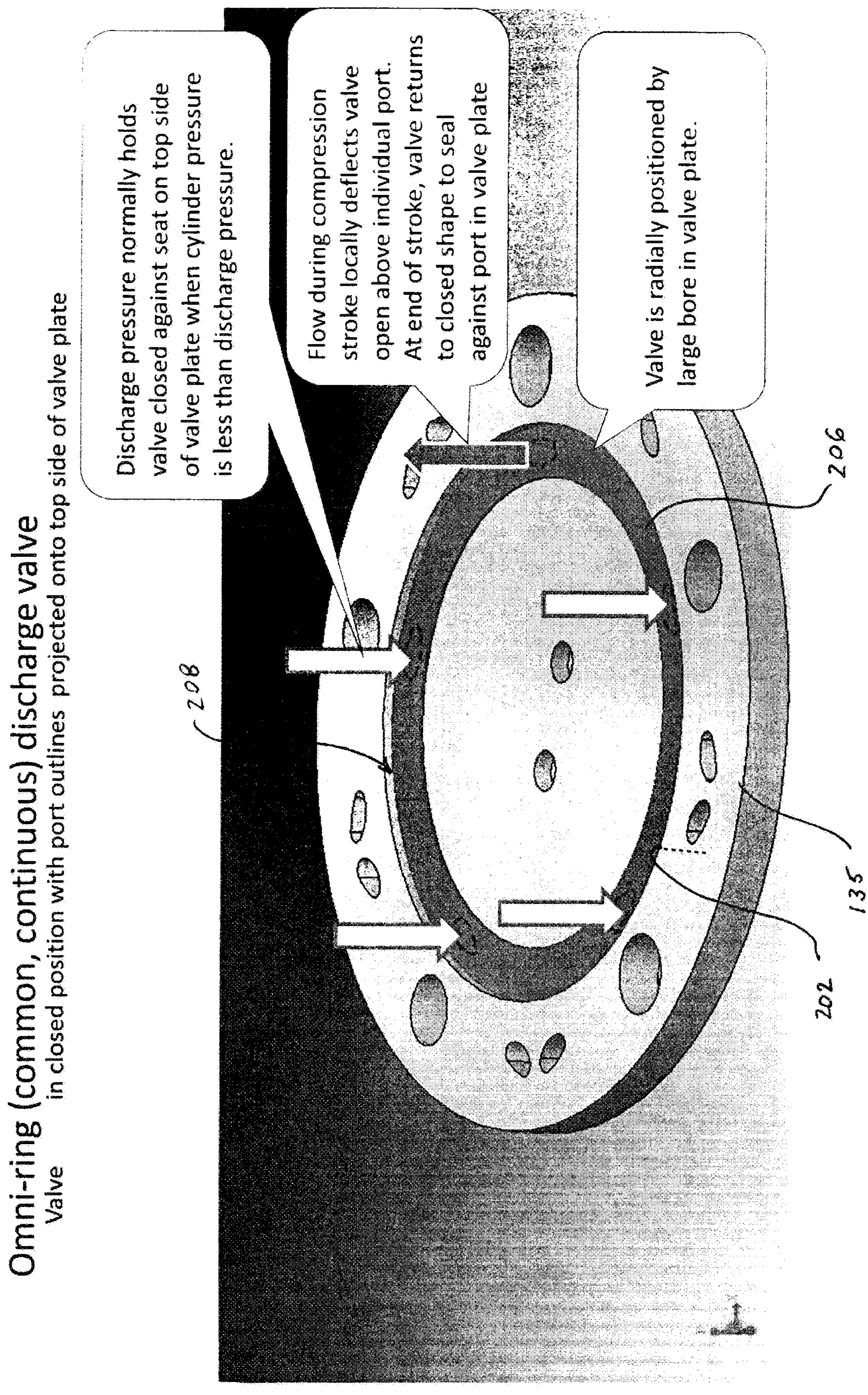
Omni-ring (common, continuous) discharge valve
Valve in closed position with port outlines projected onto top side of valve plate
Discharge pressure normally holds valve closed against seat on top side of valve plate when cylinder pressure is less than discharge pressure.
Flow during compression stroke locally deflects valve open above individual port. At end of stroke, valve returns to closed shape to seal against port in valve plate
Valve is radially positioned by large bore in valve plate.
208
206
135
202

FIG. 21

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US 17/22512

A. CLASSIFICATION OF SUBJECT MATTER
IPC(8) - F04B 39/02, F16N 7/36, F04B 39/08, F04B 27/08 (2017.01)
CPC - F04B 39/023, F04B 39/0253, F16N 7/366, F16N 7/363, F16N 7/36, F04B 39/08, F04B 27/0839

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

B. FIELDS SEARCHED

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

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

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X --- Y	US 2014/0328702 A1 (RIEDIGER et al.) 06 November 2014 (06.11.2014), entire document, especially Fig 1, 1A, 10A-10B; 11A-11B; 12A-12B; 14A-14H; para [0073], [0076], [0085], [0087], [0089], [0096]-[0097]; claim 7	1-2, 4-8 ---------- 3
Y	US 2013/0259713 A1 (HAYASHI et al.) 03 October 2013 (03.10.2013), entire document, especially Fig 1; para [0014], [0016], [0019]-[0020], [0026]	3
A	US 6,368,073 B1 (KANAI et al.) 09 April 2002 (09.04.2002), entire document	1-8
A	US 5,380,168 A (KIMURA et al.) 10 January 1995 (10.01.1995), entire document	1-8
A	US 5,286,173 A (TAKENAKA et al.) 15 February 1994 (15.02.1994), entire document	1-8

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

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
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Date of the actual completion of the international search
09 May 2017

Date of mailing of the international search report
09 JUN 2017

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