



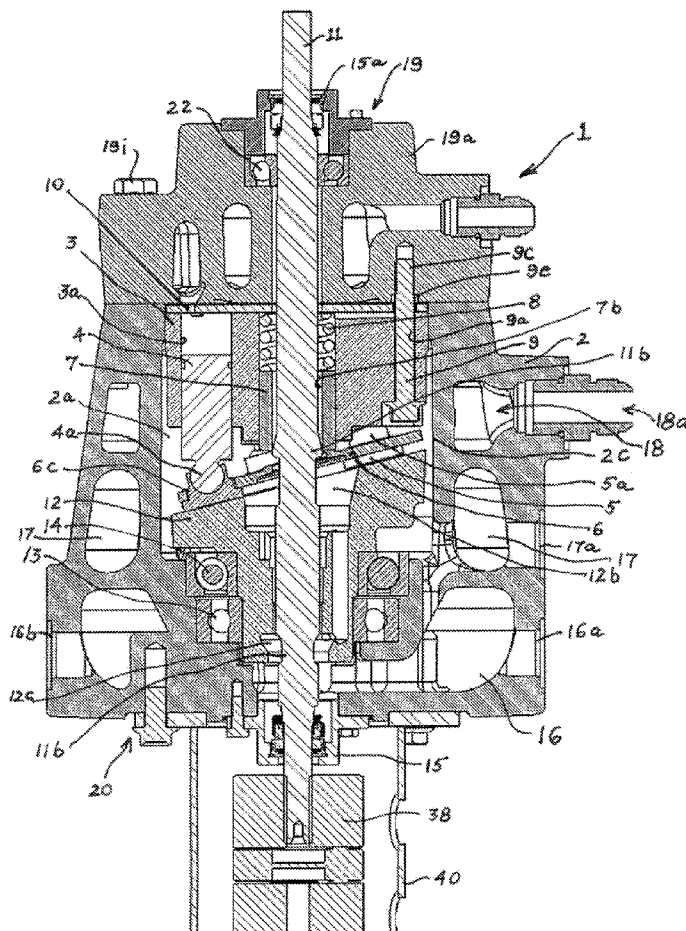
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(19) **United States**(12) **Patent Application Publication**
Riediger et al.(10) **Pub. No.: US 2014/0328702 A1**(43) **Pub. Date: Nov. 6, 2014**(54) **AXIAL PISTON HIGH PRESSURE
COMPRESSOR/PUMP****Publication Classification**(71) Applicant: **ECOTHERMICS CORPORATION**,
Peoria, IL (US)(51) **Int. Cl.**
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CPC **F04B 27/109** (2013.01)
USPC **417/269; 417/434**(73) Assignee: **Eco Thermics Corporation**, East
Peoria, IL (US)(57) **ABSTRACT**(21) Appl. No.: **14/362,448**(22) PCT Filed: **Dec. 6, 2012**(86) PCT No.: **PCT/US2012/068191**

§ 371 (c)(1),

(2), (4) Date: **Jun. 3, 2014****Related U.S. Application Data**(60) Provisional application No. 61/567,884, filed on Dec.
7, 2011.

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 center positioning mechanism for the retainer plate in which the slipper shoes placed on the wedge have a portion extending through a respective hole in the retainer plate, with each of the slipper shoes engaging a point on the side of its respective hole to restrain the retainer plate from movement parallel to the planar angular surface of the wedge. Also described is an oil separation system for removing oil from the gas to be compressed such that the removed oil drains into the oil sump. An oil lubrication system has a cavity around the drive shaft to receive oil from the sump, and disburse the oil through a passageway to discharge ports in the wedge to disburse oil on the planar angular surface of the wedge.



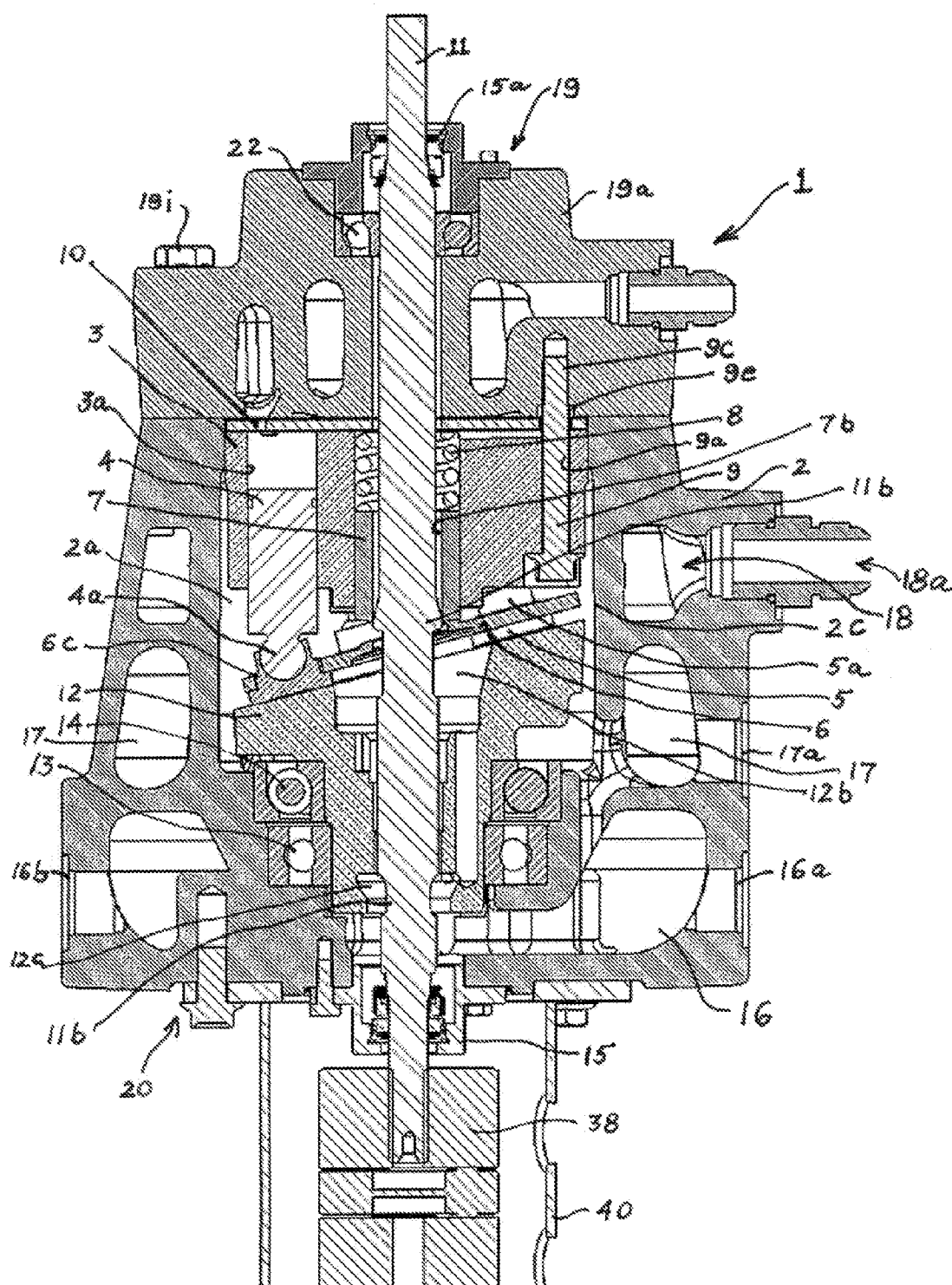


FIG. 1

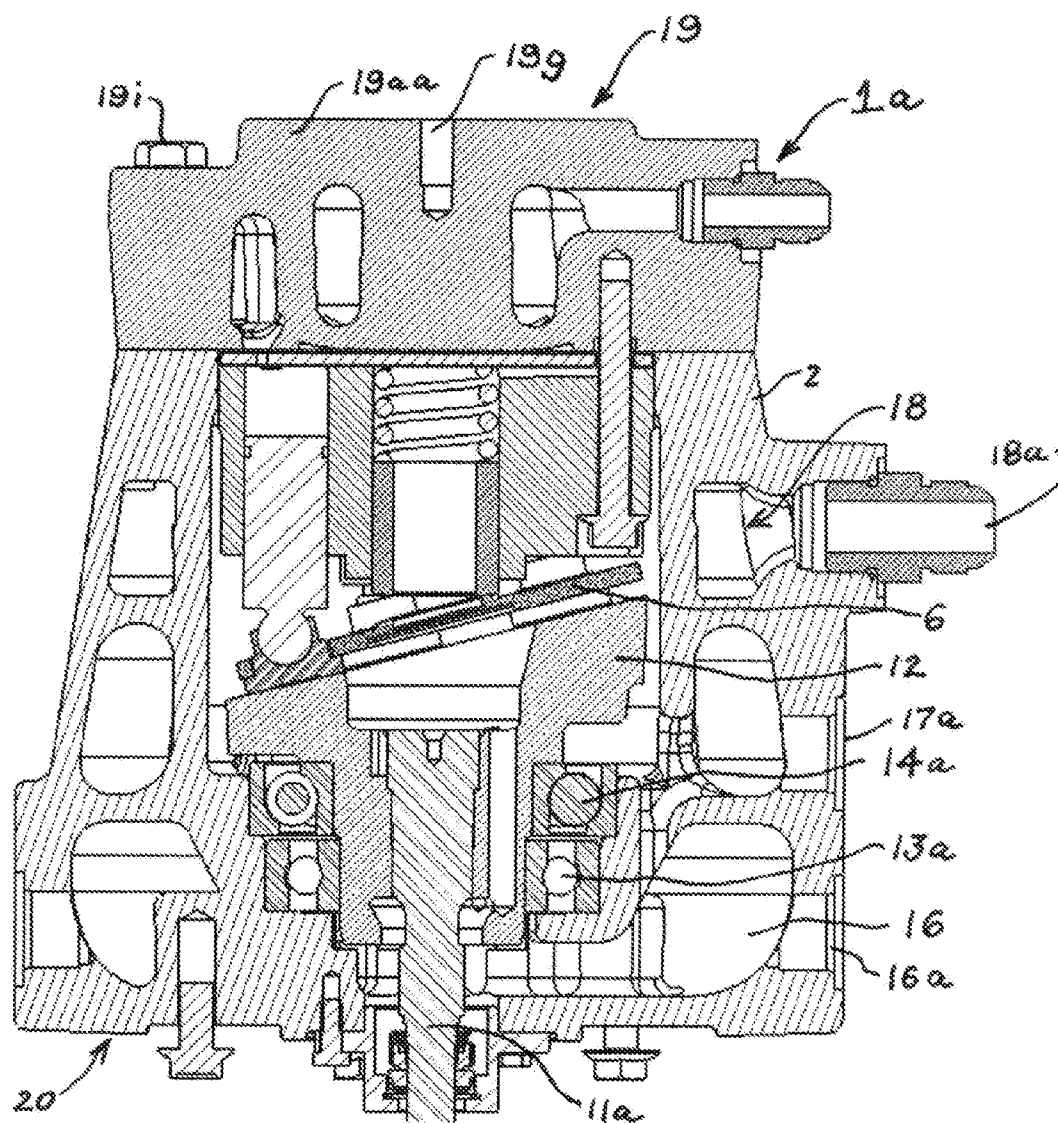


FIG. 1A

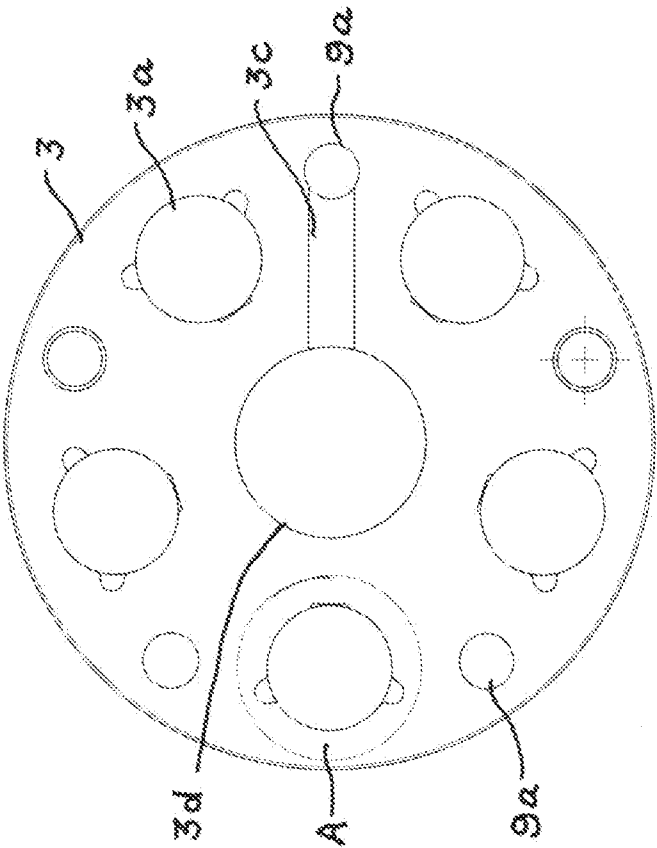


FIG. 2A

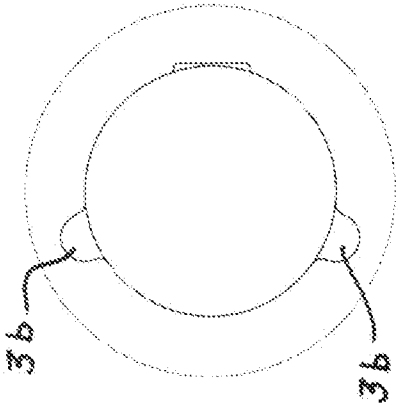
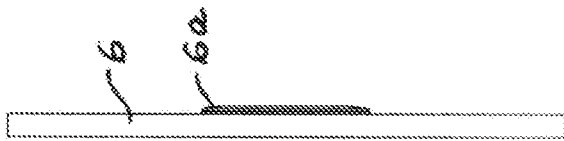
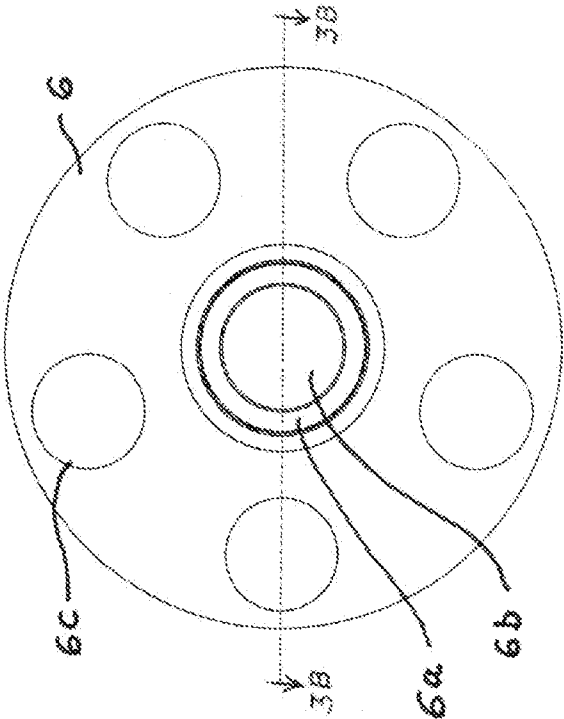
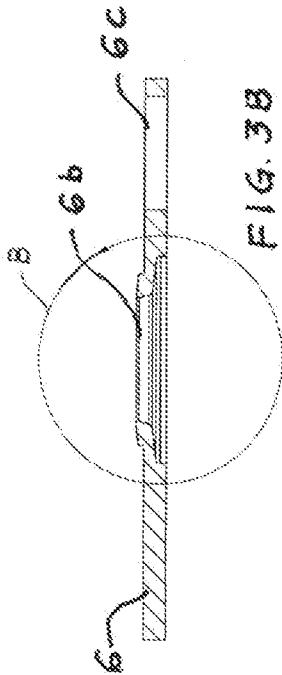


FIG. 2B



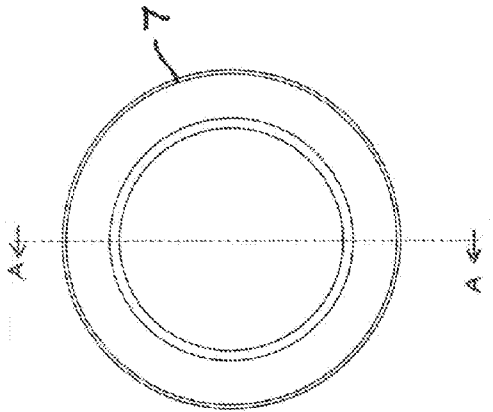


FIG. 4A

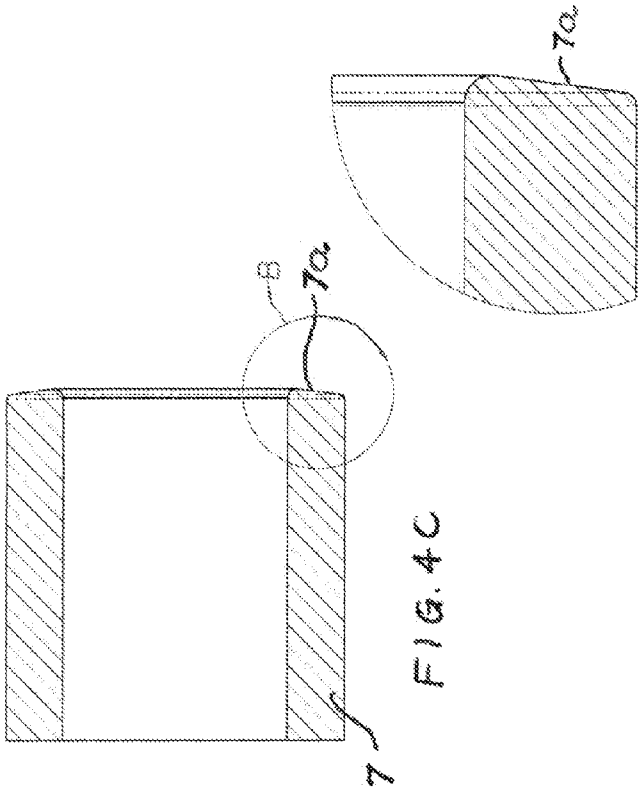


FIG. 4C

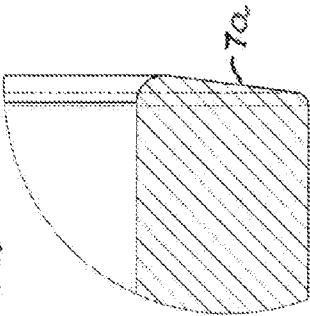


FIG. 4D

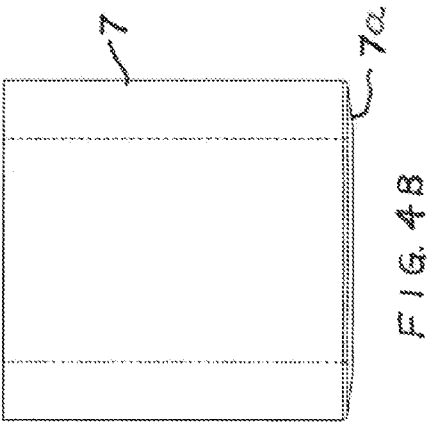


FIG. 4B

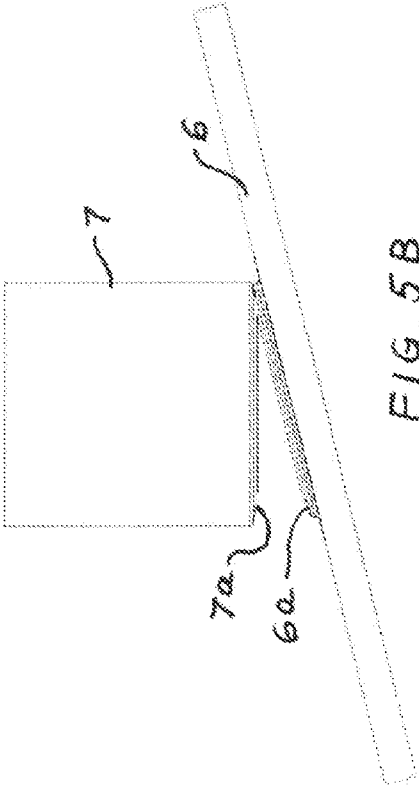
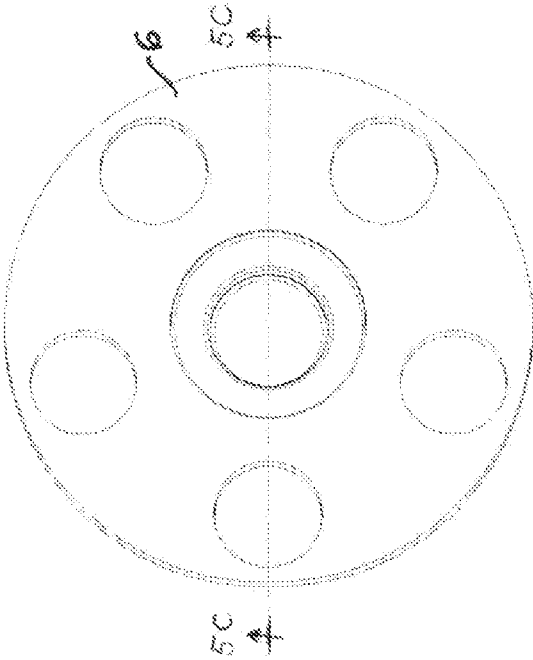


FIG. 5A

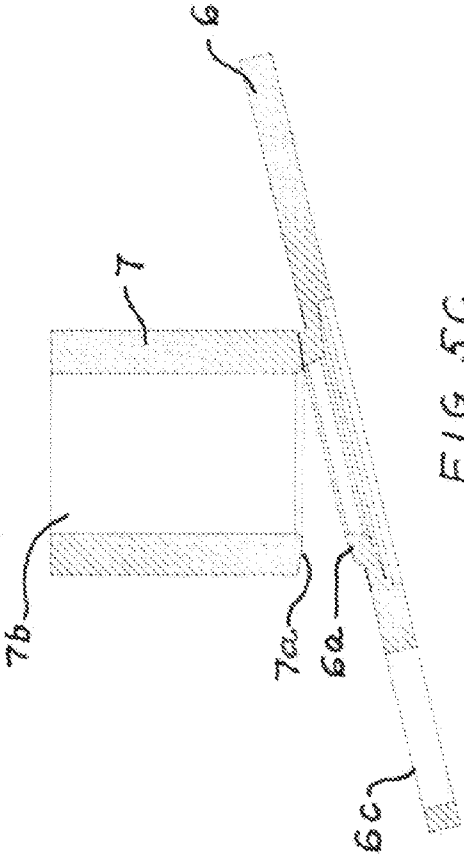
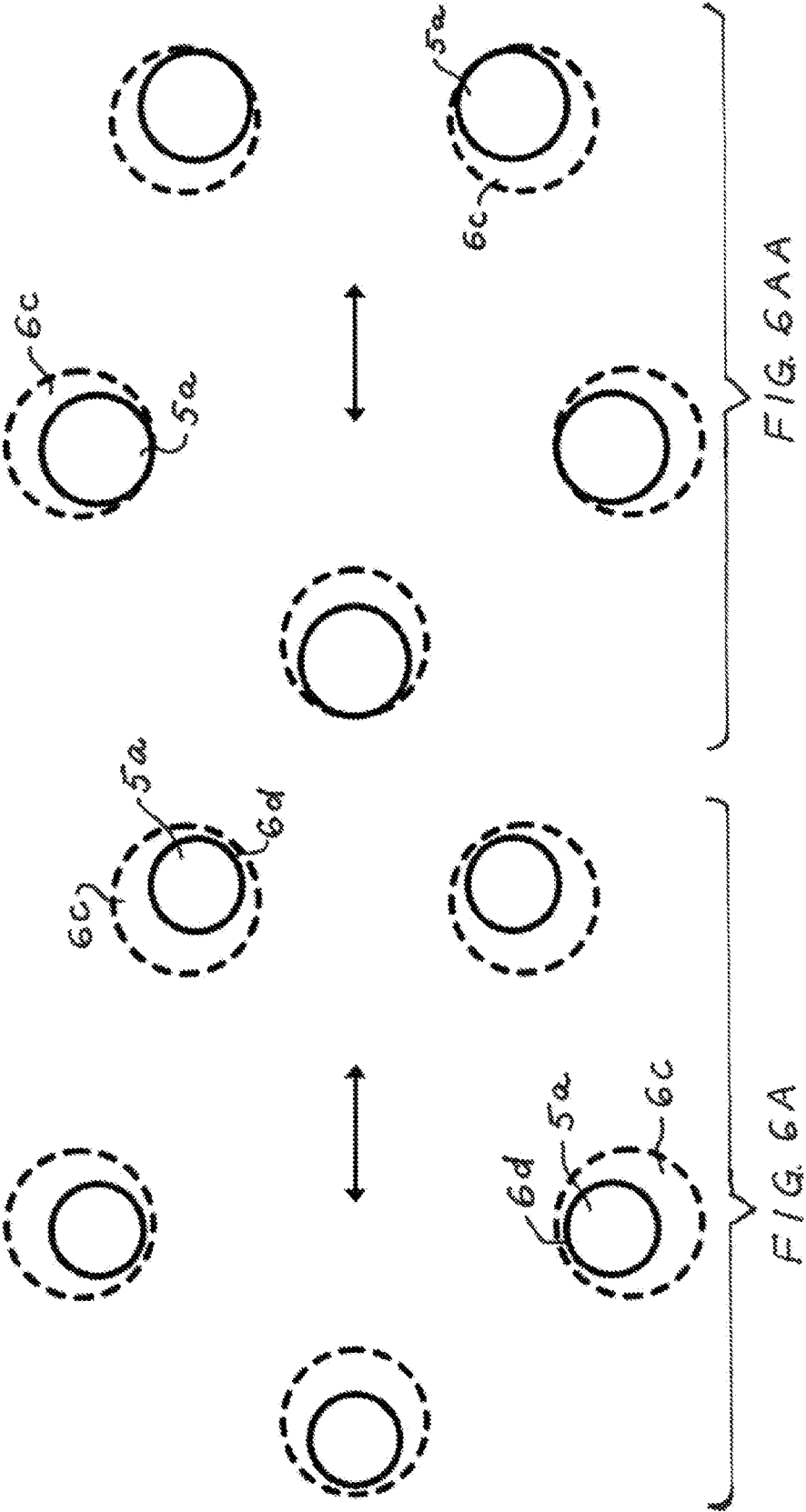


FIG. 5C



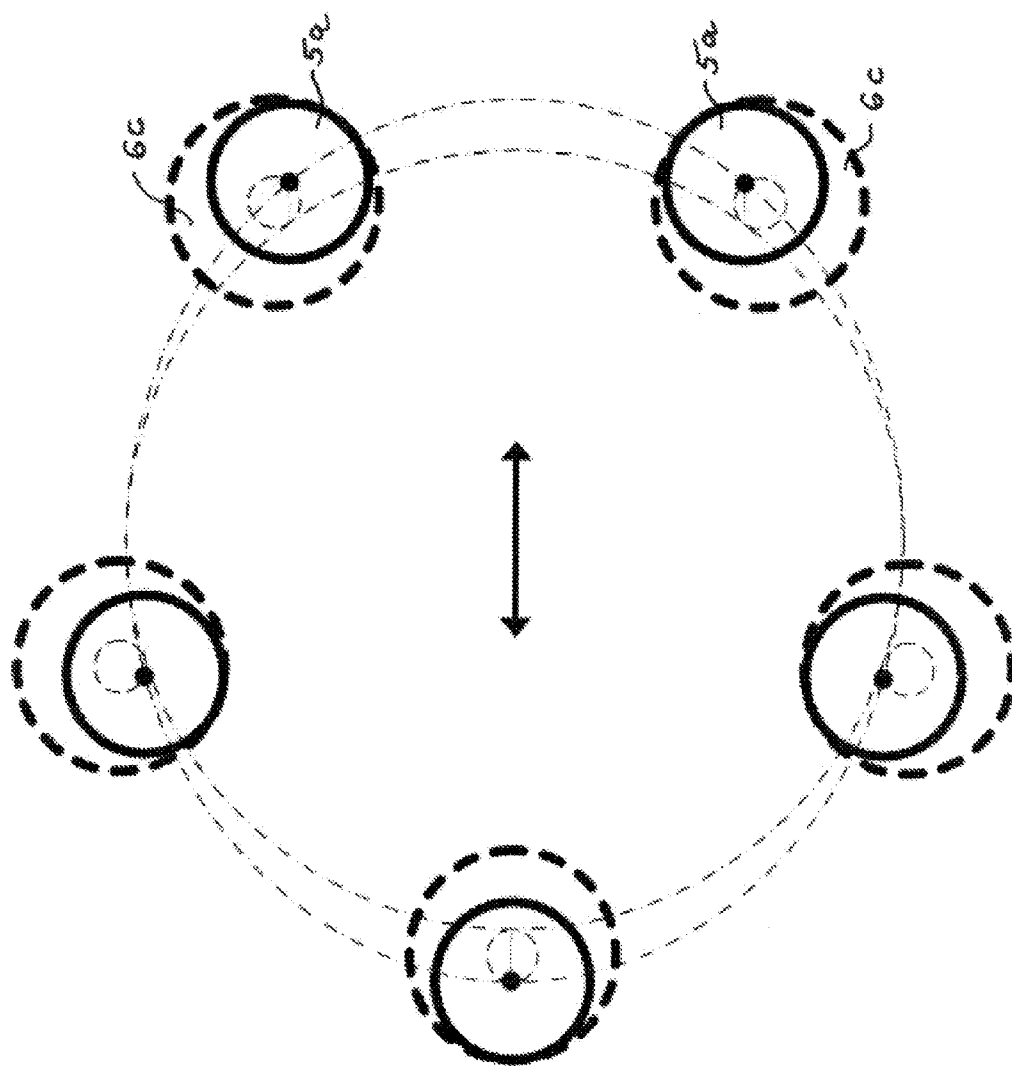


FIG. 6B

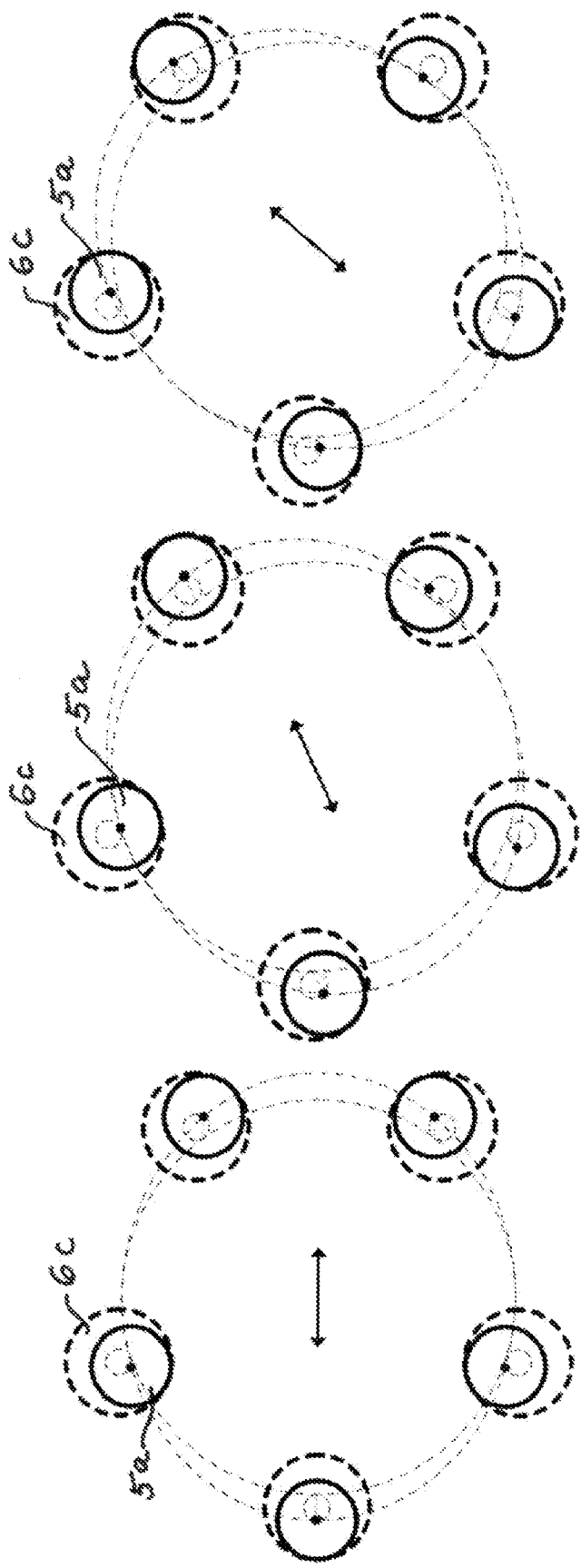


FIG. 6C

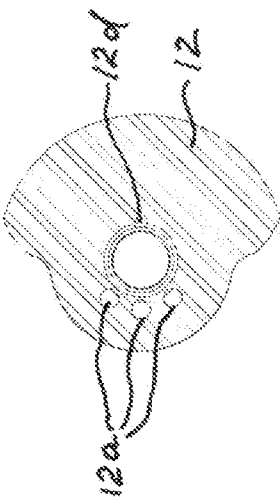


FIG. 7D

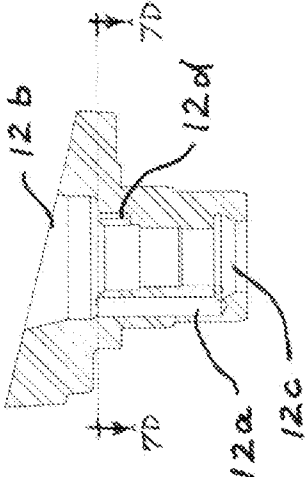


FIG. 7C

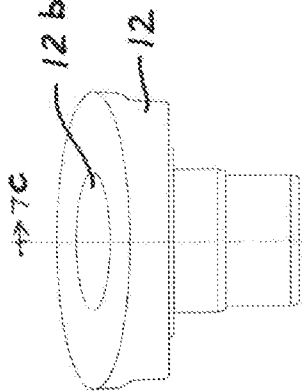


FIG. 7B

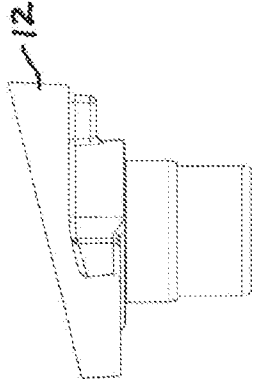


FIG. 7A

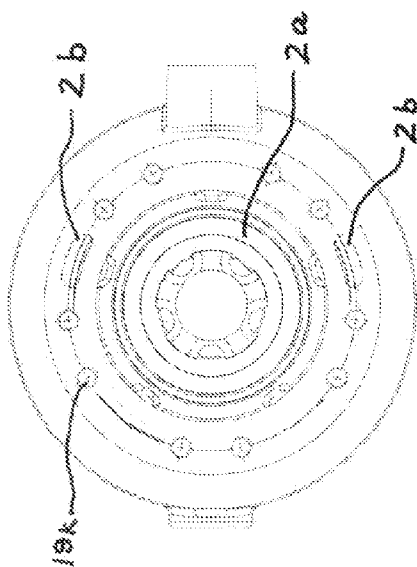


FIG. 8C

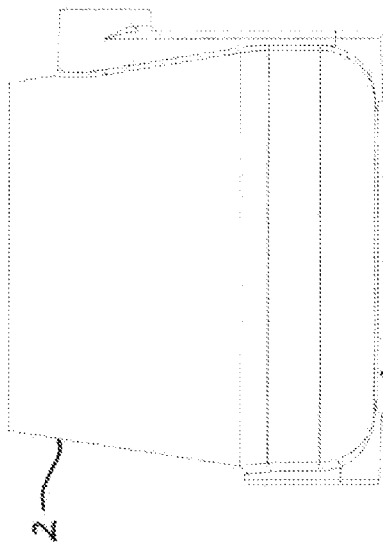


FIG. 8A

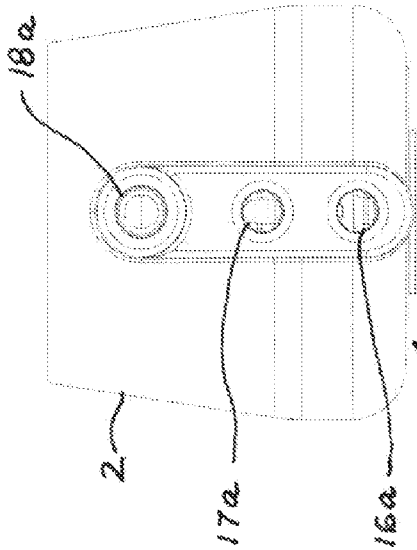
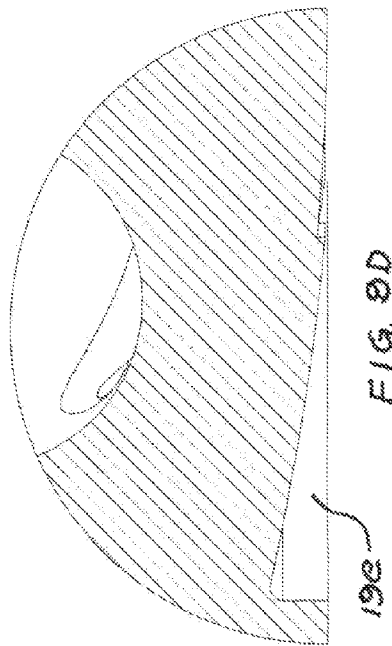
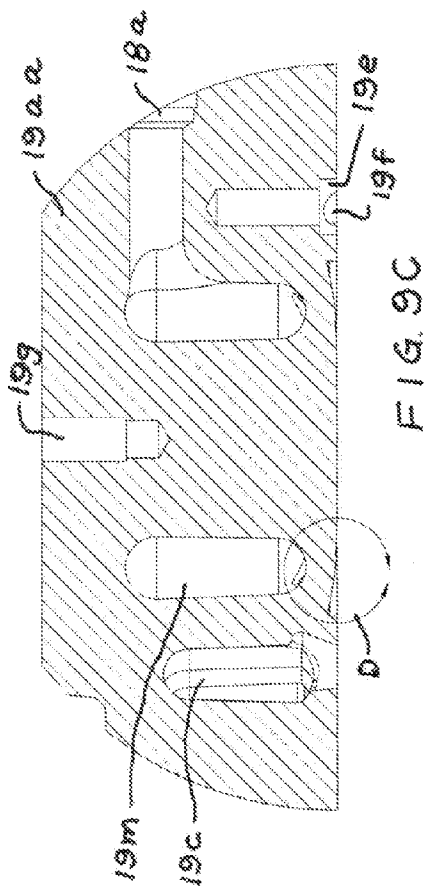
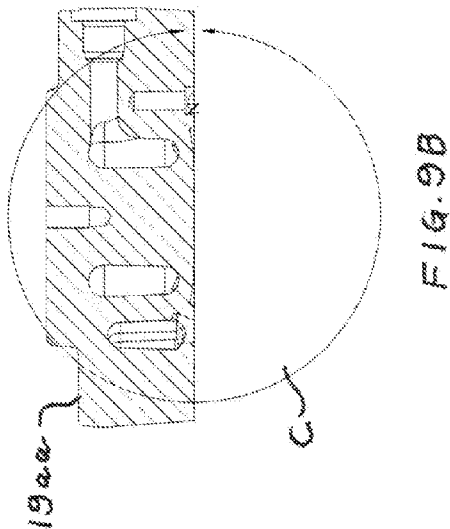
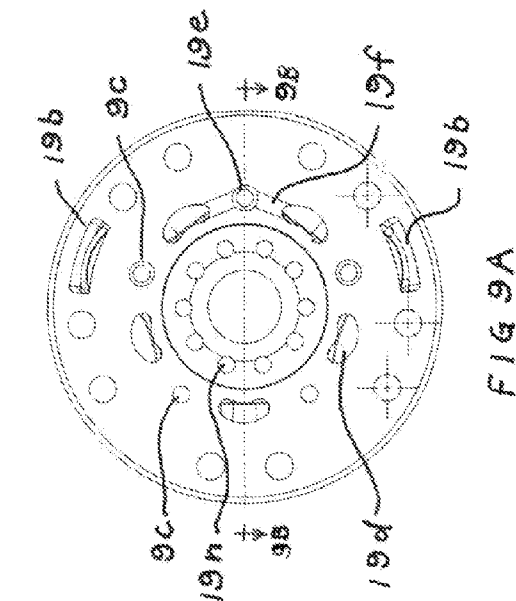


FIG. 8B



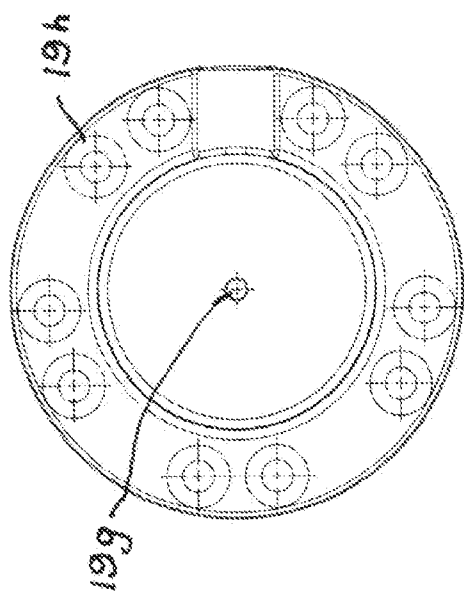


FIG. 9F

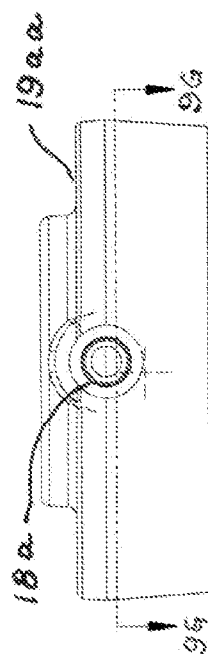


FIG. 9E

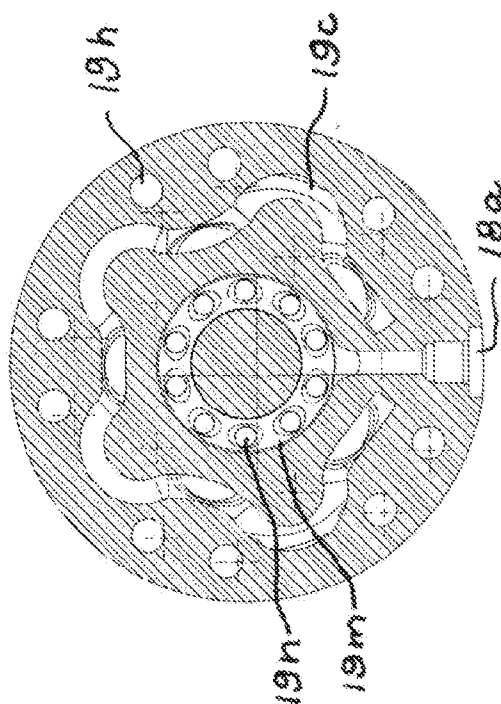


FIG. 9G

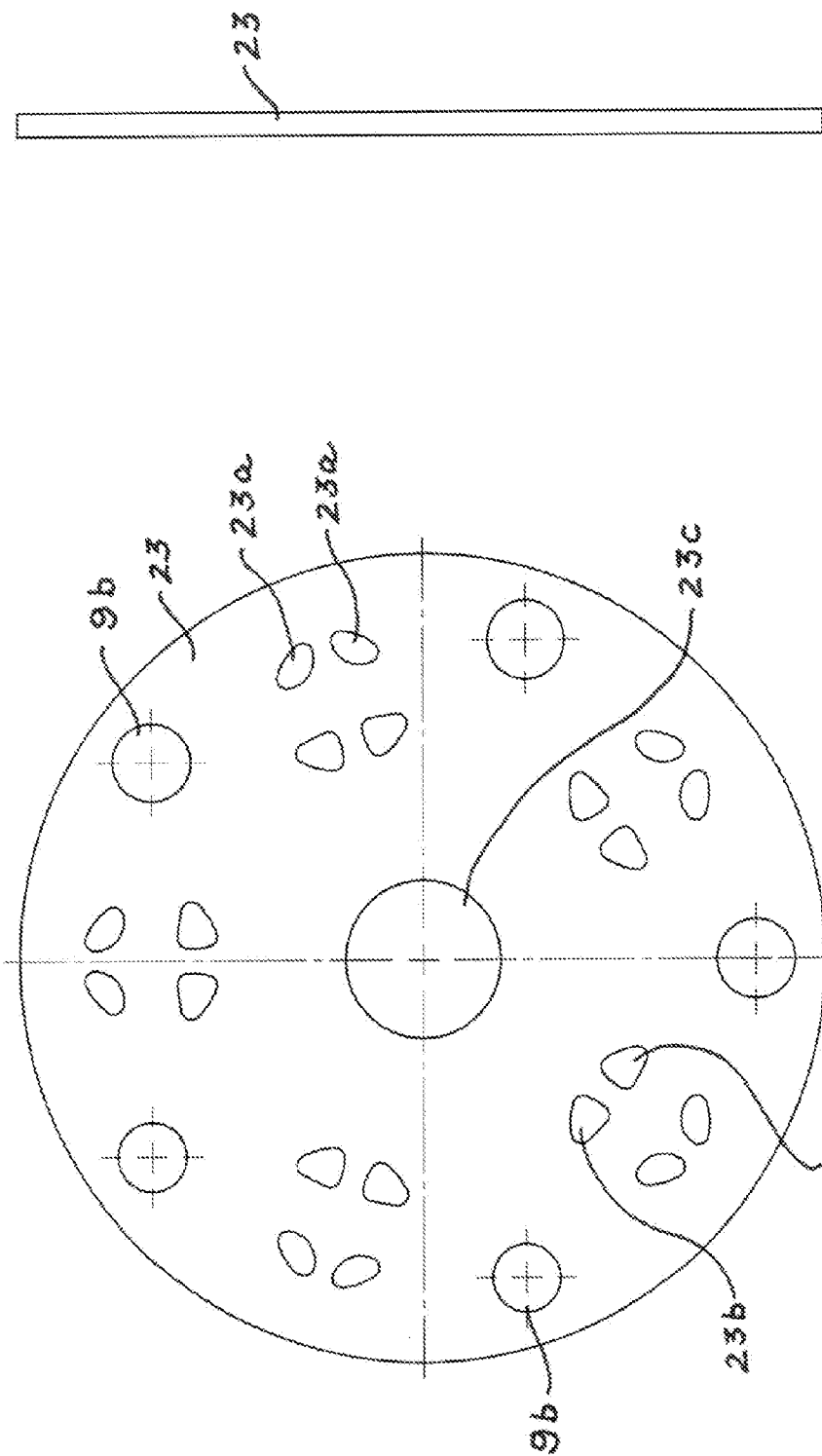


FIG. 10A



FIG. 10B

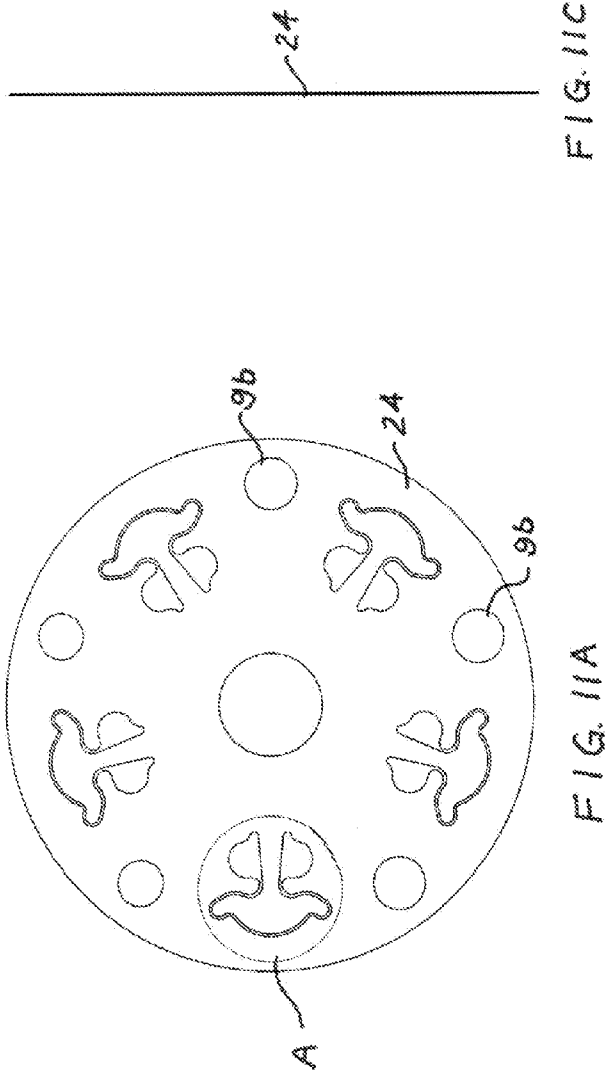


FIG. IIC

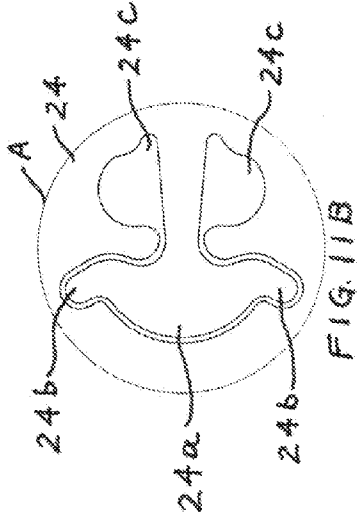


FIG. IIB

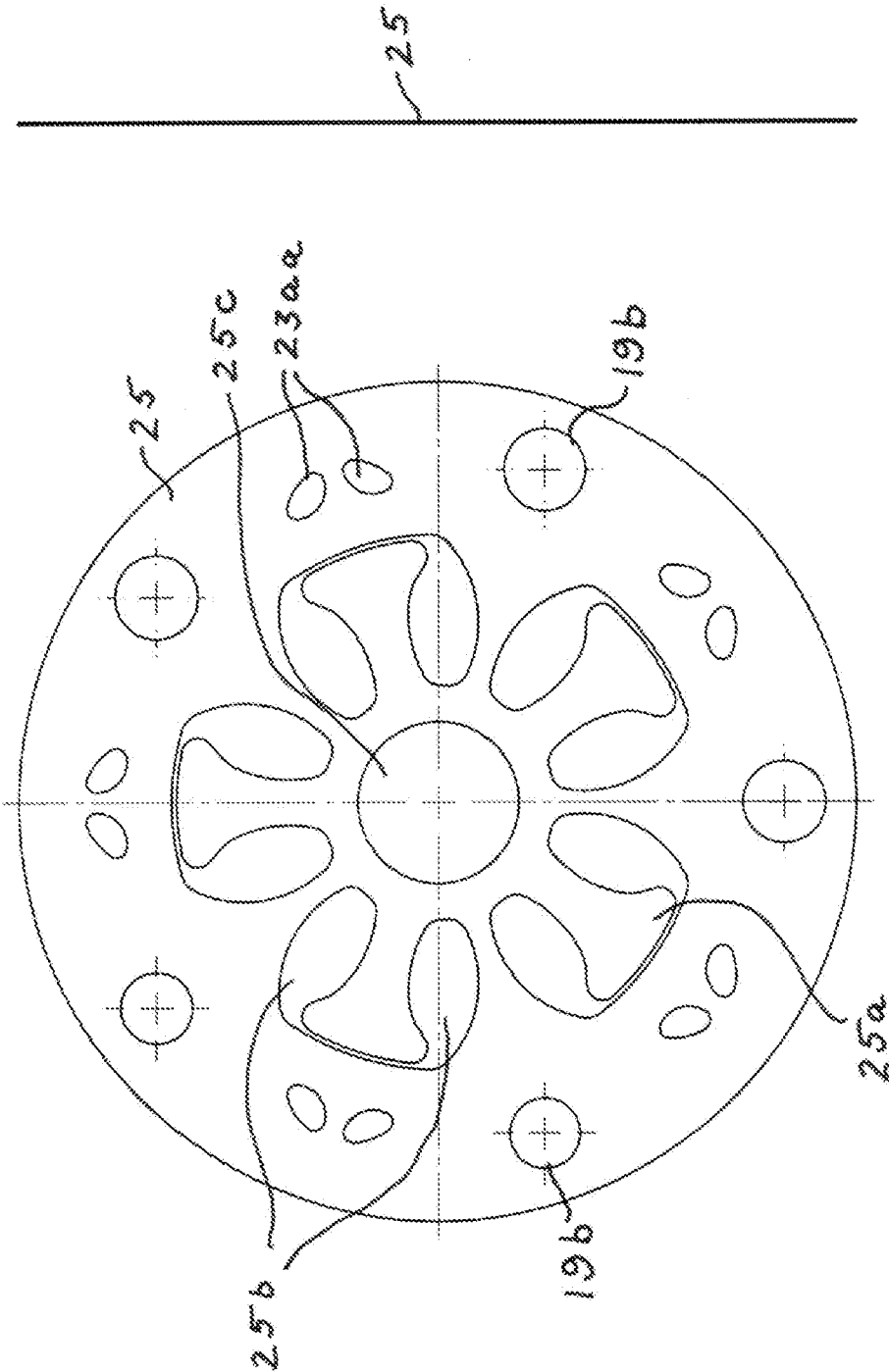


FIG. 12B

FIG. 12A

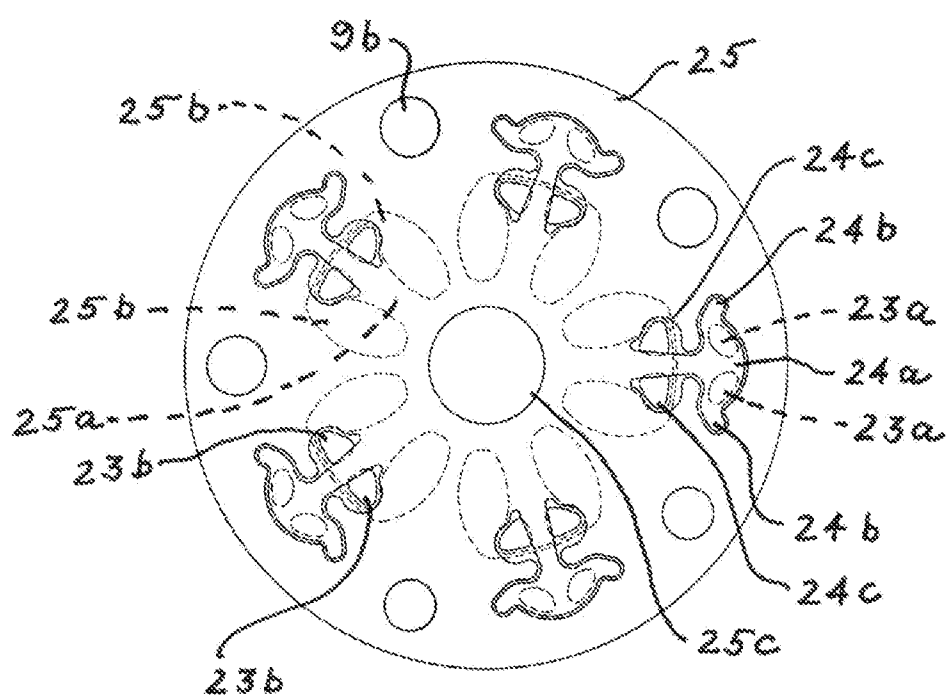
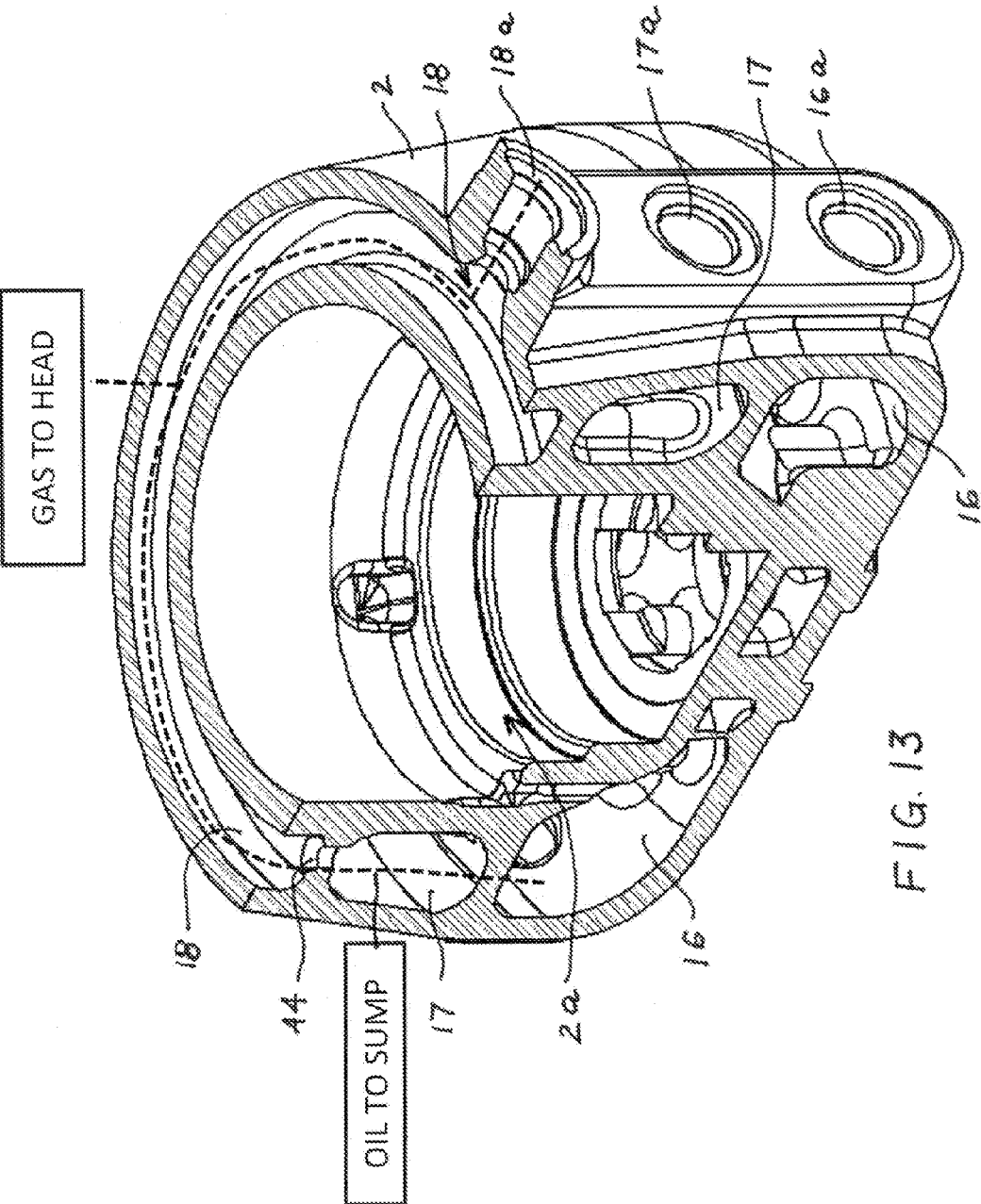


FIG. 12C



AXIAL PISTON HIGH PRESSURE COMPRESSOR/PUMP

I. CROSS REFERENCE TO RELATED APPLICATION

[0001] This application is based on and claims priority of U.S. provisional patent application 61/467,884 filed Dec. 7, 2011.

II. FIELD OF THE INVENTION

[0002] 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

[0003] 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 CO₂ as the working gas. There are several primary technology and design hurdles for improving CO₂ gas compressors; these are: (1) higher pressures required, (2) unique properties of CO₂, (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.

[0004] CO₂ 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 CO₂ 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 separa-

tion 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.

[0005] 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.

[0006] 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 CO₂ 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.

[0007] 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.

[0008] 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 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.

[0009] 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.

[0010] 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.

[0011] 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.

[0012] 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.

[0013] 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.

[0014] 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.

[0015] 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.

[0016] 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.

[0017] 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 CO₂. 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.

[0018] 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 manifold 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.

[0019] 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.

[0020] 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.

[0021] 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.

[0022] 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.

IV. BRIEF DESCRIPTION OF THE DRAWINGS

[0023] FIG. 1 is a left side cross sectional view of the inventive compressor with the shaft extending through the head of the compressor.

[0024] 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.

[0025] FIG. 2A is a top plan view of the cylinder block

[0026] FIG. 2B is an enlarged view of the encircled area A of FIG. 2A

[0027] FIG. 3A is a top plan view of the retainer plate.

[0028] FIG. 3B is a cross sectional view taken along line 3B-3B of the retainer plate of FIG. 3A.

[0029] FIG. 3C is an end view of the retainer plate of FIG. 3A.

[0030] FIG. 3D is an enlarged sectional view of detail area B encircled in FIG. 3B

[0031] FIG. 4A is a top plan view of the retainer sleeve.

[0032] FIG. 4B is front elevation view of the retainer sleeve of FIG. 4A.

[0033] FIG. 4C is a cross sectional view taken along line A-A of FIG. 4A of the retainer sleeve.

[0034] FIG. 4D is an enlarged view of the detail area B encircled in FIG. 4C illustrating the retainer sleeve nose.

[0035] FIG. 5A is a top plan view of the assembled retainer plate and retainer sleeve.

[0036] FIG. 5B is a front elevation view of the assembled retainer plate and retainer sleeve of FIG. 5A.

[0037] FIG. 5C is a cross sectional view taken along line 5C-5C of FIG. 5A.

[0038] FIG. 6A 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.

[0039] FIG. 6AA 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.

[0040] 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.

[0041] 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.

[0042] FIG. 7A is a front elevation view of the wedge.

[0043] FIG. 7B is side elevation view of the wedge.

[0044] FIG. 7C is a cross sectional view taken along line 7C-7C of FIG. 7B.

[0045] FIG. 7D is a cross section view taken along line 7D-7D of FIG. 7C.

[0046] FIG. 8A is a left side elevation view of the compressor housing.

[0047] FIG. 8B is a front elevation view of the compressor housing.

[0048] FIG. 8C is a top plan view of the compressor housing.

[0049] FIG. 9A is a bottom view of the head with front elevation to the right

[0050] FIG. 9B is an elevation cross sectional view taken along line 9B-9B of FIG. 9A.

[0051] FIG. 9C is an enlarged view of the encircled area C in FIG. 9B.

[0052] FIG. 9D is an enlarged view of the encircled area D in FIG. 9C.

[0053] FIG. 9E is a front elevation view of the head.

[0054] FIG. 9F is a top view of the head in FIG. 9E.

[0055] FIG. 9G is a plan cross sectional view taken across line 9G-9G of FIG. 9E.

[0056] FIG. 10A is a top plan view of the port plate.

[0057] FIG. 10B is an end view of the port plate in FIG. 10A

[0058] FIG. 11A is a top plan view of the suction reed valve plate and the suction valves.

[0059] FIG. 11B is an enlarged view of the encircled area A in FIG. 11A illustrating the suction valve configuration.

[0060] FIG. 11C is an end view of suction valve plate in FIG. 11A.

[0061] FIG. 12A a top plan view of the discharge reed valve plate configuration with discharge valves.

[0062] FIG. 12B is an end view of discharge valve in FIG. 12A.

[0063] 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

[0064] FIG. 13 is a perspective sectional view of the housing gas and oil separation configuration with function notes.

[0065] FIG. 14A illustrates a vertical open drive with a dry (no oil) bottom motor operation.

[0066] FIG. 14B illustrates a vertical hermetic drive with bottom motor oil (wet) or dry operation.

[0067] FIG. 14C illustrates a vertical open drive with dry (no oil) top mount motor operation.

[0068] FIG. 14D illustrates a vertical hermetic center mount upsized single motor deployed with top and/or bottom mounted compressor and alternate machine.

[0069] 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.

[0070] FIG. 14F illustrates horizontal open drive with dry (no oil) motor operation.

[0071] FIG. 14G illustrates horizontal hermetic drive with center mounted dual compressors and downsized end-mounted motors.

[0072] FIG. 14H illustrates horizontal hermetic drive with center mount upsized single motor operation. for multiple compressor operation which also allows convenient compressor staging.

V. DESCRIPTION OF THE PREFERRED EMBODIMENT

[0073] 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**.

[0074] 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**.

[0075] 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. Head **19aa** (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.

[0076] 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**.

[0077] In addition to prior art complex linkages which are wear prone and expensive to fabricate, time proven axial piston pump downstroke retention methods are exemplified by Hugelman U.S. Pat. No. 7,794,212. Conventionally, a spheri-

cal 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.

[0078] 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.

[0079] 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**.

[0080] FIGS. **6A**, **6AA**, **6B** and **6C** illustrates 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.

[0081] 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).

[0082] 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.

[0083] 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.

[0084] 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.

[0085] In applicant's invention, sleeve nose **7a** (FIGS. 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.

[0086] 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.

[0087] 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.

[0088] 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. Applicant's 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.

[0089] 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 exemplified 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.

[0090] 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).

[0091] 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.

[0092] 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 CO₂ system.

[0093] 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.

[0094] 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.

[0095] 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.

[0096] 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** (FIGS. 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 exemplified by CO₂ 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.

[0097] 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.)

[0098] 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.

[0099] 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.

[0100] 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.

[0101] Thus there has been provided an axial piston compressor that fully satisfies the objects and advantages set forth herein. While the invention has been described in conjunction with specific embodiments, it is evident that many alternatives, modifications and variations will be apparent to those skilled in the art in light of the foregoing description. Accordingly, it is intended to embrace all such alternatives, modifications and variations as fall within the spirit and scope of the appended claims.

What is claimed is:

1. In an axial machine having a wedge with a planar angular surface with respect to a plurality of pistons in the axial machine, the wedge in driving engagement with the pistons, a center positioning mechanism for a retainer plate comprising:

- a plurality of slipper shoes placed on the planar angular surface of the wedge;
- each of the plurality of pistons having one end operatively connected to the slipper shoe and the other end slidably received in a cylinder;
- a retainer plate engaging the slipper shoes for maintaining the slipper shoes in contact with the wedge;
- a plurality of holes in the retainer plate with one hole in alignment with and receiving one of the slipper shoes so that each slipper shoe is received in a complementary hole;

each of the slipper shoes engaging a point on the side of the respective hole in which the slipper shoe is received with the point of engagement moving around its respective hole as the wedge drives the pistons, the points of engagement on the slipper shoes in their respective holes

restraining the retainer plate from movement parallel to the planar angular surface of the wedge.

2. The axial machine of claim 1 wherein the slipper shoes are equally radially disposed with respect to each other.

3. The axial machine of claim 1 and further comprising drive means connected to the wedge for rotating the wedge causing the pistons to reciprocate as the angular surface of the wedge is driven across the slipper shoes.

4. The axial machine of claim 1 and further comprising a barrel in which the cylinders are disposed in the barrel and a barrel drive means connected to the barrel for rotating the barrel causing the pistons to reciprocate as the slipper shoes are driven across the angular surface of the wedge.

5. The axial machine of claim 1 and further comprising a drive shaft mounted in the center of the axial machine for providing rotative power to the axial machine, a retainer sleeve mounted in axial parallel alignment with the drive shaft, the 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.

6. The axial machine of claim 5 and further comprising spring means engaging the sleeve to maintain the bottom of the sleeve in contact with the retainer plate.

7. The axial machine of claim 5 and further comprising a beveled surface on the bottom of the sleeve, and a complementary beveled surface on the retainer plate engaging the beveled surface on the bottom of the sleeve whereby the two beveled surfaces engage each other in continuous rotating engagement as the retainer plate changes its angular orientation with respect to the pistons during operation of the axial machine.

8. The axial machine of claim 5 and further comprising a beveled surface on one of the bottom of the sleeve or retainer plate, the beveled surface engaging the non beveled surface of the other of the bottom of the sleeve or retainer plate whereby the two surfaces engage each other in continuous rotating engagement as the retainer plate changes its angular orientation with respect to the pistons during operation of the axial machine.

9. The axial machine of claim 1 wherein the axial machine has top and bottom ends and the shaft extends completely through the top and bottom ends of the axial machine.

10. An axial machine comprising:

- an axial machine housing;
- a centrally mounted drive shaft;
- a wedge mounted in the axial machine housing, 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 plurality of holes in the retainer plate with one hole in alignment with and receiving one of the slipper shoes so that each slipper shoe is received in a complementary hole;

each of the slipper shoes engaging a point on the side of the respective hole in which the slipper shoe is received with the point of engagement moving around the respective hole as the wedge wobbles, the points of engagement on the slipper shoes in their respective holes restraining the retainer plate from movement parallel to the planar angular surface of the wedge.

11. The axial machine of claim 10 wherein the drive means for causing the wedge to drivingly engage the pistons comprises a drive shaft mounted in the center of the axial machine, the drive shaft providing rotative power to drive the wedge with respect to the pistons, a retainer sleeve mounted in 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.

12. The axial machine of claim 10 wherein the drive means for causing the wedge to drivingly engage the pistons comprises a drive shaft mounted in the center of the axial machine for providing rotative power to cylinder barrel, a retainer sleeve mounted in 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.

13. An axial compressor comprising:

- a compressor housing having a compressor head;
- an oil sump in the compressor housing for containing oil for lubricating the axial compressor;
- a gas intake port in the housing for receiving a gas to be compressed;
- at least one compressor cylinder for receiving and compressing the gas;
- a manifold in the compressor housing in fluid communication with the gas intake port through which the gas to be compressed must pass, the manifold in fluid communication with the compressor cylinder; the manifold having walls and a floor that contact the gas as it passes through the manifold before entering the compressor head;
- at least one hole in the manifold floor in fluid communication with the oil sump for allowing oil removed from the gas to drain into the oil sump;
- the manifold allowing oil to separate from the gas as the gas travels through the manifold on its way to the cylinder.

14. The axial compressor of claim 13 and further comprising thermal warming means for transferring heat from the axial compressor to the gas to heat the gas as it passes through the manifold.

15. The axial compressor of claim 13 and further 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 plurality of holes in the retainer plate with one hole in alignment with and receiving one of the slipper shoes so that each slipper shoe is received in a complementary hole;
 each of the slipper shoes engaging a point on the side of the respective hole in which the slipper shoe is received with the point of engagement moving around the respective hole as the wedge wobbles, the points of engagement on the slipper shoes in their respective holes restraining the retainer plate from movement parallel to the planar angular surface of the wedge.

16. The axial compressor of claim **15** and further comprising a retainer sleeve mounted in axial parallel alignment with the drive shaft, the 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.

17. An axial machine comprising:

an axial machine housing;
 a central, vertically mounted drive shaft;
 a wedge mounted in the axial machine housing, 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 slidably received in each of the cylinders,
 drive means for causing the wedge to drivingly engage the pistons thereby causing the pistons to reciprocate;
 an oil sump in the compressor housing for containing oil for lubricating the axial machine;
 a retainer plate engaging the slipper shoes for maintaining the slipper shoes in contact with the wedge;
 a cavity around the drive shaft in fluid communication with the oil sump;
 at least one oil passageway in the wedge in fluid communication with the cavity, the oil passageway having discharge ports in the wedge to disburse oil on the planar angular surface.

18. The axial machine of claim **17** wherein the cavity is formed around a reduction in the drive shaft diameter at a location along the drive shaft.

19. The axial machine of claim **17** and further comprising centrifugal drive means along the drive shaft to centrifugally spin the oil upward from the oil sump to the cavity and then to the wedge discharge ports to lubricate the surface of the wedge.

20. The axial machine of claim **17** and at least one additional oil retention cavity in the wedge to store an oil reserve to be used on initial start up of the axial machine.

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