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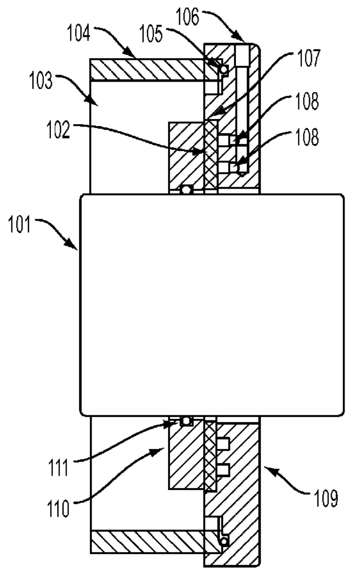

FIG. 1A

(57) Abstract: In order to effect a seal; a porous material which comprises one side of two opposing surfaces is used to restrict and evenly distribute externally pressurized gas, liquid, steam, etc. between the two surfaces, exerting a force which is opposite the forces from pressure differences or springs trying to close the two faces together and so may create a non contact seal that is more stable and reliable than hydrodynamic seals currently in use.

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AIR BEARING FOR USE AS SEAL

Reference to Related Applications

[0001] The present application claims the benefit of U.S. Provisional Application Nos. 61/582,674, filed January 3, 2012; 61/704,927, filed September 24, 2012; and 61/728,595, filed November 20, 2012, whose disclosures are hereby incorporated by reference in their entireties into the present disclosure.

Field of the Invention

[0002] The present invention is directed to a seal and more particularly to a seal for use in fields such as the following: oil and gas, power generation (including energy storage like compressed air or pumped hydro storage), aero turbines, chemical processing, paper manufacturing, aeration and water purification, gas separation and other process industries. Within these industries is likely that technology will find its way to; pumps, compressors, turbines, generators, motors, turbo expanders, turbo chargers, mixers, and refiners.

Description of Related Art

[0003] Creating and implementing effective seals for rotating equipment has been an effort for almost as long as there has been rotating equipment. And just as there is a broad array in the applications and types of rotating equipment there is also a broad array of seals that are employed in this equipment.

[0004] One of the simplest and oldest type of seals, a packing based seal, is still often employed. In this seal a gland can be tightened to compress the packing around the shaft. So there is a balance between how tight to make the packing which causes frictional losses and wears the shaft and how effective the seal is. Lip seals are in another form of contact based seals. They also can wear grooves into the shaft and are subject to wear and leakage themselves.

Labyrinth seals are a form of non-contact seal but they provide a conductance path that can result in huge flows when there is significant pressure differentials across the seal. In order to minimize the leakages clearances between the rotating in stationary sections of the seal are minimized to the extent possible. This adds significant costs and still to make them effective they often need to be relatively long axially. There are brush and ablatable type seals which are contact based seals often employing centrifugal force or pressure differentials to keep them in contact with their mating surface. These seals create particulate and are a wear item that becomes a maintenance cost, at high speeds they create significant amounts of heat and frictional losses. Additionally these contact seals create a lot of noise. Bearing isolators are commonly found in process equipment, they typically combine labyrinth and lip type seal technologies and sometimes employ and injection of a fluid or gas at a pressure above that of the volume to be sealed. In example of such a art would be; US 7,631,878 Orlowski

[0005] Mechanical seals and dry gas seals also could be considered injection type seals, as these seals often have some type of a flush or seal gas employed in them. Dry gas seals specifically use hydrodynamic air bearing affects to create very small non-contact gaps that are very effective at sealing. As the ceiling effects are dependent upon and aerodynamic bearing Be sealed only work at relatively high surface speeds between the sealing surfaces. There is a lot of engineering that goes into the seals in order to keep their bearing surfaces flat and pressed against each other so as to prevent a seizure from contact at speed between the bearing surfaces or a failure to seal because a mechanism that is used to provide axial compliance "Hangs Up" allowing there to be a large gap between the sealing surfaces. Mechanical seals also suffer from the same issues but there sealing surfaces are designed to be relatively good plain bearing partners, still because they are often contact bearings they wear and they create heat. An example of such a seal would be; US 7,823,885 Droscher which does a good job describing the problems with conventional seals and "hang-ups" and is also an example of a conventional injection type seal, additionally it is noted that this injection fluid may also help to establish an aerodynamic bearing property at the face of the seal.

[0006] The state-of-the-art today includes hydrodynamic bearings such as spiral groove and foil bearings which can become noncontact based on the viscous dragging of a fluid or gas into a gap, there are gas seals and labyrinth seals that attempt to create restriction through small gaps. Examples of gas seals include Pall Corporation and Carbone Turbograph seals. In the case of Carbone they are manufacturers of porous media carbons and graphite's but they do not employ porous media or graphite as a compensation technique for hydrostatic sealing purposes. We suggest this is evidence that using such compensation techniques through porous media is not obvious to them.

[0007] In another contemporary example US patent application US 2006/006-2499 Al specifically teaching and claiming the use of carbon graphite and ceramic materials and using pressurized gas does not employ porous media compensation, This patent is targeted specifically towards high-speed turbine engines. We suggested this as an example that the use of porous compensation is not obvious.

Summary of the Invention

[0008] Objects of the invention include: to provide more reliable gas seal and mechanical seal surfaces by teaching robust ways of; employing externally pressurized air bearing technology to create high pressures in noncontact sealing gaps; prevent or reduce ware and heat buildup between faces that are meant to run in contact or that may make occasional contact; teach how the combination of bearing and seal technology can dramatically simplify the sealing of rotating equipment and to prevent "hang ups" which is a failure of the compliance or biasing mechanism to keep the faces pressed against each other.

[0009] Modern Air bearing technology when applied to Turbo equipment has very interesting applications for replacing conventional bearings and seals (eliminating oil, improving efficiency, reducing noise).

[0010] Seals and counter surfaces designed to use one of two opposing sealing surfaces as a porous media to restrict externally supplied hydrostatic pressure into an air bearing gap between the surfaces. This technology would apply to replacing packing in large agitators, mixers or refiners; at the lower end of the seal continuum listed above and mechanical or dry gas seals on the higher end. One embodiment is the blade runner which is particularly well-suited to aero engine sealing enabling a dramatic reduction in friction, ware, and noise with improved stage sealing. Additionally the rotating mass and axial length of the seal is dramatically reduced. Another embodiment taught is the balance force face seal where mechanical face seals would run in contact with virtually no contact force even though there may be thousands of pounds of closing force.

[0011] Externally pressurized gas bearings have several fundamental advantages for use as seals;

[0012] Noncontact and operate independent of relative motion (they work at zero RPM)

[0013] Do so using the process gas in bearing the gap

[0014] Operate at the most extreme temperatures

[0015] Use high pressures but low flows

[0016] Combine sealing and bearing functionality

[0017] As non-contact seals, they have no coulomb friction and no wear. There is viscous sheer friction in the air film but this is orders of magnitude less than the bearing friction. This bodes well for reduced energy consumption and allows for strong green sales arguments. Radial and thrust or face type seals are both possible.

[0018] Even though they are noncontact they are mechanically coupled to their counter surface via the compression of the air film. As in the example of figure 350, the bushing seal is supported by the spinning shaft it is sealing on. This allows for eliminating alignment issues found in labyrinth seals. This is a self aligning capability. The seal is stationery with respect to the stator and connected to it through some sort of a flexible bellow, diaphragm or O-ring as examples of compliant mounts. Bearing isolators such as can be found from GGB, Waukesha and Crane are examples of other compliant mounts that allow for the shifting of the center of the shaft and also for angular changes of the shaft. The bearing isolators mentioned above use copious quantities of compressed air or injection fluid into a gap as they do not use compensated hydrostatic bearing technology.

[0019] The high pressure maintained in the air gap is a highly effective seal against the migration of contamination, liquids and even gases. Conventional labyrinth seals and gas seals that employ pressure flowing into one or more groves suffer from relatively low pressure (a few PSI) and high flow (tens of cubic feet per minute). An air bearing seal though, can easily generate 30 PSI in the gap and the flow is measured in cubic feet per hour.

An example of such a bearing in a thrust face configuration is our standard line of vacuum preloaded air bearings. It is counterintuitive but we successfully use 30 PSI in an annular air gap to separate atmospheric pressure from vacuum which is used to preload the air film in lightly loaded precision stages.

[0020] "Air bearings have applications to a wide set of categories in sealing technologies but they are not (yet) commonly employed."

[0021] Although porous media air bearings are not new and the idea of hydrostatic seals are also not new there is very little in the prior art and even less in practice that combines the advantages of both ideas.

[0022] Air bearings as bearings and seals have the potential to revolutionize the fundamental design of Turbo equipment.

[0023] Porous hydrostatic seals are not difficult to manufacture. A layer of porous media generally between 0.020 and .2in thick, shrunk fit and or bonded into a nonporous housing with a air distribution labyrinth between the two layers and a finish boring of the porous media face to the appropriate diameter for the Journal or turning for finish flatness for the thrust face is all that is required. Generally the air gaps are between 0.0001"and 0.001" with the sheer energy and the flow through the gap both being squared and cubed functions of the gap. The flow through the porous media is determined by the desired flow through the gap and generally being about two times the desired flow with the shaft or thrust face in place. As the speed of the shaft increases, the ideal gap thickness should also increase to minimize heat build-up through sheer energy losses. The equations used to determine this are generally known in the art.

[0024] Modern Air bearing technology when applied to Turbo equipment has very interesting applications for replacing conventional bearings and seals (eliminating oil, improving efficiency). The specification that follows teaches this in several embodiments.

[0025] In order to effect a seal; a porous material which comprises one side of two opposing surfaces is used to restrict and evenly distribute externally pressurized gas, liquid, steam, etc. between the two surfaces, exerting a force which is opposite the forces from pressure differences or springs trying to close the two faces together and so may create a non contact seal that is more stable and reliable than hydrodynamic seals currently in use. The aerostatic pressure may be adjusted to the point where the two faces are completely unloaded and zero contact pressure exists between the two faces even though the faces are in intimate contact. Because the faces are in contact there is approximately zero flow through the gap and the line pressure being fed into the porous material will exist between the two faces. This contact force can easily be adjusted by varying the input pressure to reduce ware and heat generated by friction in conventional contact seals.

Brief Description of the Drawings

[0026] Preferred embodiments of the present invention will be set forth in detail with reference to the drawings, in which:

[0027] 1A - Simplified single face gas bearing seal

[0028] IB - Single face flexible rotating element

[0029] 1C - Prior art image without description

[0030] ID - Tandem face seal in preferred embodiment

[0031] IE - Flexible stationary primary with adjustable air closing force

[0032] IF - Flexible stationary primary with mechanical closing force only

[0033] 1G - Flexible stationary primary with Torus

[0034] 2A - Double opposed simplified gas bearing seal

[0035] 2B - Flexible rotating element with double opposed preferred

[0036] 2C-1 through 2C-4 - porous versus hydrodynamic gas seals

[0037] 2D - Lift load chart for porous air bearings

[0038] 3A - Circumferential gas bearing seal

[0039] 3B - Mounting method for circumferential seal

[0040] 4 - Single blade seal

[0041] 5A - Prior art detailed description

[0042] 5B - Gas bearing eliminates seals, preferred embodiment

[0043] 6A - Multi-blade seal

[0044] 6B - Parallel flexure, aero engines

[0045] 6C - Close-up of parallel flexure

[0046] 7A and 7B - Angular seal compliance

[0047] 7C - Angular and axial seal compliance

[0048] 8 - Axial, angular, radial unvented seal compliance

[0049] 9 - Axial angular radial single source

[0050] 10 - Axial angular radial vented

[0051] 11A - Balanced force bearing drawing

[0052] 1IB - Balanced force bearing example

[0053] 12 - Method for making bearing seals with wide temperature capability

Detailed Description of the Preferred Embodiments

[0054] Preferred embodiments of the present invention will be set forth in detail with reference to the drawings, in which like reference numerals refer to like elements throughout.

[0055] With reference to figure 1A; a shaft 101 which may rotate at high speeds has a runner 110 coupled to it via an O-ring 111 (or another mounting mechanism as detailed in illustration IB or other arrangements detailed in this specification or known in the art). O-rings provide axial compliance to the runner allowing for self adjustment of the gap between it and the stationary surface and axial displacements of the shaft. If the runner is hard mounted to the shaft some axial compliance should be designed into the stationary components. The runner is free to move radialy on the air film. Conventional mechanical face seals as shown in figure IB often have a spring loading mechanism to urge the opposing faces of the seal into contact. This technique is well-known in the art and may be employed to adjust the load on the seal faces and to provide axial compliance. 103 represents a volume on one side of the seal, this could be a gearbox, a motor-generator housing, or a process fluid or gas such as a mixer, refiner, water pump or gas pipeline compressor, or a seal between compartments in a piece of rotating equipment as examples. 104 would represent the casing or the housing. There may or may not be an adapter plate as shown in 205 of figure 2A. The seal body itself 109 would likely mount to the housing casing or adapter plate with an O-ring seal 105. The seal body in illustration 1A refers to a seal that would be lightly loaded, it should be recognized that the seal body and its mounting needs to be stiff enough so that it does not perform significantly under the pressure differentials that are being sealed. The seal body is equipped with conductive passages 106 to communicate the pressurized fluid to the labyrinth 108 which evenly distributes the pressurized fluid to the back of the porous media

107. The porous media 102 may be comprised of graphite, carbon, silicon carbide, Tungsten carbide, alumina or basically any porous and or sintered material. These materials are typically found as face seals and mechanical seals and as runners and runner faces in dry gas seals. Just instead of filling or sealing this porosity the porosity is used for air bearing functionality. Porous media air bearing compensation is only one potential solution, orifice, step, grove, inherent or pocketed compensation among other compensation techniques known in the art may be employed. Porous air bearings are known in the art and are described by the inventor in previous applications. Also methods for providing clean fluids at pressure are well-known and readily available. In 100 B the difference from 180 is that the shaft 101 is equipped with a sleeve 112 that is fixed on the shaft and a spring element 113 pushes or biases the runner or mating ring against the porous bearing seal face and or primary ring.

[0056] The thickness of the gap is a function of the hydrostatic input pressure, the forces urging the faces together (from pressure differences, spring forces, dynamic forces, etc.), the restriction of the porous media and the ratio of surface area to leak edge of the surfaces. These variables may be controlled to create highly effective noncontact seals.

[0057] With reference to figure 1C; A shaft 151 for a piece of rotating equipment such as a compressor or turbine is fit with a sleeve and meeting ring 152 which cooperates with a primary ring 150. 153 the compressor case receives 159 seal cartridge as is common in the art and detailed by API standard 682. 154 porous face of the primary ring which does not rotate is fed pressure through the seal cartridge. When the primary ring is arranged to be a "flexible element", that pressure may be introduced through a port 155 into a Plenum 157 which is sealed with O-rings 156 as also described in illustration 160, 170, 180 or with a method already known in the art including a tube which would screw directly into the primary ring (not shown here but see illustration 350). A biasing force which keeps the air bearing primary ring pressed against the rotating mating ring is affected through a spring 158 or a diaphragm type flexure, which is known in the art, and or methods using air pressure, of which two are taught below. In a tandem seal the same description is repeated with regard to the secondary seal.

[0058] In figure ID a primary ring 169 with a porous face 160 and a Plenum to distribute air pressure behind the porous face 161 is contained inside the seal cartridge by O-rings 162. Port 165 distributes the air from Plenum 167 which the pressure is introduced to through port 168. 166 is a spring or diaphragm which provides a biasing force pushing the primary ring against the mating ring. It is desirable the primary ring always be pushed up against the mating ring to avoid any potential leakages. A seal "hang up" is when the compliant ring is not forced up against its mating ring for some reason. This allows for undesirable back flows. In order to help prevent "hang ups" the air pressure being fed to the face of the bearing may also be employed on the back of the, in this case, primary ring. The differences in the diameters between 163 and 164 may be designed such as to maintain the desired closing forces between the seal faces. So as the pressure drop through the porous media is likely to be on the order of 50% if the area described within the diameter of 164 equals 50% of the area at the bearing face the forces would be equal. This does not account for forces from other pressure differentials or from the biasing springs or flexures which should also be considered and designed for as one competent in the art could well do.

[0059] In figure IE a primary ring 179 with a porous face and 170 and a Plenum to distribute air pressure behind the porous face 171 is constrained between O-rings 172 within the seal cartridge. The air pressure input for the bearing functionality through port 178 conducts to port 175 before reaching Plenum 171. Vent 173 is used to be sure that pressure meant for the air bearing face is isolated exerting a force on the back of the primary ring. In this way only the spring or diaphragm forces will urge the primary ring towards the mating ring.

[0060] Figure IF shows a primary ring 189 with a porous face 180 and a Plenum to distribute air behind the porous face 181 is constrained in the seal cartridge via an O-ring 182 and a Torus 183 (a segment of a sphere or a curve with constant diameter) cooperates with a close fit inside of the seal cartridge diameter 184. Air pressure to the bearing is introduced though port 188 and communicates to port 185 though plenum 187 then the labyrinth 181 to the porous media 180.

[0061] As shown in Figure 2A, a shaft which may rotate at high speeds 201 has a runner 214 coupled to it via an O-ring 211, O-rings or another mounting mechanism as described in other attached figures. O-rings provide axial compliance to the runner to allow it to find center between the air bearing faces and provide for small axial displacements of the shaft. If the runner is hard mounted to the shaft some axial compliance should be designed into the stationary components. The runner is free to move radially between the air films. 202 represents a volume on one side of the seal, this could be a gearbox, motor-generator housing, or a process fluid or gas such as a mixer, refiner, water pump or gas pipeline compressor as examples. 203 would represent the casing or the housing. 205 illustrates that there might be an adapter plate that would employ a stationary O-ring seal like 204. The seal body itself with mount up to the casing housing or adapter plate as shown including a potential O-ring seal 206 at that interface. The seal itself comprises two rings 213 and 216 with annular and opposed air bearing faces 212. The rings are separated by a spacer 215 which has axial dimensions similar to the runner 214. The spacer also provides for a vent or drain 208 to atmosphere so that pressure does not build up in the volume between thrust faces 209. The spacer may be slightly larger than the runner or slightly smaller than the runner depending on the design objectives. A larger spacer would allow more clearance, a smaller spacer would provide for a clamping functionality in the event hydrostatic pressure is lost, in this case, and with the runner fixed to the shaft, the seal would act like a conventional contact face seal. The rings 216 and 213 which are comprised of a nonporous material or are sealed except for the intended bearing/seal faces, provide for conductivity of the hydrostatic fluid though port 207 and a labyrinth 210 to evenly distribute said fluid to the backside of the porous media or an area close to the intended faces. The porous media 212 may be comprised of graphite, carbon, silicon carbide, alumina or basically any sintered material. These materials are typically found as face seals and mechanical seals and as runners and runner faces and dry gas seals. Just instead of filling or sealing this porosity the porosity is used for air bearing functionality. Porous media air bearing compensation is only one potential solution, orifice, step, grove, inherent or pocketed compensation among other compensation techniques known in the art may be employed. Porous air bearings are known in the art and are described by the inventor in previous applications. Also methods for providing clean fluids at pressure are well-known and readily available.

[0062] In figure 2B shaft 221 for a piece of Turbo equipment has a seal cartridge 222, within the cartridge is a mating ring 223 which in this case is a rotating flexible element. The mating ring as shown is integral to a sleeve but the sleeve and ring may be separate components. The mating ring with integral sleeve is supported axially on the shaft via Springs 232 and the mating rings 223 and 233 are locked together axially by clamping ring 237. The mating ring 223 runs against a stationary primary ring 229 which in this preferred embodiment has a porous bearing face 224 and the necessary labyrinth 226 and input porting 227 to create an effective hydrostatic gap using the porous media 225 as the restrictive element. In this embodiment as an example of a double seal there is a second mating ring 233 and a second primary ring face 230 on the opposite side of 229 from the other air bearing primary ring face. Both bearing systems are fed external pressure and vented using the same systems and porting. The volume between seal face 224 and 230 is vented though port 228, preventing a pressure build up there. The same is the case between seal faces 230 and 231, they are vented. It should be noted that some of the flow exiting seal gap 224 will flow to the process side. The amount would depend on the pressure differences. So if volume 234 is pressurized to 1000 PSI and the flow out of vent 228 is at ambient pressure most of the flow will be out of the vent rather than into the process. The input pressure to the porous media 225 should be 4 to 6 bar above the pressure it is sealing against so 1060 to 1090 PSI. It is of course possible to regulate a cascading lower pressure to each of the successive faces so as to drop the pressure down over stages. If each stage has a 1000 PSI pressure drop the seal could effectively seal 3000 PSI.

[0063] A novelty here independent of the use of porous media is that the air bearing sealing gaps between the faces of the porous bearings on 229 in the inside facing surfaces of mating rings 223 and 233 is fixed at assembly. It is not a spring-loaded fit and so there is no possibility for a hang-up, as noted as a main problem in the current art by patent number 7,823,885 Droscher which leaves the seal faces open. The robustness this of the porous media air bearing seal technology means that even if the mating rings get hung up on the shaft and there is an axial displacement of the shaft relative to the seal cartridge the bearing faces will not be materially damaged and the mating ring sleeve will move on the shaft. This effect can be increased by adding more mating and primary ring faces, bearing seal face 231 in primary ring 236 is an example of this. The thickness of 235 is fit at assembly. Additionally another porous carbon bearing face could run on the opposite side of mating ring 223 in the space described by 234. This is taken to its logical conclusion in the teaching of illustration 600.

[0064] It will be recognized that the API suggests a number of different arrangements for seals and that these include face-to-face, back-to-back, double opposed, and tandem seals. In some cases flexible elements spin on the shaft and in some cases flexible elements are integrated into the Stator. Is meant and here illustrated that this porous air bearing technology can be employed in the faces of all these seal arrangements.

[0065] Figures 2C-1 through 2C-4 are meant to illustrate the advantages of externally pressurized porous media air bearings over aerodynamic bearings in gas seal applications. In figure 2C-1, we see the introduction of a seal gas at a higher pressure than process gas at port 253, the runner 252 rotating at a high speed with the shaft 251 has aerodynamic features 254 etched in its faces to help establish an air bearing film on each side of the runner. The seal gas flows into the gap from the outside edge 255 across the gaps to exit the lower pressure edge 256. In figure 2C-2, an axial or angular change in the shaft with respect to the Stator makes the gap on one side smaller 257, as the gap on the other side gets bigger 258. It is noted that the stiffness of the air films would resist such a motion, still such motions do occur. At this point the flow of seal gas to the side with the smaller gap 257 would be reduced as the pressure would take the path of least resistance which is the larger gap 258 on the opposite side. In this case where the runner is closest to the counter surface the flow is restricted and there is lower flow into the area where we would like to have the highest pressure in order to avoid contact. At the point of contact all the pressure and flow will be through the large gap side 258, driving the runner to the opposite side 257. This is an unstable situation. Although the hydrodynamic features may be attempting to pump air pressure into this area it is difficult to get air to flow into the small gap especially if it was more of an axial rather than in angular change.

[0066] In figure 2C-3 high pressure gas is introduced through port 259, into the plenum 260, then through the porous media 261 which restricts the flow into the bearing gap 262, there are no features etched in the runner 263. With a similar axial or angular change in the position of the runner towards one of the bearing faces 263 the bearing pressure in the gap at 263 will automatically increase until the runner actually makes contact, at which point the pressure attempting to exit the porous media will approach the input pressure. The relative force between the runner and the bearing face is mitigated by the pressure attempting to exit the bearing face at 263. At the same time; the opposite side 264 has a lower pressure as the gap is larger and the restriction is coming from the porous media instead of the edge of the gap so the bigger gap results in lower pressure. This results in a naturally stable situation where the side with the smallest gap is always building the highest pressure and the side with the biggest gap is having the lowest pressure. In the aerodynamics seals illustrated in 2C-1 and 2C-2 the reverse is the case.

[0067] Looking to figure 2D can be seen that the stiffness of an air bearing film changes with its thickness. The thinner the air gap the higher the stiffness. The chart in figure 2D is a lift load curve; the slope of the curve is representative of the bearing stiffness at that point. A horizontal line represents zero stiffness and a vertical line represents infinite stiffness. Whether you are dealing with an orifice in the face of an externally pressurized bearing or the gap at the perimeter of an aerodynamic bearing, smaller the gap harder it is to get enough air to distribute across the full surface of the bearing. With the porous bearing the air is issuing from the whole face of the bearing directly into the gap, there is no issue with trying to get the air to flow across the gap. This makes the porous bearing a more robust gas bearing. - Additionally it is worth noting the flow through a gap is a cubed function of the gap, so doubling the gap results in an eight fold increase in the flow. The stability of porous bearings allows such small air gaps with a high degree of safety and reliability and are preferred.

[0068] As seen in Figure 3A, a shaft 301 which may rotate at high speeds is rotated inside of a stationary cylindrical bearing seal 310. Contamination or pressures that exist in volume 308 are sealed and denied egress into the gap 309 by hydrostatic pressure exiting from the gap 309. The housing or casing represented by 311 may be equipped to receive the cylindrical air bearing seal directly or an adapter block 315 may be used, in which case an O-ring 313 would provide a static seal at that interface. In this embodiment it is preferred to have a retainer 303 on the low pressure side of the seal and clearance 302 should be provided between this retainer in the shaft. A passageway 306 is required to conduct high pressure fluid to the cylindrical seal assembly. O-rings 312 can provide multiple functions, one of these functions is to seal plenum 307 so that this high pressure fluid may be conducted into the seal body 310 through a single whole 306 without directly connecting a fitting to the seal body. These O-rings also may be used to provide for radial and angular compliance, the shaft is completely free to move axially on the air film. The O-rings 312 may also be used to contain epoxy which may be injected through a hole 314 which will fill the cylindrical gap 304 between housing or mounting block and the seal body mounting it ridgeley if desired. With air pressure on, the seal will self align to the shaft which should be held on the design centerline of the machine as the epoxy cures, hard mounting the seal in place. (See new way air bushing mounting information).

[0069] The high pressure fluid entering through Aperture 306 and finding its way through the hole in the seal body will be distributed axially and radially between the seal body 310 and the porous media 316 by a labyrinth 305 which may be in the porous media or the seal body. Although porous media compensation is the preferred embodiment other compensation methods are possible. Porous media air bearing compensation is only one potential solution, orifice, step, grove, inherent or pocketed compensation among other compensation techniques known in the art may be employed. Porous air bearings are known in the art and are described by the inventor in previous applications. Also, methods for providing clean fluids at pressure are well-known and readily available. The porous media 316 may be comprised of graphite, carbon, silicon carbide, alumina or basically any sintered or porous material. These materials are typically found as face seals and mechanical seals and as runners and runner faces in dry gas seals. Just instead of filling or sealing this porosity which is a common practice the porosity is used for air bearing functionality.

[0070] With reference to figure 3B; there is a shaft 351 and a housing 352 which are coupled through a bearing system 353. Being desirous to isolate the bearing from the process or environment in area 357 an aerostatic gas seal 355, consistent with the illustration in figure 300 (except in this example the aerostatic pressure is plumbed through a flexible tube 356) just shorter axially, is coupled to the shaft 351 through a high-pressure air film which supports the seal 355 in a noncontact fashion with respect to the shaft. So the shaft may rotate at a high rate of speed with virtually no torque transmitted to the seal because of the low shear forces in the air gap, but the seal is able to follow motions of the shaft without contact due to the radial stiffness of the air film. The mechanical bellows allows the seal to follow the shaft rather than keeping it rigidly coupled to the housing. Additional methods for providing compliance are detailed in illustration's 700 though 1,000.

[0071] In contrast to labyrinth seals cylindrical air bearing seals are coupled to the shaft via the stiffness of the air film. In example 350 the bushing seal is supported by the spinning shaft it is sealing on. This allows for eliminating alignment issues found in labyrinth seals. The seal is stationary with respect to the Stator and connected to it through some sort of flexible bellows arrangement 354, diaphragm or an axial O-ring as examples of compliant mounts. It would also be possible to take a circumferential seal and mount it between axial face seals as described in figure 200 and 800.

[0072] Bearing isolators that look like figures 2A, 3B or 7-10 can be found from Garlock, Waukesha and Crane and are examples of other seal manufacturers employing compliant mounts that allow for the shifting of the center of the shaft, angular excursions of the shaft and axial displacements. In some cases these bearing isolators used pressurized air or water though an uncompensated annular grove to help affect the seal. These are characterized by high flows and low pressures due to their large gaps and lack of compensation.

[0073] A shaft 401 which may rotate at high speeds has a blade runner 405 coupled to it using a mounting ring 413 which is fixed to the shaft by set screws 403 and or a shoulder. An O ring 410 may be employed to seal clearance at 402. You will notice that there are two illustrations in figure 4; in view A the blade 405 is not up against the porous seal bearing face and the gap 406 allows for a view of the blade runner 405, in view B, the blade runner is in place and the gap 406 between it and the porous face 412 is as it would be in operation, less than 25 microns. The blade itself may be coupled directly to the shoulder if the shaft is so equipped with a shoulder (a shoulder would be the axial face created by a step in the diameter) see 600. The blade runner is characterized by being thin axially and so differentiated from conventional runners. The blade may be any thickness but likely between 0.1 and 1 mm thick. This blade runner has the advantage of being light weight and so it has a minimal effect on the moment of inertia of the shaft and on potential imbalances caused by the runner. Because the pressure to be sealed in volume 404 is the same everywhere in the volume it acts uniformly on the back of the blade flexure, urging it against the air bearing seal face with constant per-unit area force. For this reason it is not necessary to have a heavy rigid runner connected to the shaft. The gap 406 will vary but a force equal to and opposite forces existing in volume 404 will be generated in the air gap. This embodiment may be well suited for replacing brush type seals especially in turbines designed as aero engines. As it would seal more effectively, have zero or at lease relatively low friction or ware and occupy significantly less space axially.

[0074] High pressure gas some bars higher than what exists in volume 404 is introduced into port 408, which conducts the pressure to the Plenum 409, which distributes the air pressure uniformly to the backside of the porous media 412 which will create a pressure in the gap 406 at its face and between the runner 405

[0075] The volume 404 represents a volume on one side of the seal, this could be a gearbox, a motor-generator housing, or a process fluid or gas such as a mixer, refiner, water pump or gas pipeline, or a seal between compartments, impellers or stages in a piece of rotating equipment like a compressor as examples. 414 would represent the casing or the housing. There may or may not be an adapter plate as shown in fig 200 number 205. The seal body itself 411 would likely mount to the housing casing or adapter plate with an O-ring seal 407.

The seal body in illustration 100 refers to a seal that would be lightly loaded, it should be recognized that the seal body and its mounting maybe designed to be stiff enough so that it does not deform significantly under the pressure differentials that are being sealed. Alternatively it may be designed so that it does flex and so may flex to cooperate with the conformable nature of the Blade Runner which is essentially a flat spring steel flexure.

[0076] In the prior art of Figure 5A, a conventional centrifugal compressor employs a sealing and bearing system described here (but this is descriptive of many other potential applications in rotating equipment); shaft 501 comes from the compressor camber 504 though a labyrinth seal 502, into the seal cartridge 503 which fits into the seal chamber within the compressor casing 505. Then a face or dry-gas seal affected between the primary ring 507 and mating ring 506, which we will refer to as the primary seal 508. Between the labyrinth 502 and the primary seal 508 a buffer/flush gas is introduced though port 524, most of this gas flows back to the process side as the labyrinth seal has a high degree of flow even with only a bar's worth of pressure difference. This buffer gas is important to keep the primary seal gap clean. Some of the gas flows across the mechanical face or aerodynamic primary seal 508 and into the Plenum 509, finally exiting through vent 510. Then there is a seal gas or inert gas introduced through port 512, as before most of this flows through labyrinth seal 511 and out vent 510. Some of this gas does flow through the secondary seal made up of the mating ring 513 and primary ring 514. This is because the pressure being introduced at 512 is higher than the pressure in the volume 515. This flow is exhausted through vent 516. Then there is a separation gas introduced through port 517 which flows through the separation seal 518. Some of that flow migrates into volume 515 and vents through 516 and some of that flow makes its way through labyrinth seal 519 (if so equipped) and into the bearing chamber 520. So we have process and buffer gas flowing out of vent 510 and this is mixed with seal or inert gas that was introduced through 512. This needs to be reprocessed or sent to flare. The gas flowing across the secondary seal and into volume 515 mixes with the separation gas being introduced through 517 and then exits out of vent 516 and also needs to be sent to flare or otherwise processed or reported as an emission. Additionally separation gas flowing into the bearing chamber 520 will find its way out vent 521 and becomes yet another environmental headache. The bearing chamber has oil pumped in at pressure through port 522, the oil then needs to be drained out through port 523 (that may be positioned at the bottom) filtered and cooled to control its viscosity which important because it is very temperature sensitive. With all of those tubes coming and going from each end of the compressor, more than one operator thought he was looking upon Medusa.

[0077] In the preferred embodiment shown in Figure 5B, the services, complications and environmental headaches listed above are eliminated by the following novel teachings. With reference to figure 550 please notice that oil has been removed as a lubricating medium for the bearings supporting the compressor shaft. Instead gas bearings operating on the gas being compressed in the compressor are used to create an aerostatic air bearing support 560 for the shaft 551. The bearing cartridge 555 and the bearing chamber and or seal chamber in the compressor casing 554 may change in new designs to take advantage of the much more compact design that is possible, but this is not necessary as the gas bearing cartridge can fit in the same space that the oil bearing cartridge fits into.

[0078] The preferred embodiment is to use a porous media restriction 558 at the face of tilting pad externally pressurized air bearings 560. These bearings can be fed using the same buffer gas that had been employed in the prior art but this buffer gas is instead pumped into the externally pressurized air bearings 560. The bearings require a higher pressure differential, likely in the range of 4 to 20 bar above the pressure on the other side of labyrinth seal in volume 552, but the volume of this buffer gas flow, that is now bearing gas, is dramatically less than was required of buffer gas in the prior art, likely less than one cubic foot per minute per bearing. The buffer gas may be taken from the high-pressure side of the pump, or the suction side, conducted through filters or dryers, compressed if taken from the suction side, and then introduced through port 556, into the bearing 560, distributed to the labyrinth 557 restricted by the porous media 558 and then finally exiting under pressure through the final bearing restriction, gap 559. After the gas has exited the bearing gap 559 it acts to raise the pressure in the bearing compartment slightly as the used gas will flow back into the process through the labyrinth seal 553 or some other ring or separation seal that may be used in that location.

[0079] All vents are eliminated, there is no reason to have a process flow into the bearing chamber there is no were for it to go. This eliminates having to flare or report vents to atmosphere, and is a huge environmental advantage. And as there is only one gas to deal with, services are dramatically simplified, improvements in maintenance costs and downtime and the reduction in capital costs as seal services capital cost can be a multiple of the cost of the seal. Safety is also improved, as the elimination of venting also eliminates the possibility of entraining oxygen into flammable gases being compressed or allowing dangerous gases to escape.

[0080] Rotor dynamics are dramatically improved by the use of this invention, the length of shaft that had previously been consumed by seals may be eliminated, dramatically stiffening the shaft 551. The diameter of the shaft can be increased due to the higher speed capability of the gas bearings again stiffening the shaft and providing more area for squeeze film damping in the gas bearing.

[0081] The environmental problems and mess associated with oil are eliminated, there are no more oil leaks. No oil can make it to a face or dry gas seal and carbonize. Oil no longer controls the temperature which the bearing compartment can operate at. Gas bearings may operate at the most extreme temperature ranges, from cryogenic to supper heated steam. It is noted here that conventional techniques for gluing porous media 558 to the stainless steel or aluminum bearing housings 560 is not appropriate for extreme temperatures..

[0082] The compressor or also in the case of a gas turbine or large generator will have the rotor supported on a frictionless gas film even at zero RPM. This reduces the risk at startup and shutdown, allows for slow roll and standby operations without danger of seal hang-ups or bearing damage and enabling frictionless startups and shutdowns.

[0083] Because of the excellent aerodynamic properties available from the smooth porous face, external pressure to the bearing maybe often turned off once the compressor or turbo machinery is at sufficient speed, as at that point the shaft will be supported on aerodynamic effects. So the auxiliary compressor (if so equipped) may be run only at startup and shutdown, or slow roll conditions. If this auxiliary compressor failed during operation it would not affect the operation of the main compressor and the rotor could spin to a stop in a loss of pressure without damage due to the excellent tribological properties of the steel shaft on a carbon graphite bearing face. Additionally the technology is appropriate for canned compressors targeted towards subsea compression as the bearings can take their pressure from the high pressure side of the pump and have acceptable life as plain bearings in the start stop cycles. This is a much simpler and more compact way of eliminating oil than magnetic bearings.

[0084] But without sealing - and without venting - the bearings operate under extreme pressures. If the suction pressure of the pump is 100 bar, and the output side of the compressor was 200 bar, then the bearings could be fed at 106 bar and the flow through these bearings becomes the buffer gas. Bearings that operate in a 100 bar environment, actually only see a 6 bar pressure difference.

[0085] As shown in Figure 6A, a shaft 601 which may be turning at a high velocity has connected to it multiple thin blades as described in figure 400 above. These blades 614 are fixed to the shaft 601 via a shoulder and bolt 616 and are separated from one another by precision spacing rings 615. The porous bearing seals 604 are connected to the Stator 603 via a shoulder and bolt 612. The porous bearing seals 604 are also separated by precision spacers 605 approximately the same size or slightly thicker, but preferably not more than 10 μ thicker, than the Blade Runners. There is clearance 610 between the inside diameter of the porous bearing seals and the outside diameter of the shaft. There is complementary clearance 611 between the outside diameter of the Blade runners and the inside diameter of the Stator. This clearance provides for radial motion of the shaft. If there exists a pressure differential between volume 602 and volume 609, for instance a higher pressure in volume 602, that pressure will act against the first blade runner urging it against the first porous bearing seal. But because higher pressure is being introduced through ports 606 and this pressure is conducted circumferentially by groove 607 and then radially through the porous bearing seal by radial hole 608. This pressure then conducts through the porous media and the face between the blade and the bearing creating a separating force that is also a seal.

[0086] Regarding illustration 6B; this embodiment is likely pertinent to aero engines as may be found on jet airplanes and or gas turbines that are employing brush or centrifugal seals. These contact type seals are a maintenance issue, they create friction and heat which cause efficiency losses and they are noisy. These issues are in large part solved by employing porous carbon air bearing technology. Bearing technology is taught in multiple other locations within the specification. The specific arrangement has a turbine shaft 651 fitted with a mechanism to retain runners which are flexure mounted to the shaft using parallel flexure technology. These runners 656 cooperate with a stationary air bearing seal 653 which in this preferred embodiment uses porous media compensation 654. The stationary part of the seal is mounted to the engine/compressor/generator housing 655 using conventional techniques similar to what would have been employed to mount the stationary section of the friction based seal. Spacers 657 are used to approximately locate the runners axially with the stationary parts of the seal add key off of 652 that is connected to the shaft and secured by 659. Parallel flexures 658 allow the runner to translate axially with respect to the shaft, which will happen for instance under the acceleration of take off, and yet remain parallel to the face of the stationary part of the seal.

[0087] Figure 6C is a close-up of the flexure seal runner shows the bearing face 661, the flexure components 662, one of the through holes for mounting 663 and the area 664 which was either machined, ground or EDM away from a solid stainless steel blank. There may be other ways to manufacture a flexure-based runner.

[0088] Methods for providing additional compliance; Starting with the simpler embodiments; in figures 7A and 7B we have a shaft 701 of a piece of equipment that is carrying with it a runner 711 that has a spherical outside diameter. The runner with a spherical OD couples to the shaft through two O-rings 702. This is advantageous because many shafts have experienced damage and/or out of round at their ends, keyways will often have raised edges, these high spots can damage a precision air bearing/sealing surface while being slid over these damaged features and into position. O-rings can tolerate these types of high spots due to their resiliency. Another advantage is that it reduces tolerance concerns for the fit at 705. This may not be an issue if a cartridge seal employing this technology is associated with an accompanying sleeve for the shaft that goes with the cartridge seal. The OD of the spherical runner mates to complementarily shaped spherical air bearings, which in the preferred embodiment would be porous media restricted. Spherical air bearings are mounted in yoke 712 which is split vertically, split not shown, and air is fed in to the back of the porous restrictive elements 703 through air input port 706 and distribution labyrinth 704. Using this technique as taught will provide for an air gap with several bar's pressure between the porous carbon restrictive element 703 the OD of the spherical runner 711. This air film provides for a frictionless and ware free way of providing angular freedom to the shaft and avoiding over constraint from angular changes in the shaft as indicated by 713, 707 and 710. 709 provides a vent in between the two spherical bearings, this avoids a pressure buildup between the two bearing elements and so the bearings see more pressure drop and their performance increased.

[0089] With reference to image 7C. The porous media restrictive element 751 is shrunk fit in to the nonporous housing 752 which may be made of aluminum or steel or stainless steel or some other suitable material. A plenum 753 comprising approximately 50% of the surface area between the housing and the porous media and having a conductance at least 10 times that of the free flow through the porous media may be disposed on the ID of the housing or the OD of the porous media or some of both. An air feed hole 754 to the plenum provides air flow to the plenum and then through the porous media and into the bearing gap. The air bearing gap provides an axial degree of freedom, but as noted above in areas where a precision shaft is not available may be wise to rely on the O-rings illustrated in 700 or the use of a sleeve for the shaft that comes with the seal cartridge. It should be noted that in embodiment 750 the shaft may spin within journal gas bearing as well still leaving axial shaft freedom.

[0090] Figures 8A and 8B show an embodiment appropriate for providing freedom for frictionless radial displacements of the shaft. This is accomplished by taking the yoke 824 and 817 also described in 700 and 750 and suspending it between thrust faces similar to those in figure 200. The yoke 824 is keyed with an anti-rotation to pin (not shown) to keep it from rotating with the shaft. This anti-rotation pin is provided with enough clearance that allows free motion of the components over the limited range compliance is intended. This yoke is split at 821 and O-ring seal 820 is employed. A thrust plate or collar 806 of appropriate strength for the application is provided with porting 807 and plenum 805 for distributing the air pressure to the back of the porous media 816. The thrust collars 806 may be sealed at joint 819 by an O-ring in a groove such as 818. The spherical gas bearings have a plenum 804 and air input port 808 and a restrictive porous element 803 also as described before in Figure 700 and 750. In this embodiment though, there is no vent between the spherical bearings, instead the high pressure developed in this region is used to conduct pressure through to the Journal bearing in a non-contact fashion. External pressurization ported through 810 in stationary housing 822 then to the Plenum area 823, which is sealed at each side by the thrust bearings, and then through port 809 into the area between the spherical bearings where it can pass in a noncontact fashion through port 811 Into Plenum as described in 753, then through the restrictive element 802 into the gap between the rotor of the restrictive element bearing face at 814.

[0091] This embodiment provides for axial freedom of the shaft, angular freedom of the shaft, and radial displacements of the shaft in a frictionless manner using bearings which are also seals in all motion locations.

[0092] The shaft 801 may spin and move axially within the bearing element 802 and the runner 826. The runner is not coupled to anything except through air bearing films and so it may rotate also. This would allow them to share the speed of the shaft, so for a shaft spinning at 20,000 RPMs 10,000 RPMs could be taken by the Journal bearing 814 if the runner were spinning at 10,000 RPMs and the other 10,000 RPMs could be taken between the spherical bearings 803, 815 and the runner 826.

[0093] The difference between figures 8A and 8B and figures 9A and 9B is that porting for the opposed axial air bearings which provide friction free radial motion for the shaft have had the porting 908 and the Plenum 905 moved internal to the yoke 924 from the thrust plates 906 where they were in figure 800. This simplifies the manufacturing of the thrust plates 906 and allows them to retain a higher stiffness for their given axial thickness, because the yoke is seeing a compressive load in what amounts to column stiffness where the thrust cap sees a cantilevered bending stiffness which is not as strong. It should be noted that in each of the bearing arrangements in figure 800 and figure 900 that the interface between the porous media gas bearing surface and the guide way it acts upon 916, 915 and 914 only one side of the bearing elements are open to ambient pressure, this reduces the effect of the externally pressurized air bearings but the bearings do retain significant load capacities.

[0094] The difference between figures 8A-9B and figure 10 is that figure 10 is fully vented, that is all of the gas bearings see the full pressure drop between the external input pressure and ambient or a process pressures that exist at the escape edges of the bearings. So the external pressure is ported into the thrust plates 1008 via 1009 and distributed behind the porous media restrictive element by Plenum 1007. Additionally a through hole 1005 has been drilled through the porous media and directly in to the Plenum 1007. This whole aligns roughly with a hole in the yoke 1006, which has a counter bore 1004 that maintains conductance between 1005 and 1006 during the designed displacements of this compliance device. It stays aligned due to the anti-rotation pin discussed in figure 800. Hole 1006 provides conductance of pressure to both the spherical bearing element 1003 and the Journal porous bearing element 1002 via the cross hole 1010 drilled in the yoke 1028 which is later plugged 1011. 1010 delivers the pressure and flow to the Plenum 1020 providing the external pressurization for the spherical restrictive elements/ bearings 1020. Cross hole 1010 also communicates with hole 1012 which is threaded to accept a fitting, the fitting is connected to a flexible tube which provides motion compliance in the conductance of pressure to be spherical runner which is now also keyed to the yoke to provide anti-rotation relative to the yoke. The pressure and flow for the journal restrictive element 1002 are provided through the fittings 1014, 1016 via the tube 1015 and into Plenum 1030.

[0095] An annular groove in the center of the Journal gas bearing portion provides even communication of the bearing/seal flow to a center vent hole. This is the radial hole in the spherical runner 1028 and is shown next to fitting 1016. This whole exhausts in the space between the spherical bearings and both of these bearings are able to exhausts through the hole that the fittings 1014, 1016 and tube 1015 partially consume. Hole 1013 through housing 1026 provides the exhaust for these flows plus flow from the opposed axial faces 1021.

[0096] In order to effect a seal, as shown in Figure 11A, a porous material 1102 which comprises one side of two opposing surfaces is used to evenly distribute hydraulic pressure from an external source of pressurized fluid (gas, liquid, steam, ect.) between the two surfaces. The pressure is ported through 1106 to Plenum 1108 then through the porous media 1102 and into the gap 1107. This hydrostatic pressure creates a force which is opposite the forces from pressure differences or Springs trying to close the two faces together, the other face being the bearing seal side of 1110. Also see figure 100 for teaching regarding this illustration 1100. This hydrostatic pressure may be adjusted to the point where the two faces are completely unloaded and zero contact pressure exists between the two faces even though the faces are in intimate contact 1107. Because the faces are in contact there is approximately zero flow through the gap and the line pressure being fed into the porous material will exist between the two faces.

[0097] So in another illustrative example with reference to figure 11B, if there is 1000 pounds of force or in this illustration a 1000 pounds of mass represented by 1124 urging the face of the seal body 1123 together with the counter face 1121 and the seal faces have 10 $in.^2$ of area between them and 100 PSI air pressure is fed in at port 1125 and this pressure is distributed cross the back of the porous medium using a Plenum as taught multiple times before in this specification, the porous seal face will have exactly zero contact force between the faces as the hydrostatic force between the faces will equal the mass or force urging the seal faces together. This contact force can easily be adjusted by varying the input pressure to reduce ware and heat generated by friction.

[0098] This technique combines the high stiffness and damping of plane bearings and contact seals with the low friction and high speed capability of fluid film bearings and seals.

[0099] The porous media may be comprised of graphite, carbon, silicon carbide, Tungsten carbide, alumina or basically any porous and or sintered material. These materials are typically found as face seals and mechanical seals and as runners and runner faces in dry gas seals. Just instead of filling or sealing this porosity the porosity is used to conduct and evenly distribute hydrostatic pressure.

[00100] Orifice pocketed or step type air bearing compensations will not work in this application as only a uniform porous media is capable of evenly distributing a hydrostatic pressure with zero gap. For instance, if orifices were employed, when the faces were in contact hydrostatic pressure would only be exerted over the area of the orifices.

[00101] With reference to figure 12, this is an illustration of a solid carbon graphite tilting pad radial air bearing 1201. When manufacturing it from a single part rather than attempting to laminate two parts together the trouble of joining two components together for use under extreme temperatures is avoided. Most carbon graphite will not start to oxidize until it is in an environment over 800° C, so this provides a very extensive temperature range. In this case a Plenum to distribute the air to the back of the porous media face is accomplished by drilling cross holes 1203. These cross holes are threaded and plugged 1204 with high temperature ceramic or glaze which is later fired so that it becomes co-sintered with the carbon graphite. A ceramic insert from a metal cutting tool is sintered in at 1206 at the same time to distribute the load of the Hertzian contact of the tilting pad mechanism. 1205 represents a high temperature fitting known in the art. 1202 represents a diameter which would be complementary to a shaft that such a bearing would support.

[00102] It is also possible to co-fire separate ceramic components, for instance a nonporous housing with a porous media face. The co-firing essentially makes a monolithic part but there was an opportunity to machine plenums or labyrinths into the green parts before they were sintered or fired together. Alternatively a glass bonding, similar to a glazing operation done on the outside of a piece of pottery may be employed as a high temperature glue to bond separate ceramic components into a single high temperature part that could be used as a bearing or a seal in extreme temperature environments

[00103] While preferred embodiments have been set forth in detail with reference to the drawings, those skilled in the art who have reviewed the present disclosure will readily appreciate that other embodiments can be realized within the scope of the invention, which should therefore be construed as limited only by the appended claims.

What is claimed is:

1. An aerostatic or hydrostatic bearing seal assembly comprising:

a rotatable shaft with a runner coupled to the rotatable shaft;

a housing located concentric to the shaft and defining a cavity, the housing being coupled to an annular seal body via a first **O**-ring seal, the seal body including a conductive passage for communicating pressurized fluid to a labyrinth which distributes the pressurized fluid to a porous media positioned between the runner and the seal body, the pressurized fluid creating an annular air film between the rotatable shaft and seal body.

2. The aerostatic or hydrostatic bearing seal assembly of claim 1, wherein a second **O**-ring seal couples the rotatable shaft to the runner.

3. The aerostatic or hydrostatic bearing seal assembly of claim 1, wherein the porous media is selected from a group of graphite, carbon, silicon carbide, Tungsten carbide, alumina, and combinations thereof.

4. The aerostatic or hydrostatic bearing seal assembly of claim 1, wherein an adapter plate is positioned between the housing and the seal body, and a third **O**-ring is positioned between the adapter plate and the seal body.

5. An aerostatic or hydrostatic bearing seal assembly comprising:

a rotatable shaft with a runner coupled to the rotatable shaft;

a housing located concentric to the shaft that defines a cavity,

an adapter plate coupled on one side to the housing via a first **O**-ring seal and coupled on an opposite side to a seal body via a second **O**-ring seal,

the seal body including a first and second ring each including porous media having an annular opposed air bearing face, a spacer located concentric to a runner separates the first and second ring,

wherein the first and second ring each include a conductive passage for directing fluid to the porous media, the spacer includes a drain in communication with a space defined between the rings and the runner, and the fluid creates an annular air film between the rotatable shaft and the seal body.

6. The aerostatic or hydrostatic bearing seal assembly of claim 5, wherein a third O-ring seal couples the rotatable shaft to the runner.

7. The aerostatic or hydrostatic bearing seal assembly of claim 5, wherein the porous media is selected from a group of graphite, carbon, silicon carbide, Tungsten carbide, alumina, and combinations thereof.

8. The aerostatic or hydrostatic bearing seal assembly of claim 5, wherein an adapter plate is positioned between the housing and the seal body, and a third O-ring is positioned between the adapter plate and the seal body.

9. An aerostatic or hydrostatic bearing seal assembly comprising:

a rotatable shaft, and

a housing located concentric to the shaft which defines a cavity, the housing is coupled to an adapter block via a first O-ring seal, the adapter block is arranged concentric to a cylindrical seal and is coupled with the cylindrical seal via a second O-ring seal, the adapter block and the cylindrical seal include conductive passages for communicating pressurized fluid to a porous material located between the cylindrical seal and the shaft, the pressurized fluid enters an air gap located between the porous material and the shaft to create an annular seal.

10. The aerostatic or hydrostatic bearing seal assembly of claim 9, wherein a retainer element is located on a low pressure side of the adapter block and clearance is provided between the retainer and the shaft.

11. The aerostatic or hydrostatic bearing seal assembly of claim 9, wherein the adapter block includes a hole and epoxy is inserted through the hole and between a cylindrical gap defined between the adapter block and the cylindrical seal.

12. A blade runner air bearing seal assembly comprising:

a rotatable shaft with a blade runner coupled to the rotatable shaft; and

a housing located concentric to the shaft and defining a cavity, the housing being coupled to an annular seal body via a first O-ring seal, the seal body including a conductive passage for communicating pressurized fluid to a labyrinth which distributes the pressurized fluid to a porous media positioned between the blade runner and the seal body, the pressurized fluid creating an annular air film between the rotatable shaft and seal body.

13. The blade runner air bearing seal assembly of claim 12, wherein a second O-ring seal couples the rotatable shaft to the blade runner.

14. The blade runner air bearing seal assembly of claim 12, wherein the shaft includes a shoulder and the blade runner abuts the shoulder.

15. The blade runner air bearing seal assembly of claim 12, wherein blade runner is between 0.1 and 1.0 mm thick.

16. A multi-blade runner air bearing seal assembly comprising:

a rotatable shaft with a plurality of blade runners coupled to the shaft via a shoulder on the shaft and a first bolt, the plurality of blade runners are separated from each other via spacing rings;

a stator with a plurality of porous bearing seals coupled to the stator via a second bolt, the plurality of porous bearing seals are separated from each other via precision spacers;

a first clearance is defined between an inside diameter of the bearing seals and an outside diameter of the shaft, and a second clearance is defined between an outside diameter of the blade runners and an inside diameter of the stator; and

passageways are provided in the stator that communicate with respective holes in the plurality of bearing seals, pressurized fluid flows from the passageways to a circumferential groove arranged adjacent to the stator, the pressurized fluid also flows through the plurality of bearing seals through respective holes to spaces between the plurality of bearing seals and the plurality of blade runners.

17. The multi-blade runner air bearing seal assembly of claim 16, wherein the precision spacers are approximately the same size as the blade runners.

18. The multi-blade runner air bearing seal assembly of claim 1, wherein the precision spacers are less than 10 μιη thicker than the blade runners.

19. A spherical runner bearing seal assembly comprising:

a rotatable shaft with a spherical runner coupled to the shaft via two O-ring seals, the spherical runner including an outer diameter that mates to a complementarily shaped porous restrictive element, a housing supports the porous restrictive element and includes a passageway for feeding pressurized fluid through the passageway and a labyrinth, a gap between the housing and spherical runner allows the pressurized fluid to create an air film between the porous restrictive element and the outer diameter of the spherical runner, and a vent located between the two O-ring seals in the housing prevents pressure buildup of the pressurized fluid.

20. A spherical runner bearing seal assembly comprising:

a rotatable shaft arranged within a stationary housing with two thrust collars, a yoke is suspended between opposed faces of the thrust collars, the stationary housing includes an input port for pressurized fluid and the thrust collars include secondary ports for the pressurized fluid, the input port communicating with a first passageway in the yoke and feeding into a second passageway in a spherical runner, the secondary ports communicating with chambers and porous media arranged adjacent to the spherical runner, the pressurized fluid flows from the input port and secondary ports to a gap between the spherical runner and the shaft creating an air film for the pressurized fluid.

21. A spherical runner bearing seal assembly comprising:

a rotatable shaft arranged within a stationary housing with two thrust collars, a yoke is suspended between opposed faces of the thrust collars, the stationary housing includes an input port for pressurized fluid and the yoke includes secondary ports for the pressurized fluid, the input port communicating with a first passageway in the yoke and feeding into a second passageway in a spherical runner, the secondary ports communicating with chambers and porous media arranged adjacent to the spherical runner, the pressurized fluid flows from the input port and secondary ports to a gap between the spherical runner and the shaft to create an air film.

22. A spherical runner bearing seal assembly comprising:

a rotatable shaft arranged within a stationary housing with two thrust collars, a yoke suspended between opposed faces of the thrust collars, the thrust collars each include input ports for pressurized fluid and the stationary housing includes an exhaust port for the pressurized fluid, the input ports communicate with a respective chamber and porous media to direct the pressurized fluid into a bore in the yoke, the yoke includes a fitting and tube for directly communicating with a gap between the rotatable shaft and a spherical runner, additional porous media is provided in an area between the yoke and the spherical runner for allowing the pressurized fluid to flow from the input ports to the gap between the rotatable shaft and the spherical to create an air film.

23. A hydrodynamic bearing seal assembly comprising:

a rotatable shaft with a runner coupled to the rotatable shaft;

a housing located concentric to the shaft and defining a cavity, the housing being coupled to an annular seal body via a first O-ring seal;

the seal body including a conductive passage for communicating pressurized fluid to a labyrinth which distributes the pressurized fluid to a porous media positioned between the runner and the seal body and the pressurized fluid exerts a force on a face of the porous media; and

an element provides a opposing force to the pressurized fluid force, such that the face of the porous media and a face of the runner are completely unloaded and zero contact pressure exists between the porous media face and the runner face.

24. A aerostatic or hydrostatic bearing seal assembly comprising:

a rotatable shaft;

a housing located concentric to the rotatable shaft, the housing including:

a cavity, and

axial air bearings located at each end the cavity, the air bearings supporting the rotatable shaft on an air film between the rotatable shaft and sealing a gas inside the cavity.

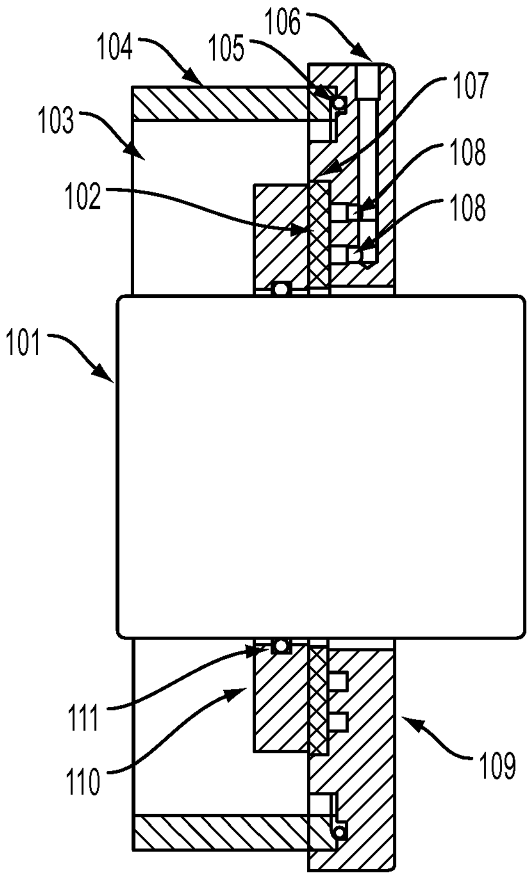
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FIG. 1A

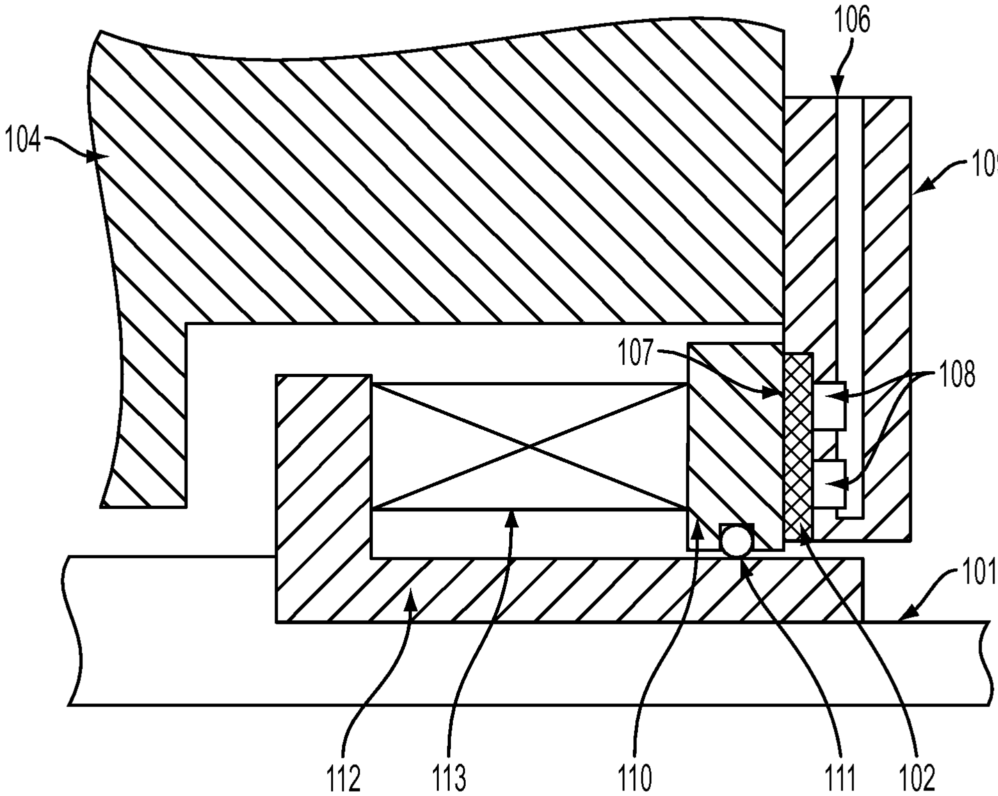
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FIG. 1B

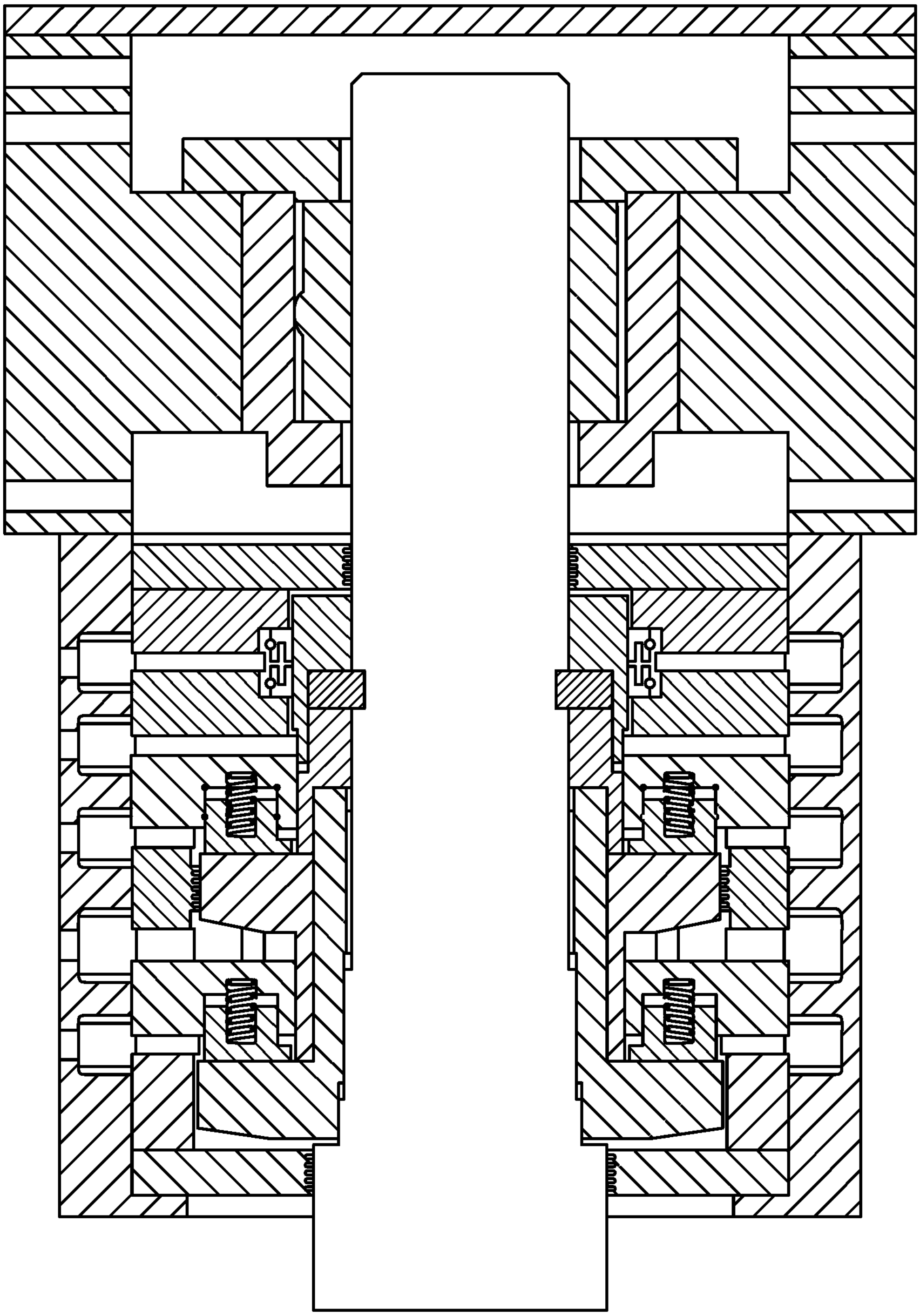

FIG. 1C

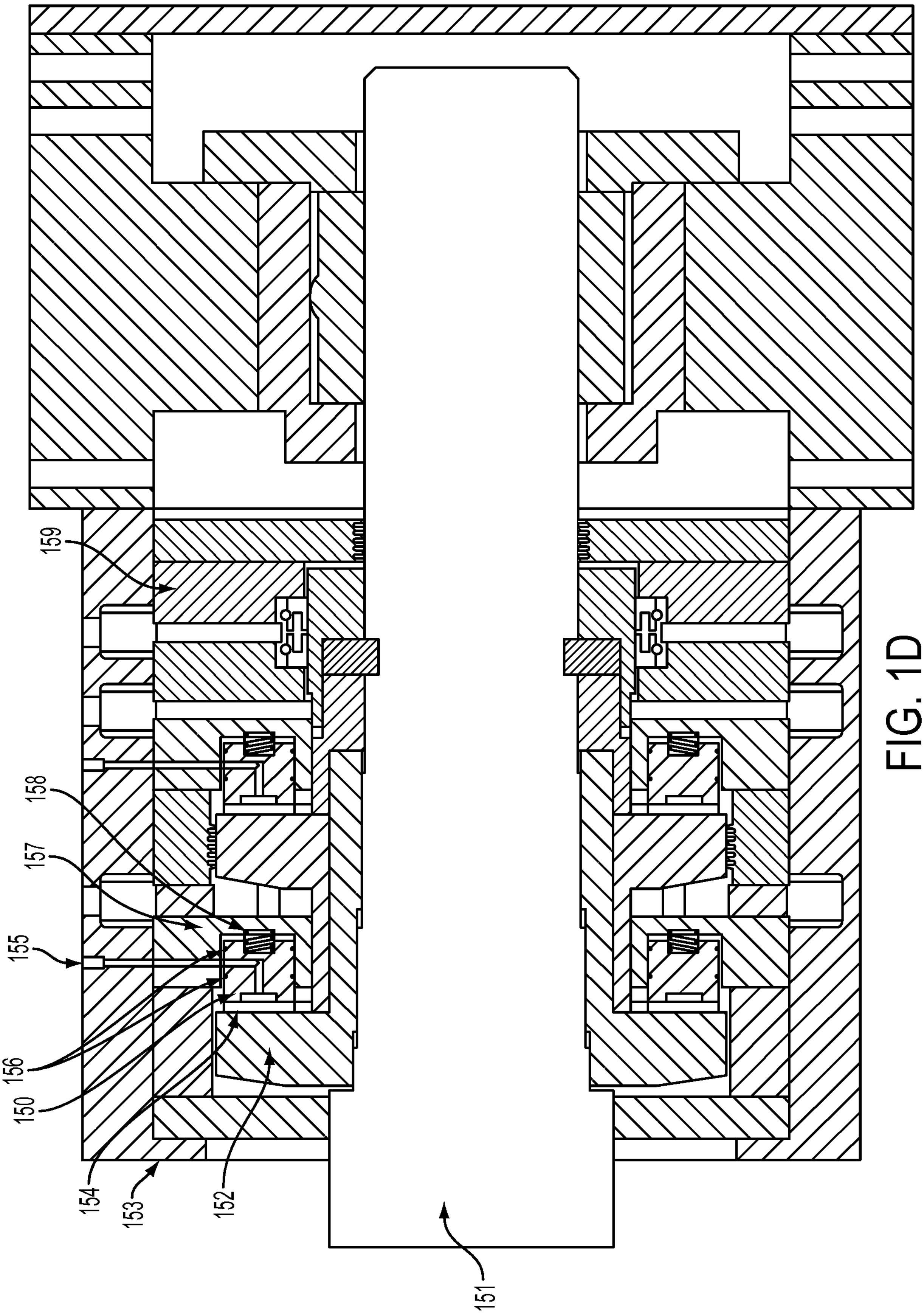

FIG. 1D

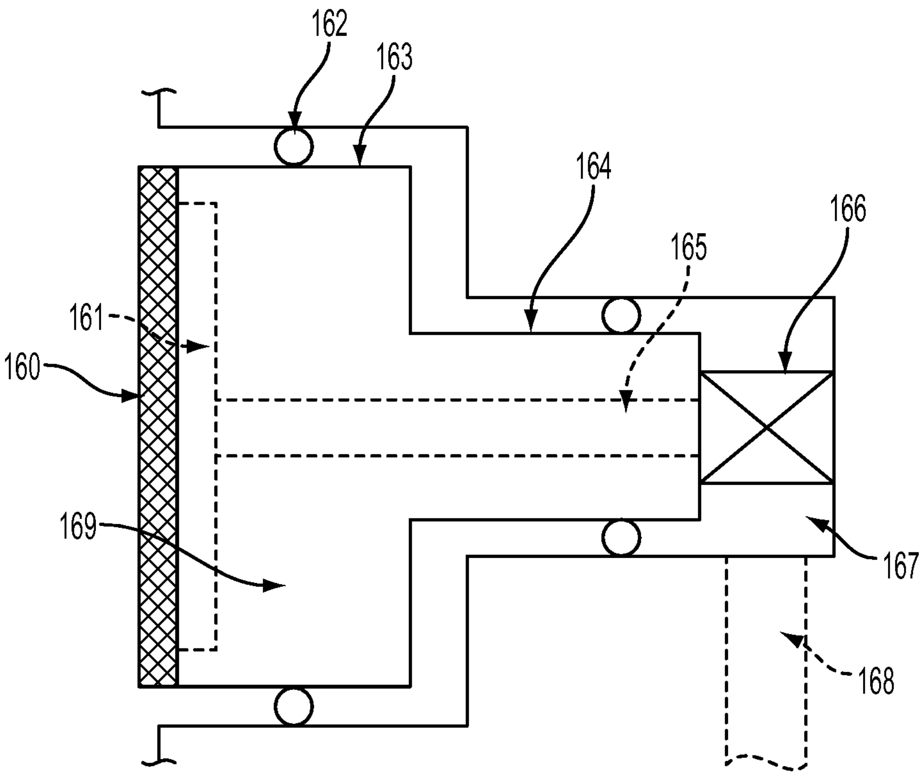
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FIG. 1E

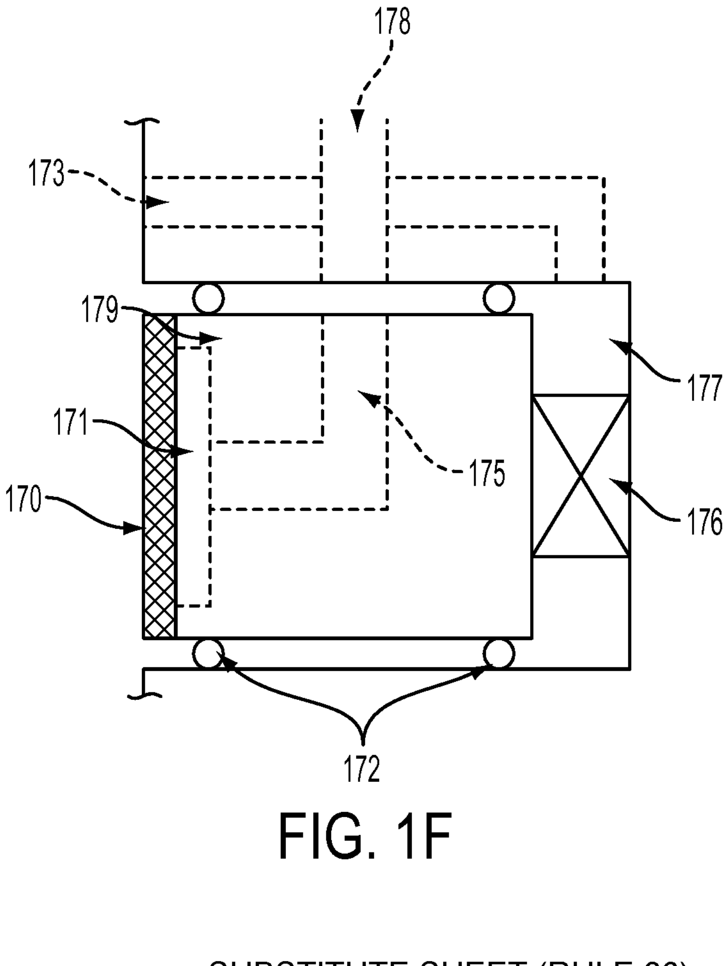
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FIG. 1F

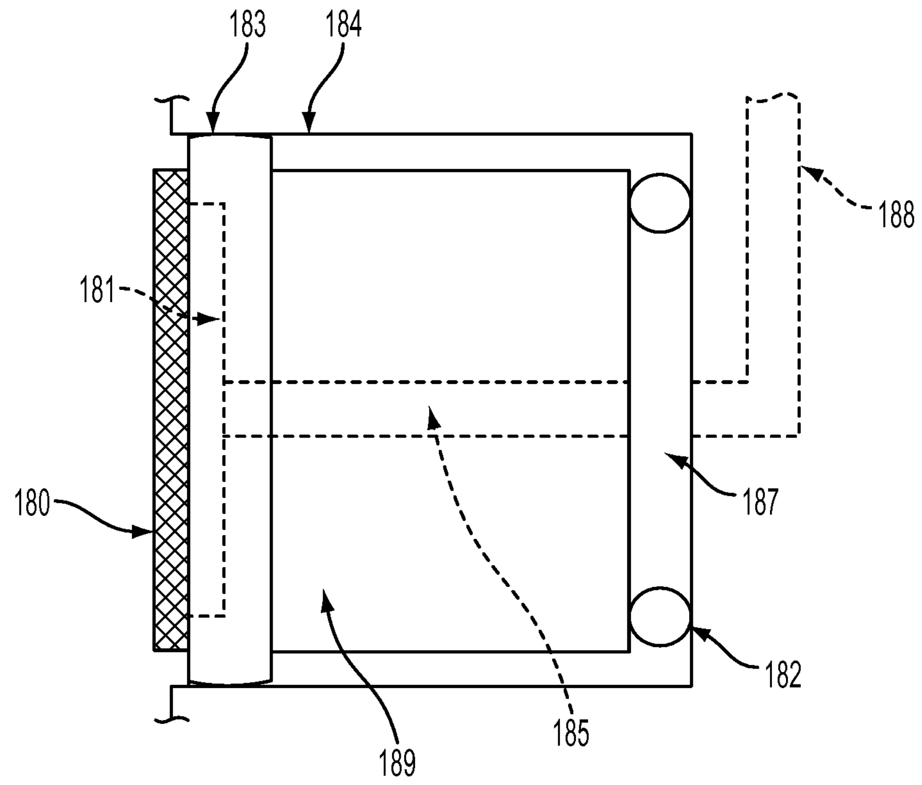

FIG. 1G

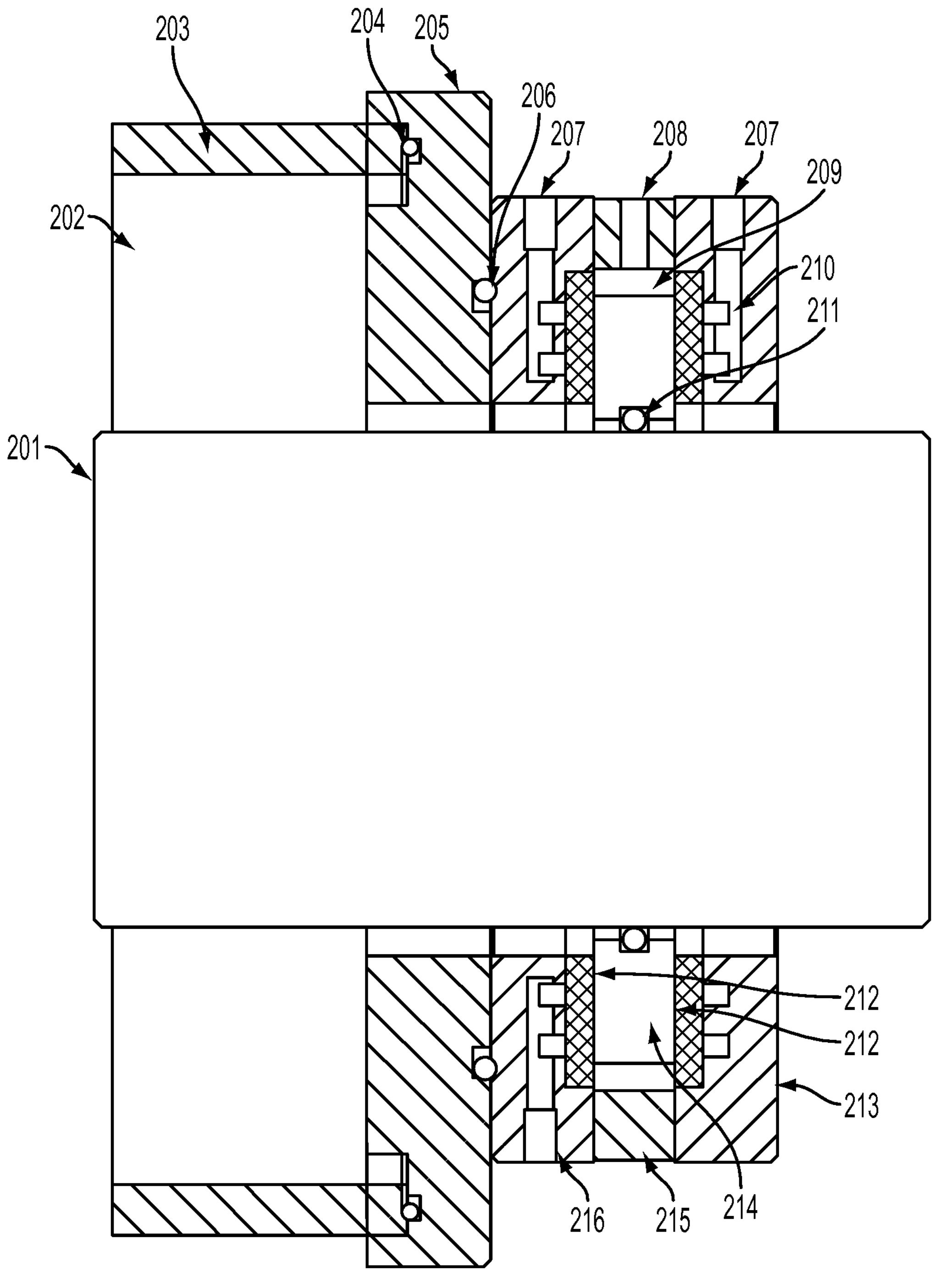
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FIG. 2A

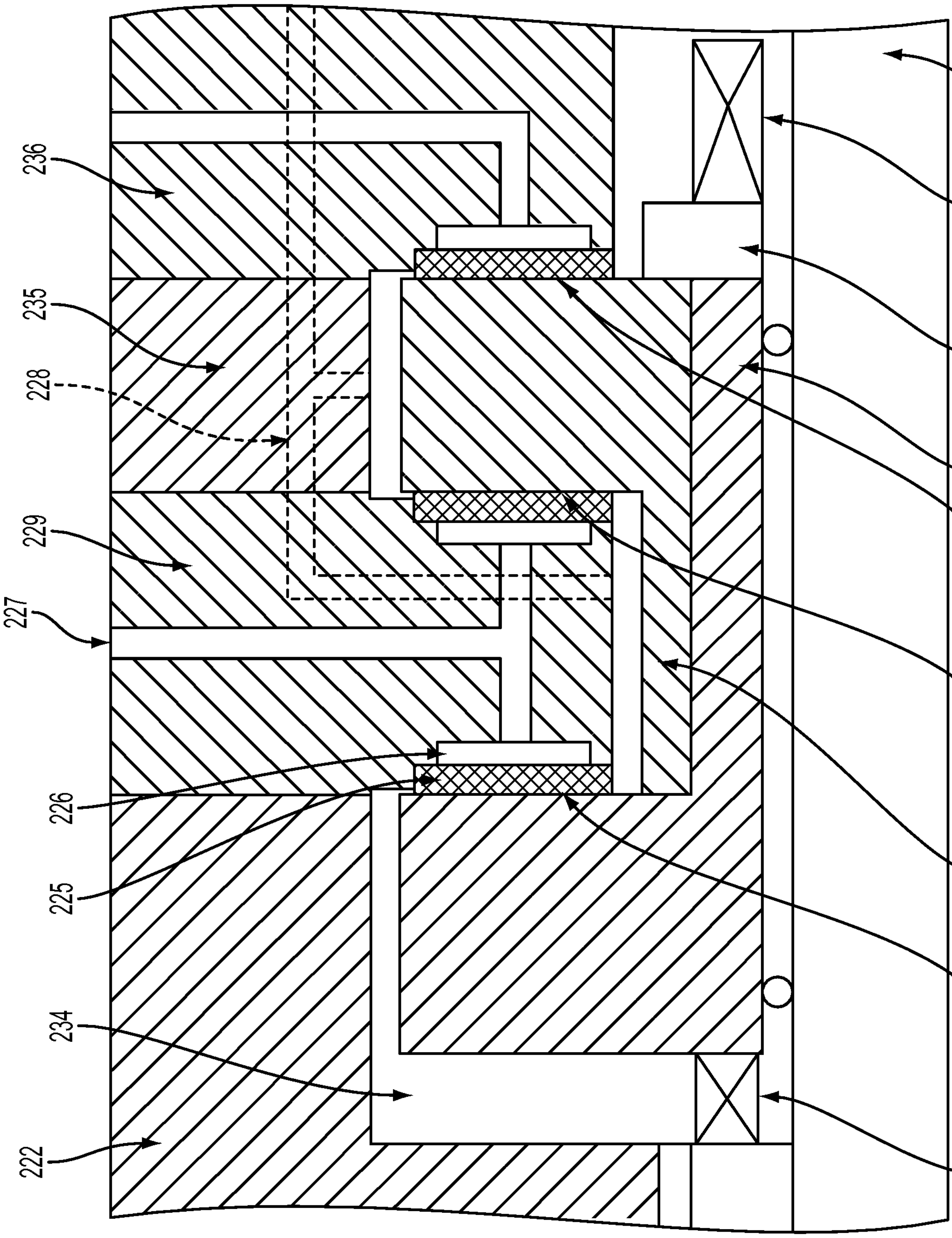
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FIG. 2B

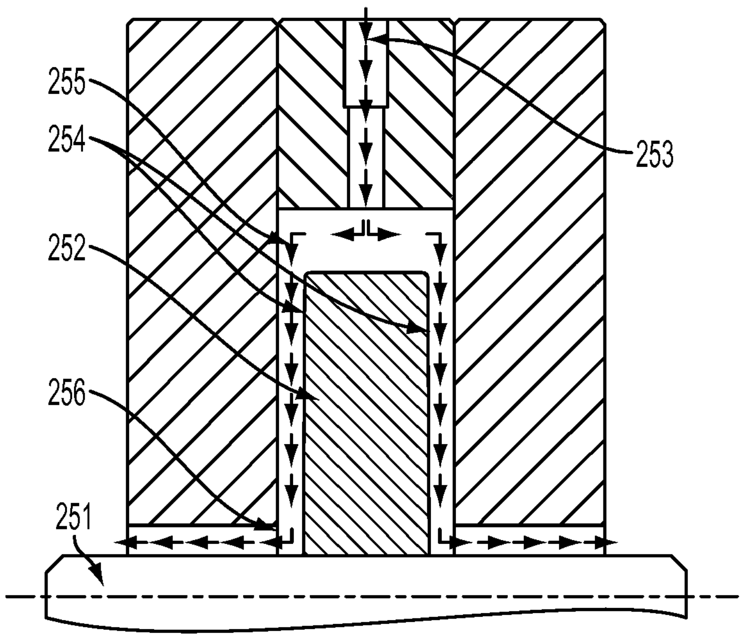

FIG. 2C-1

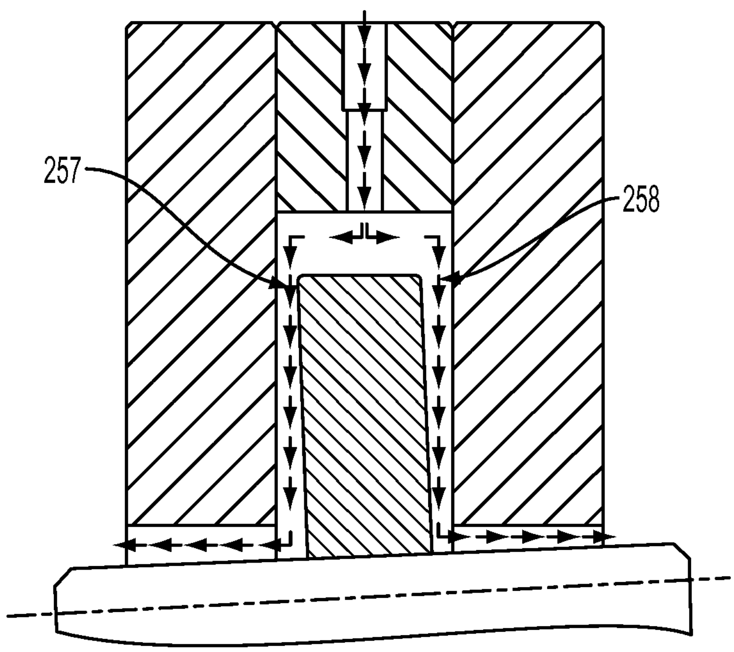

FIG. 2C-2

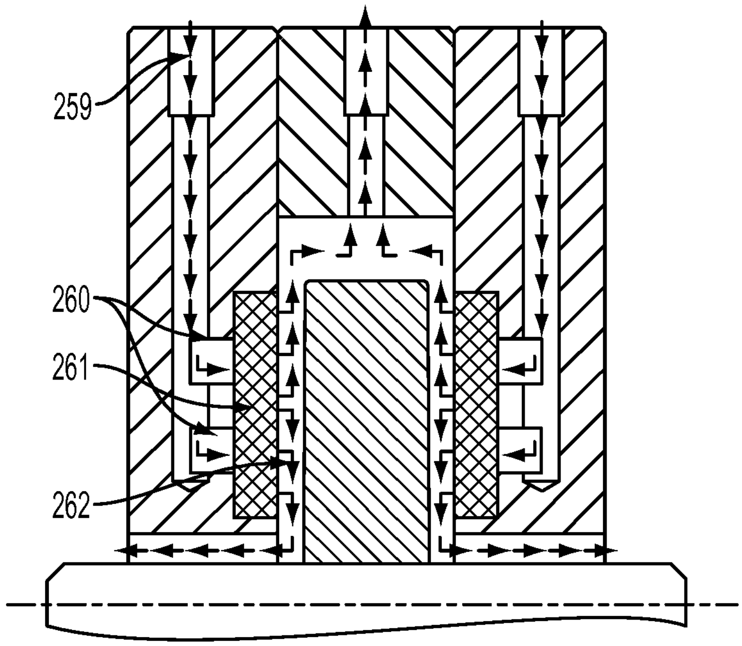

FIG. 2C-3

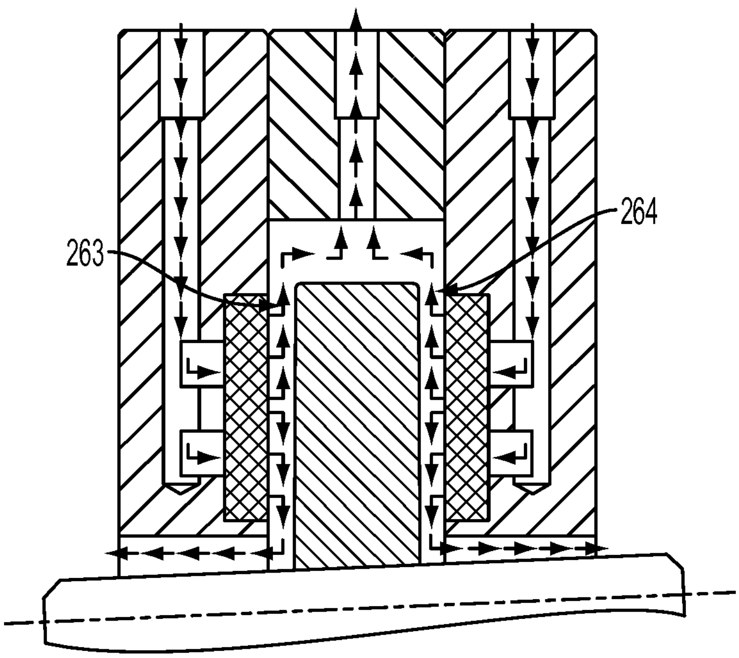

FIG. 2C-4

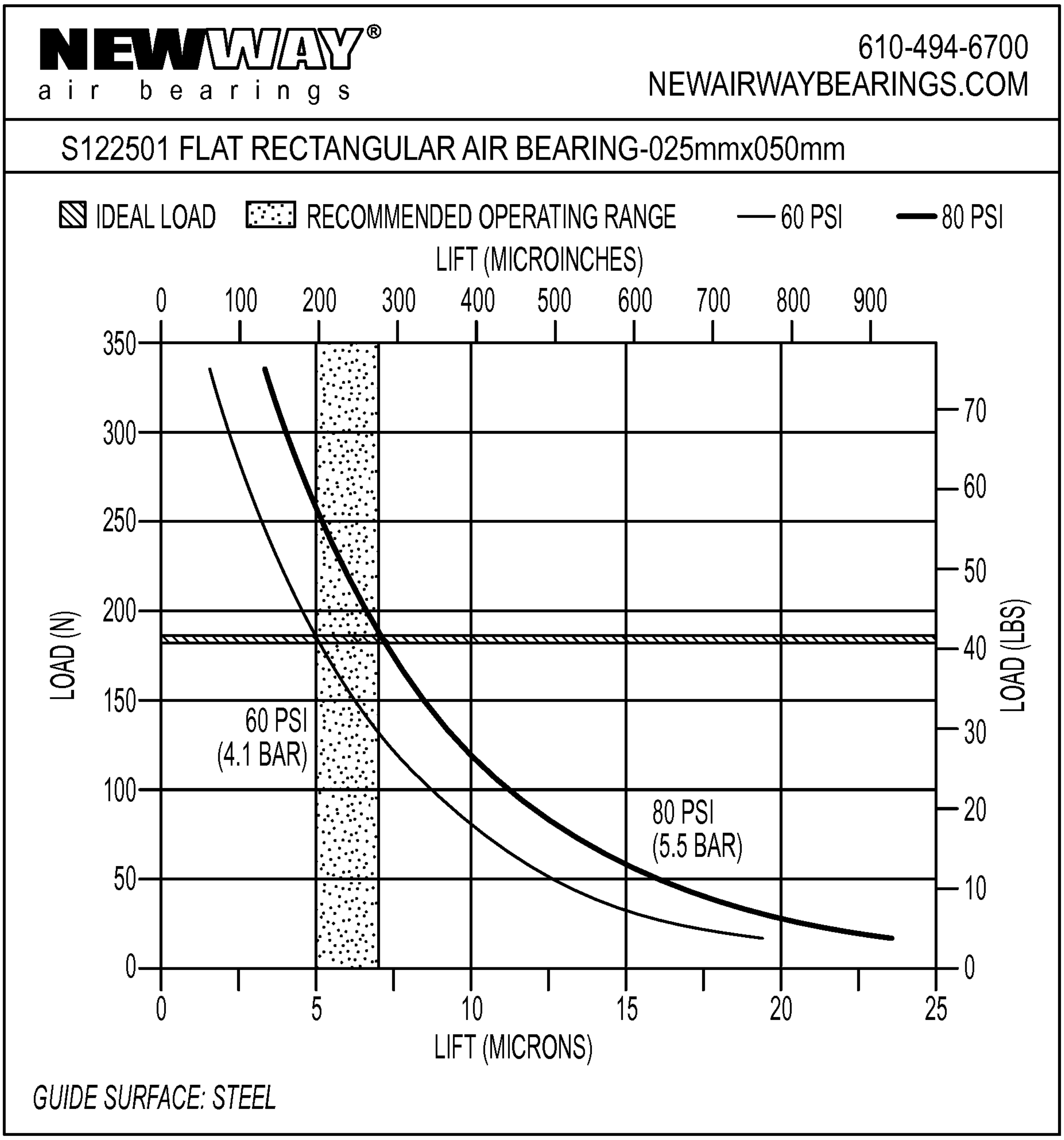
NEWWAY®
air bearings
610-494-6700
NEWAIRWAYBEARINGS.COM
S122501 FLAT RECTANGULAR AIR BEARING-025mmx050mm
IDEAL LOAD
RECOMMENDED OPERATING RANGE
60 PSI
80 PSI
LIFT (MICROINCHES)
0
100
200
300
400
500
600
700
800
900
LOAD (N)
350
300
250
200
150
100
50
0
LOAD (LBS)
70
60
50
40
30
20
10
0
60 PSI
(4.1 BAR)
80 PSI
(5.5 BAR)
0
5
10
15
20
25
LIFT (MICRONS)
GUIDE SURFACE: STEEL

FIG. 2D

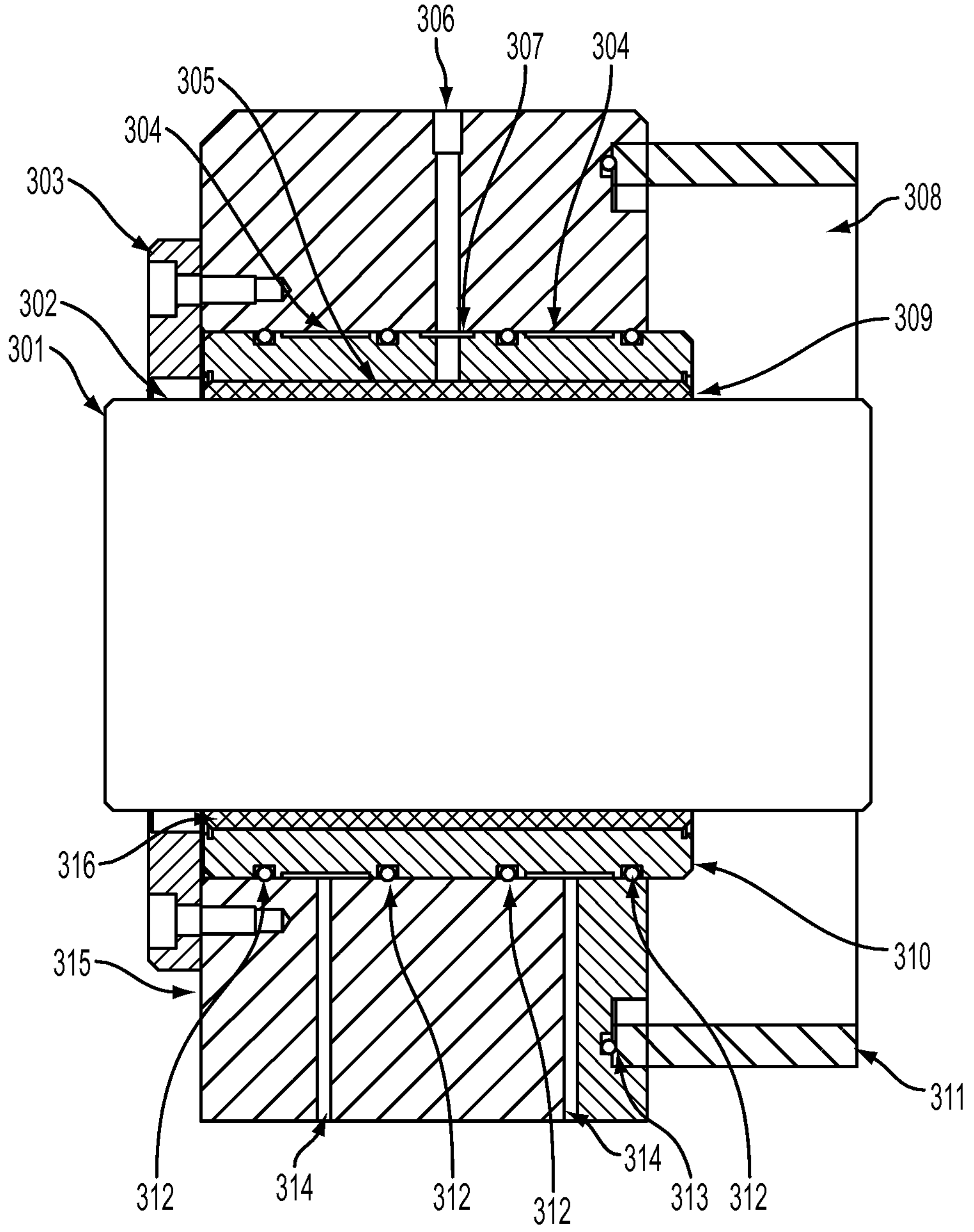
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FIG. 3A

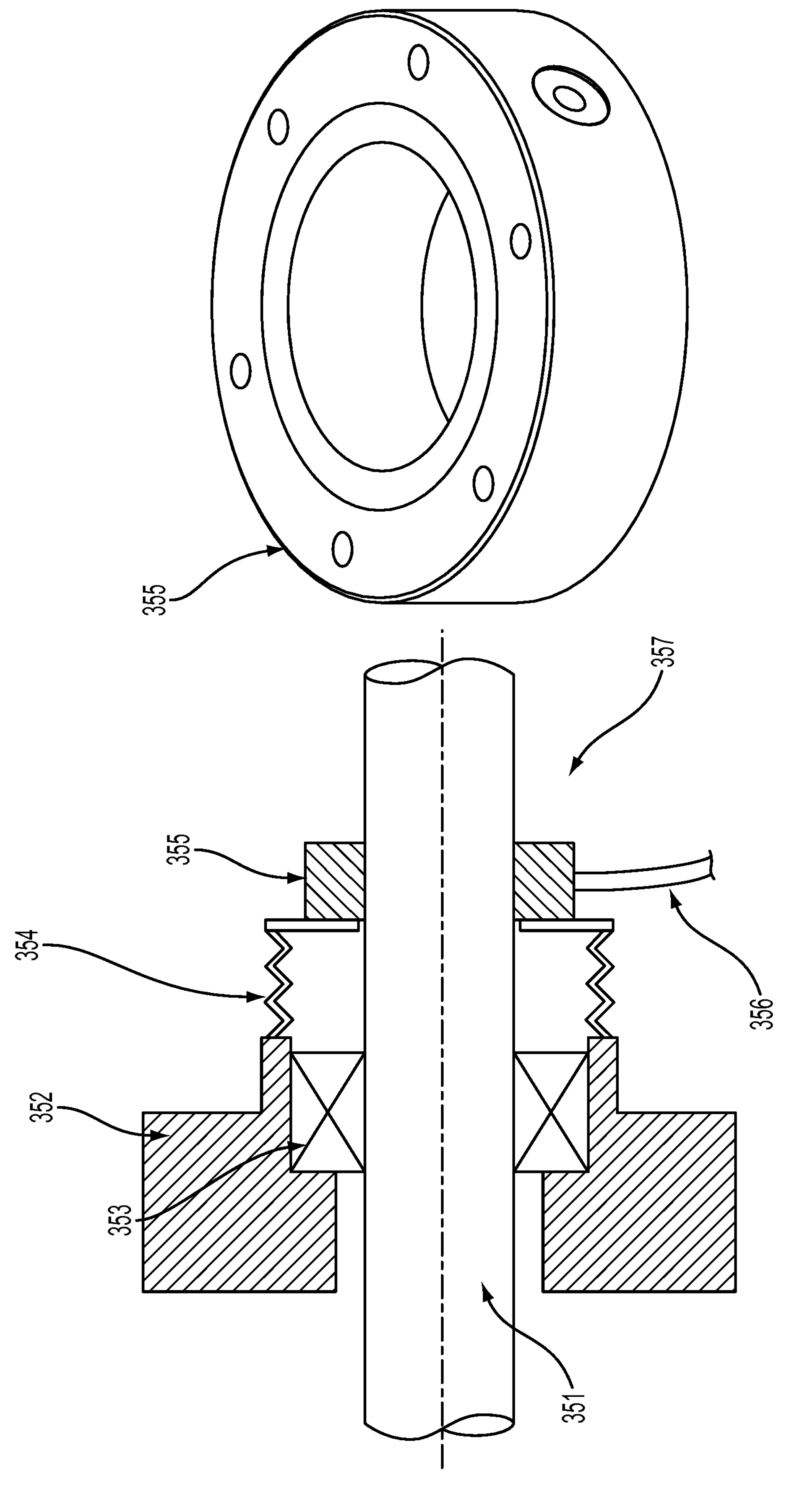

FIG. 3B

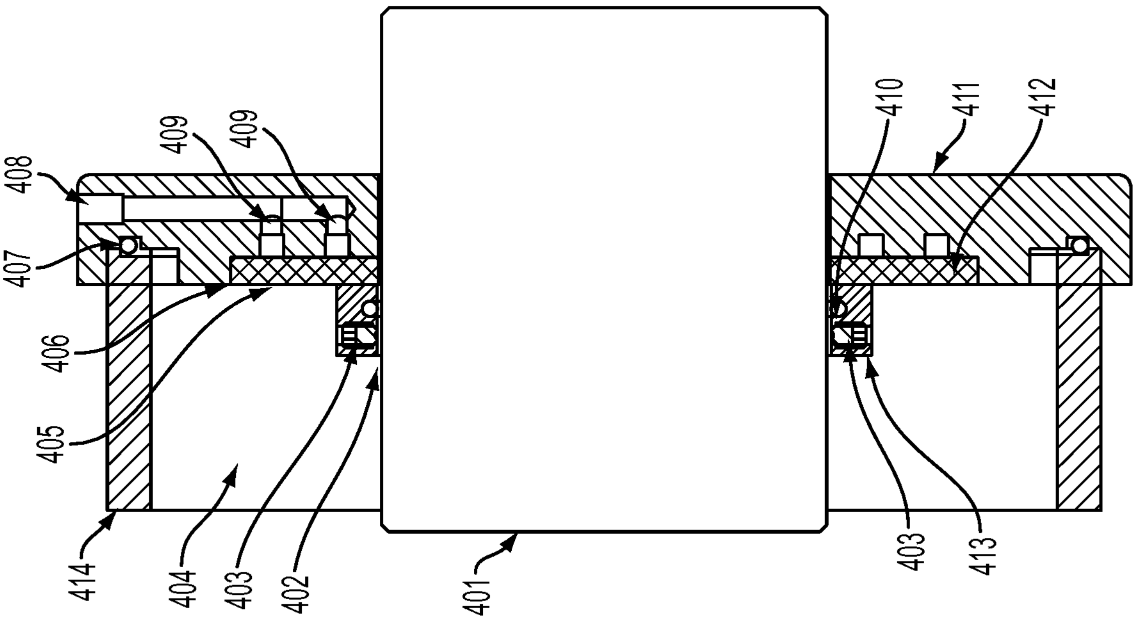
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FIG. 4B

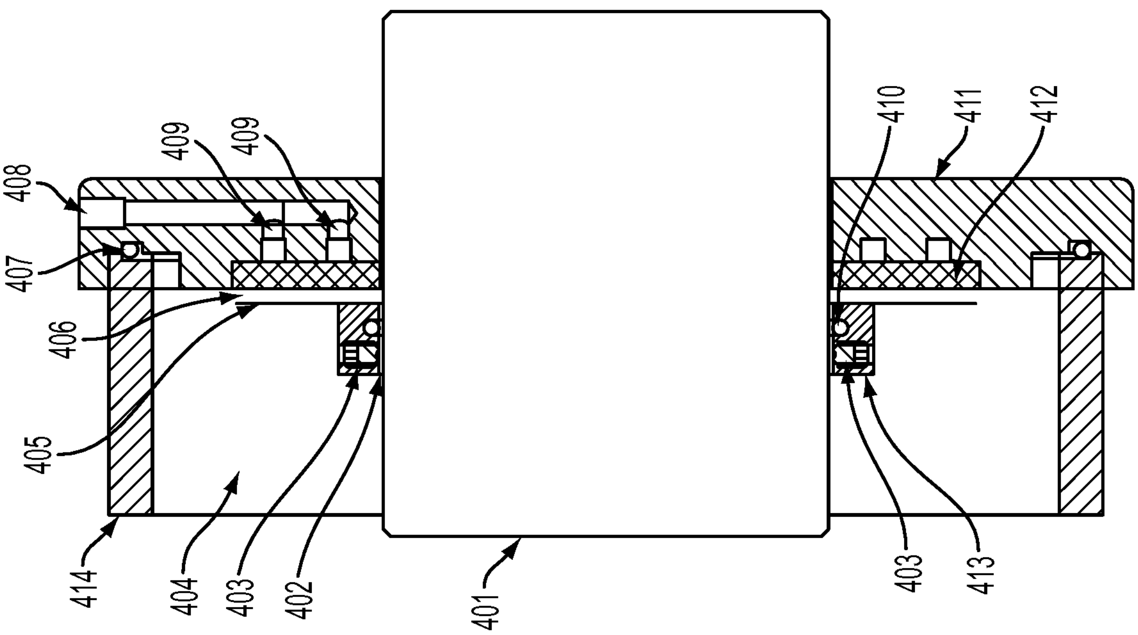
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FIG. 4A

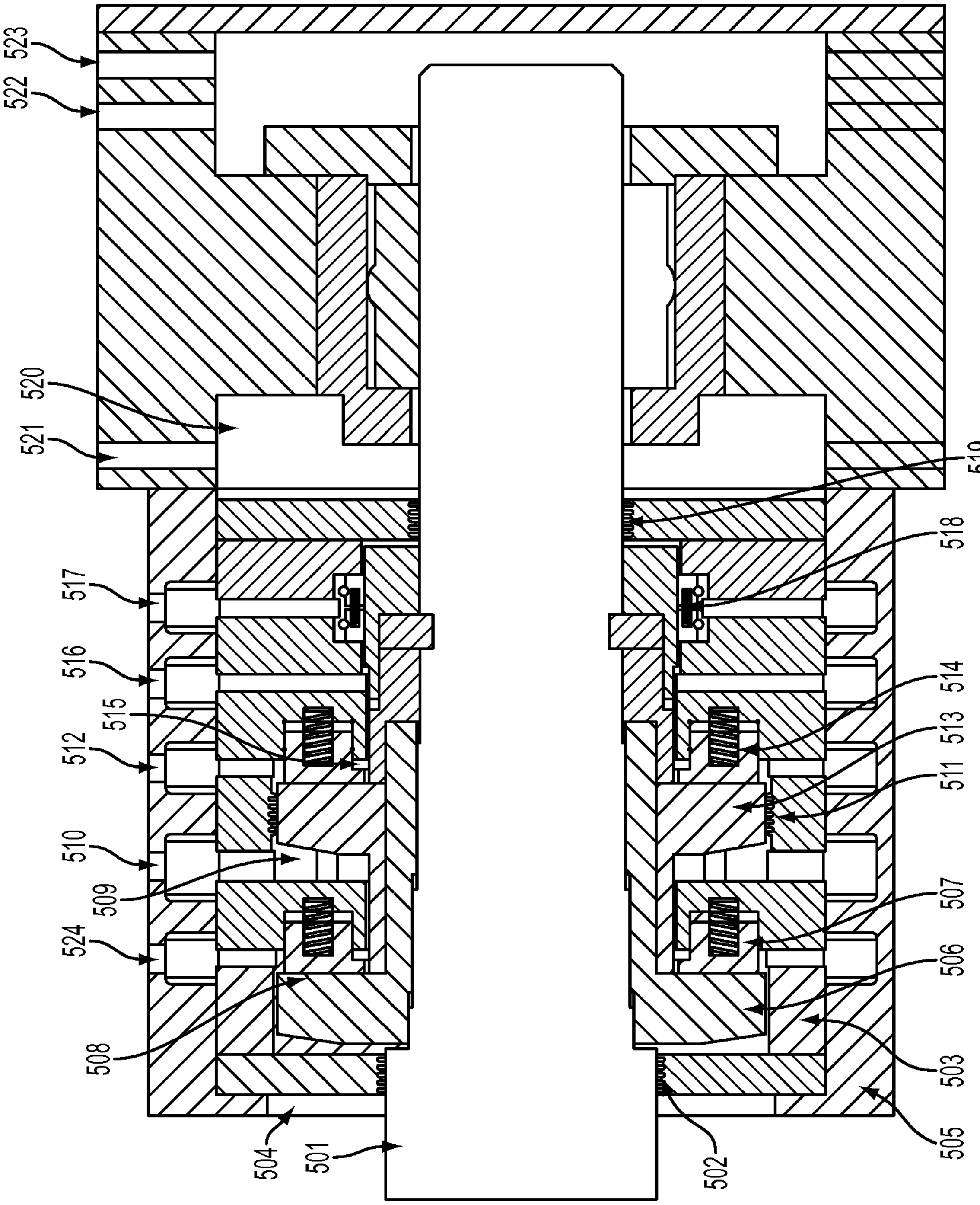

FIG. 5A
PRIOR ART

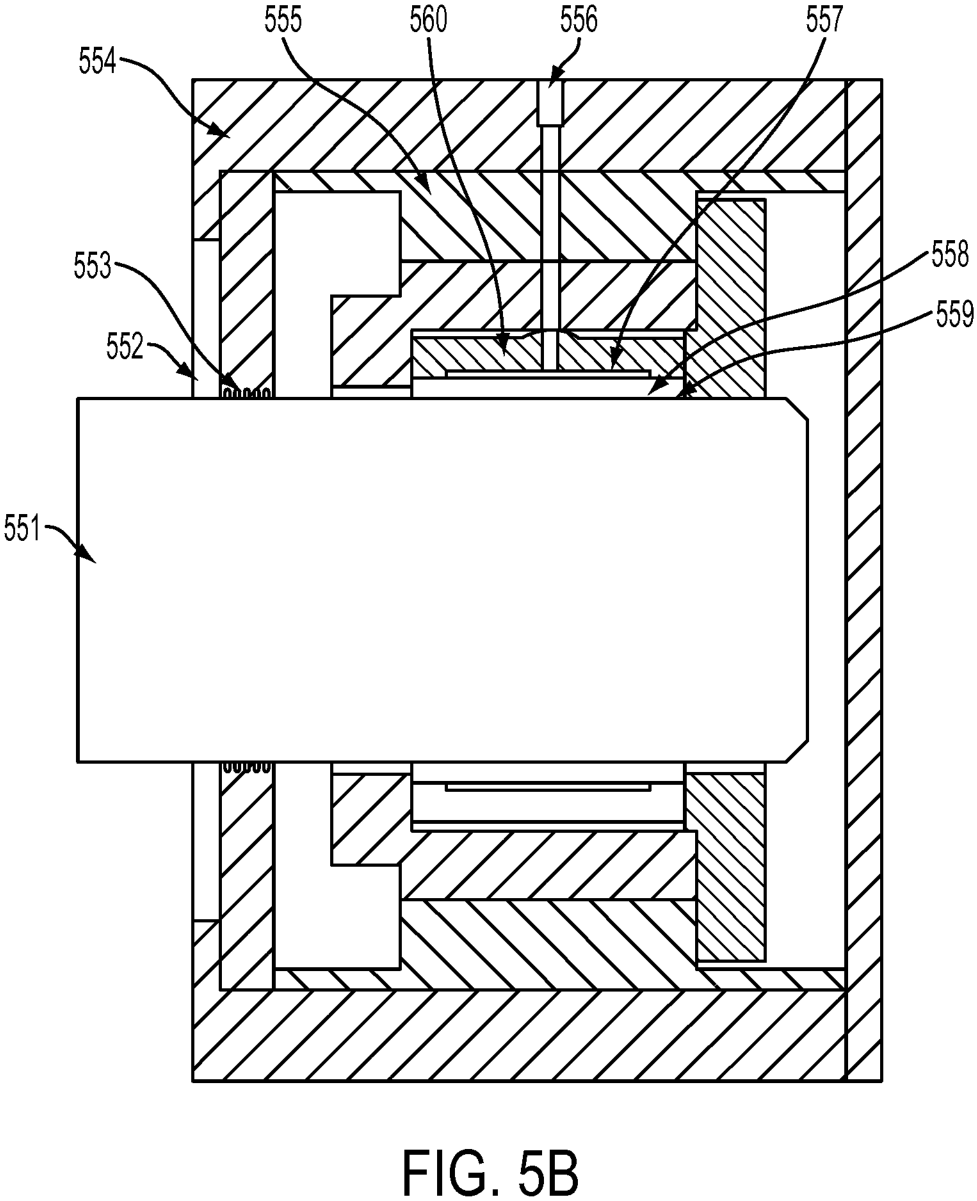

FIG. 5B

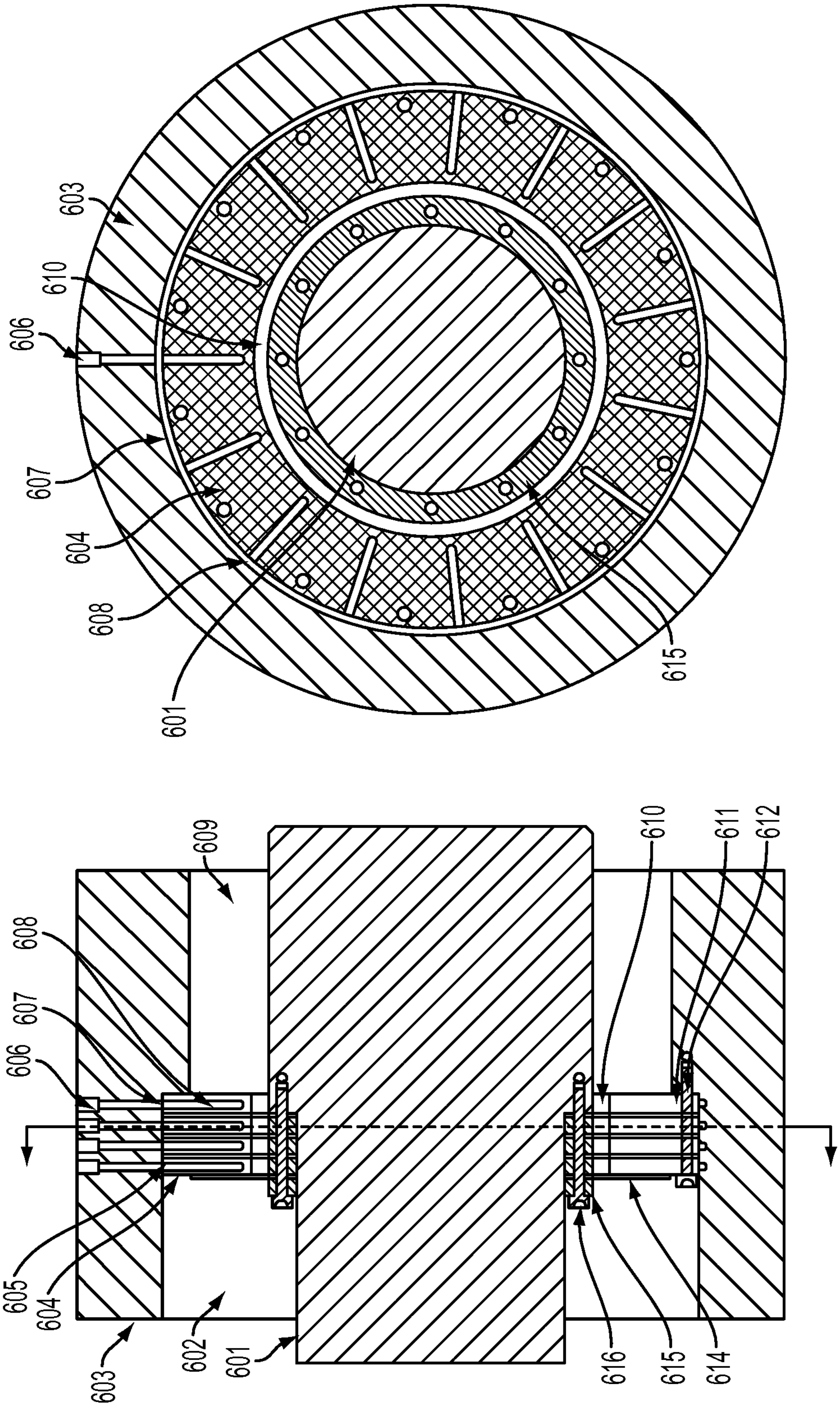
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FIG. 6A

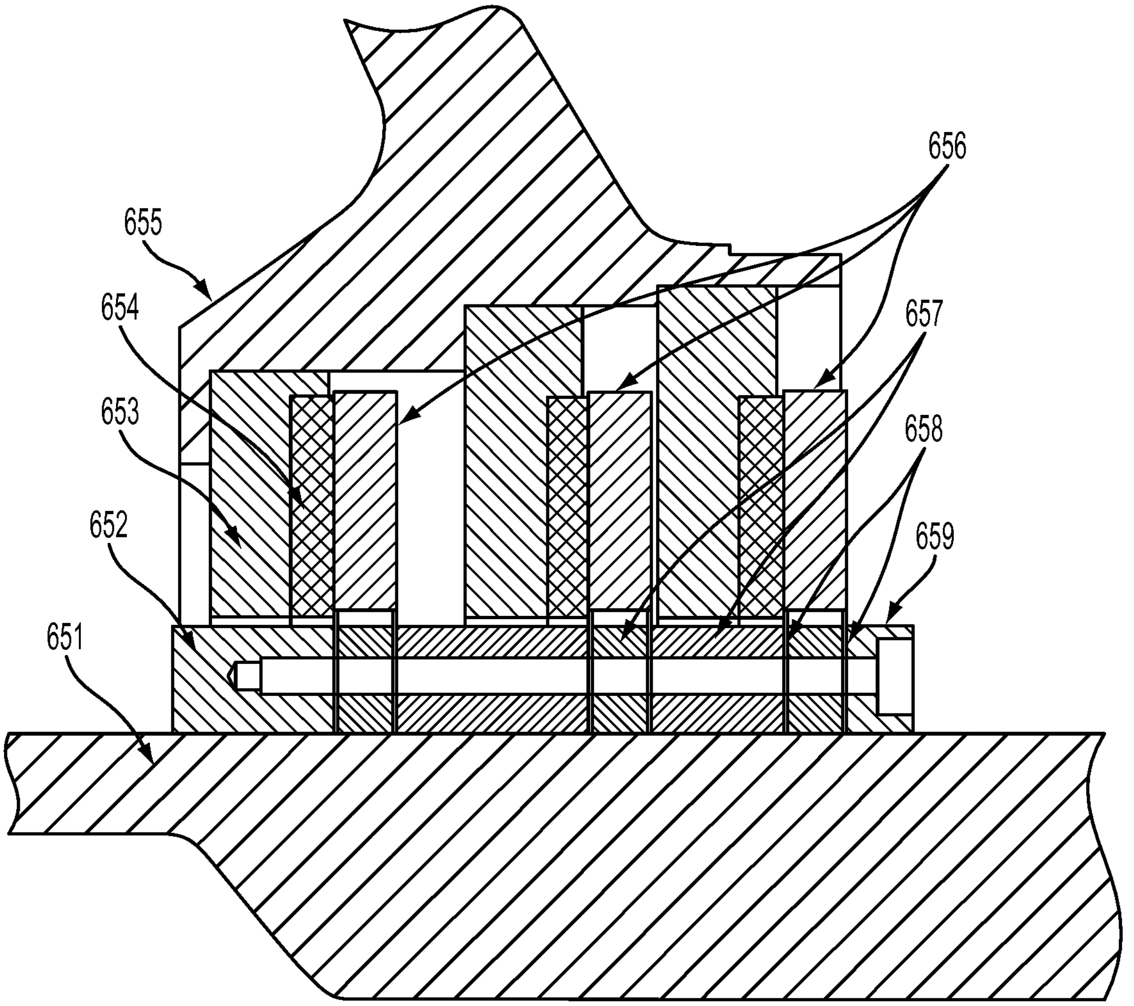

FIG. 6B

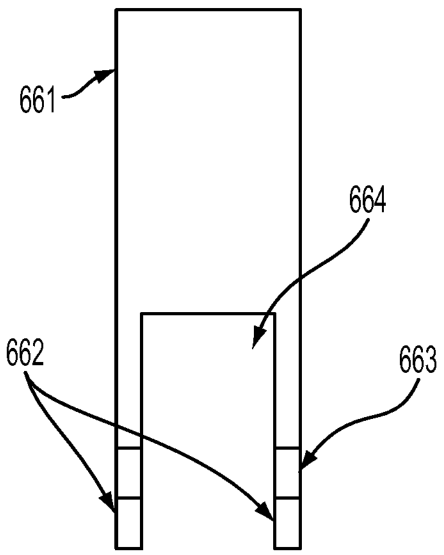

FIG. 6C

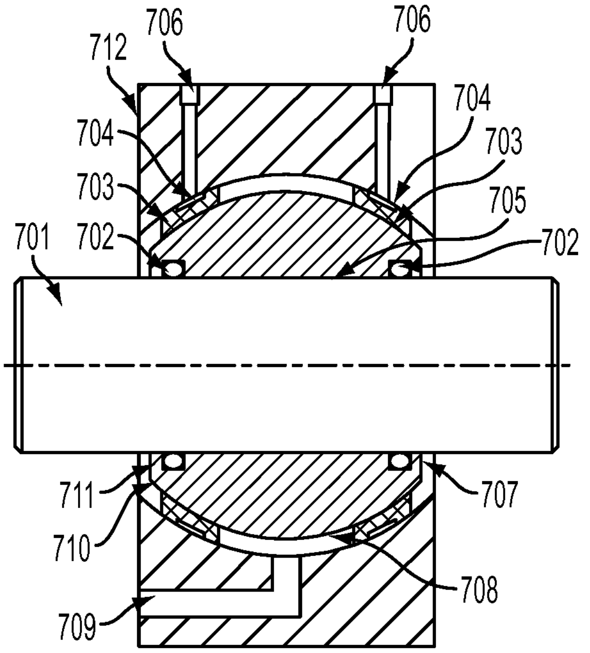

FIG. 7A

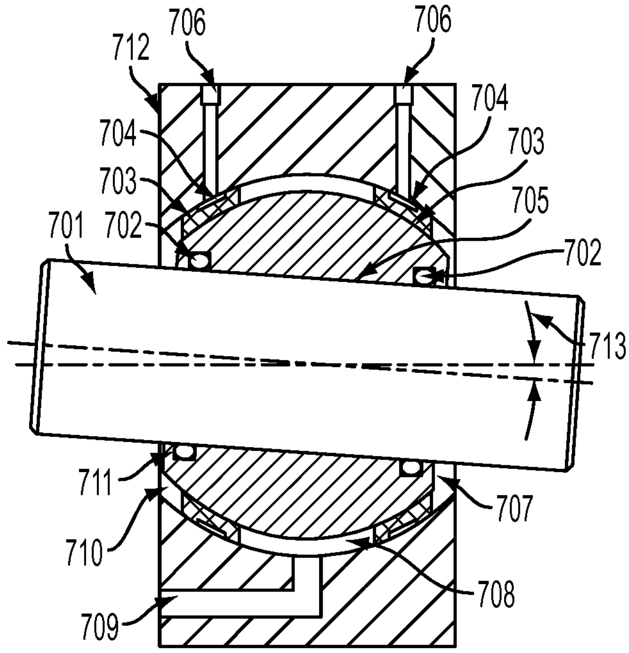

FIG. 7B

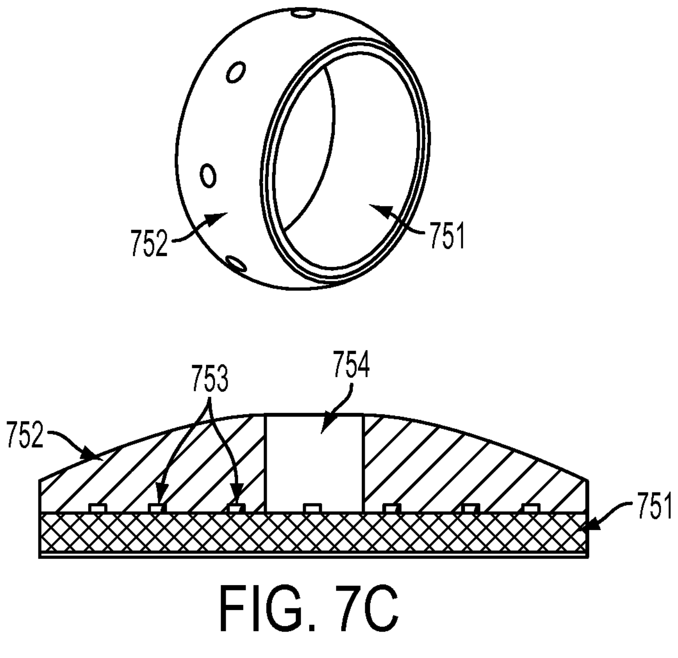

FIG. 7C

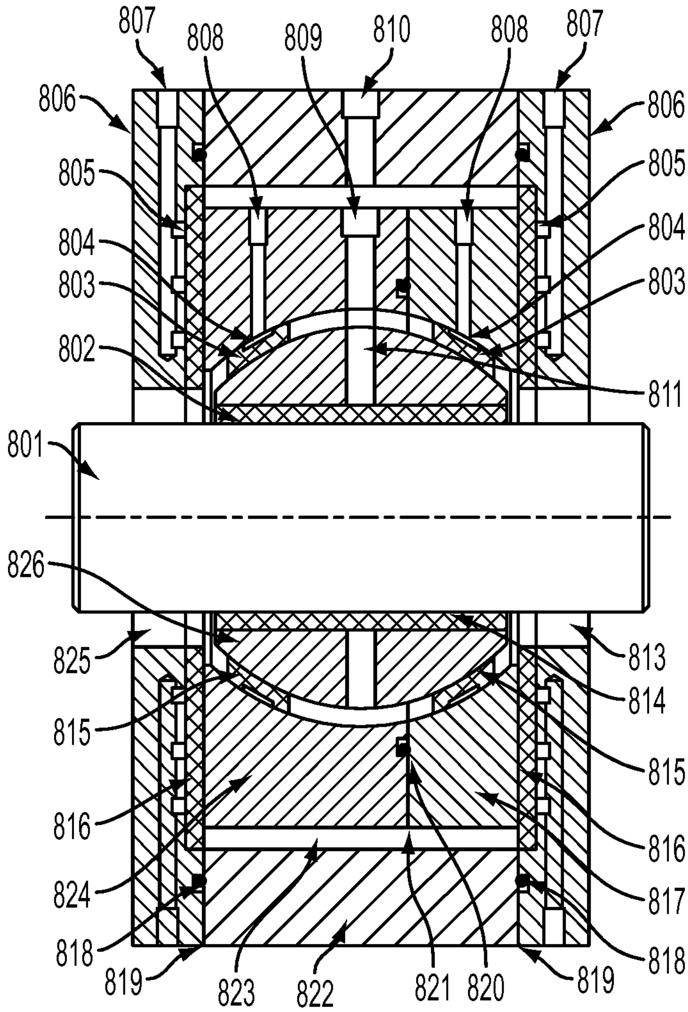
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FIG. 8A

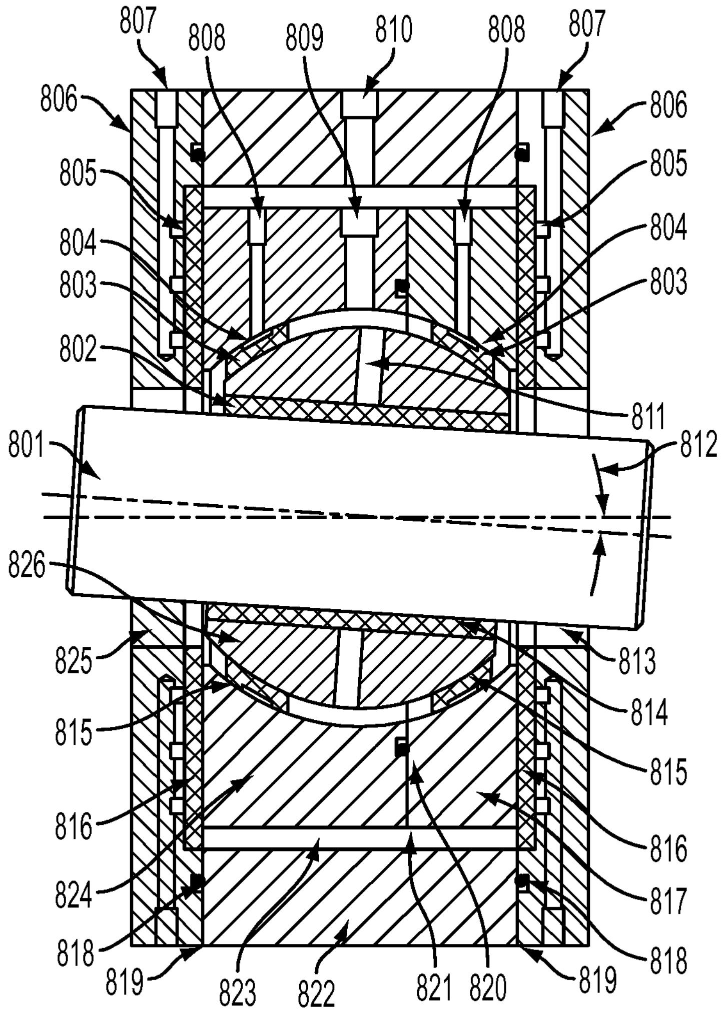
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FIG. 8B

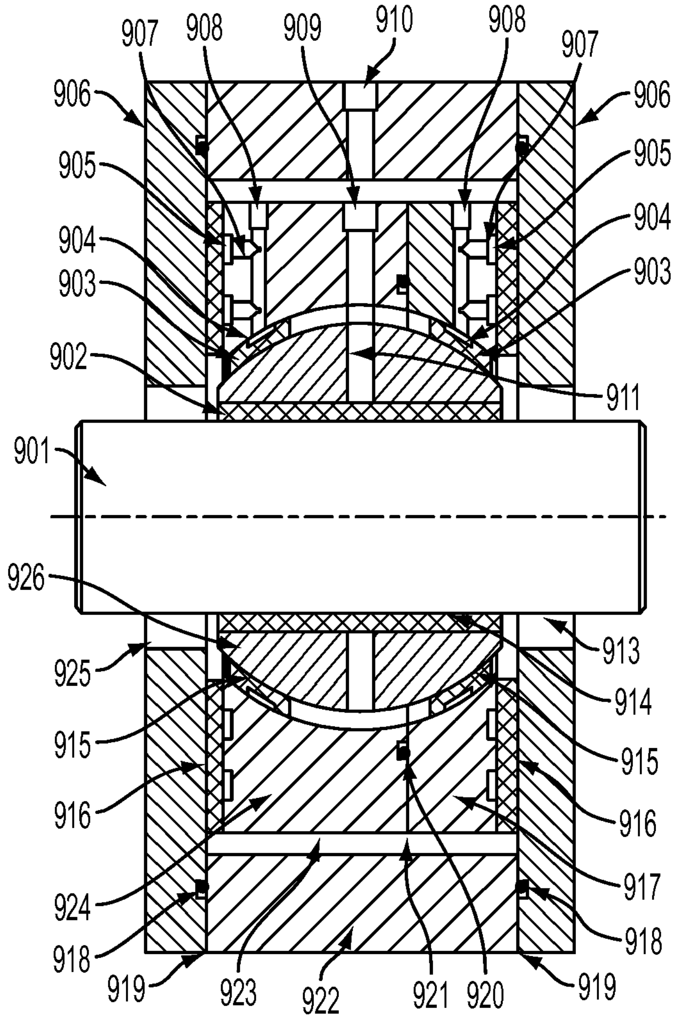
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FIG. 9A

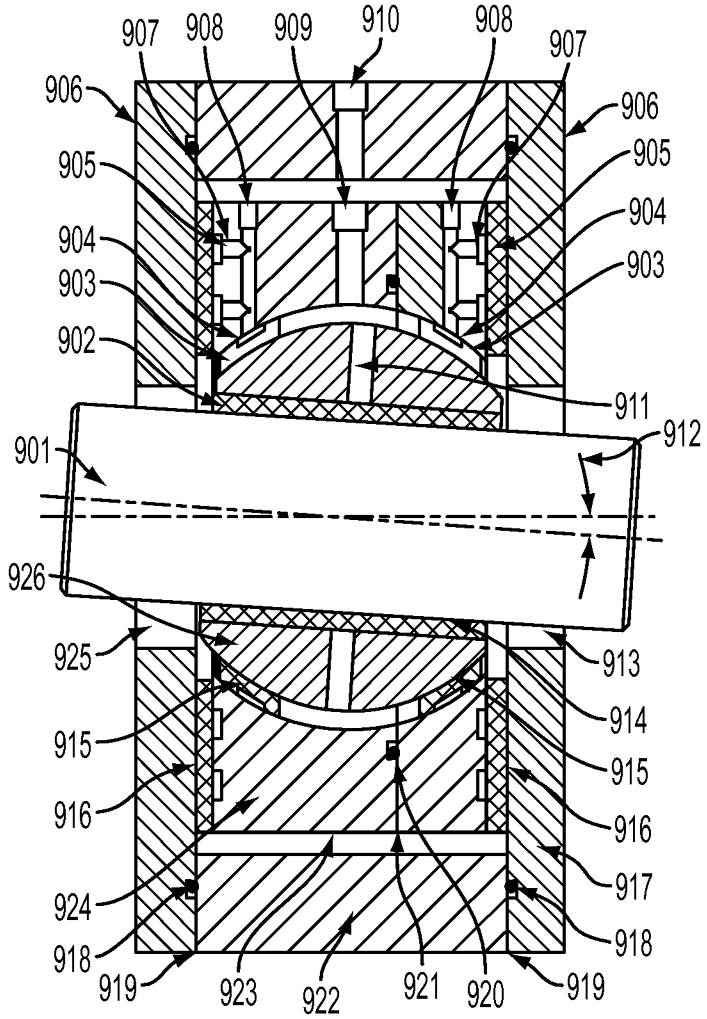
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FIG. 9B

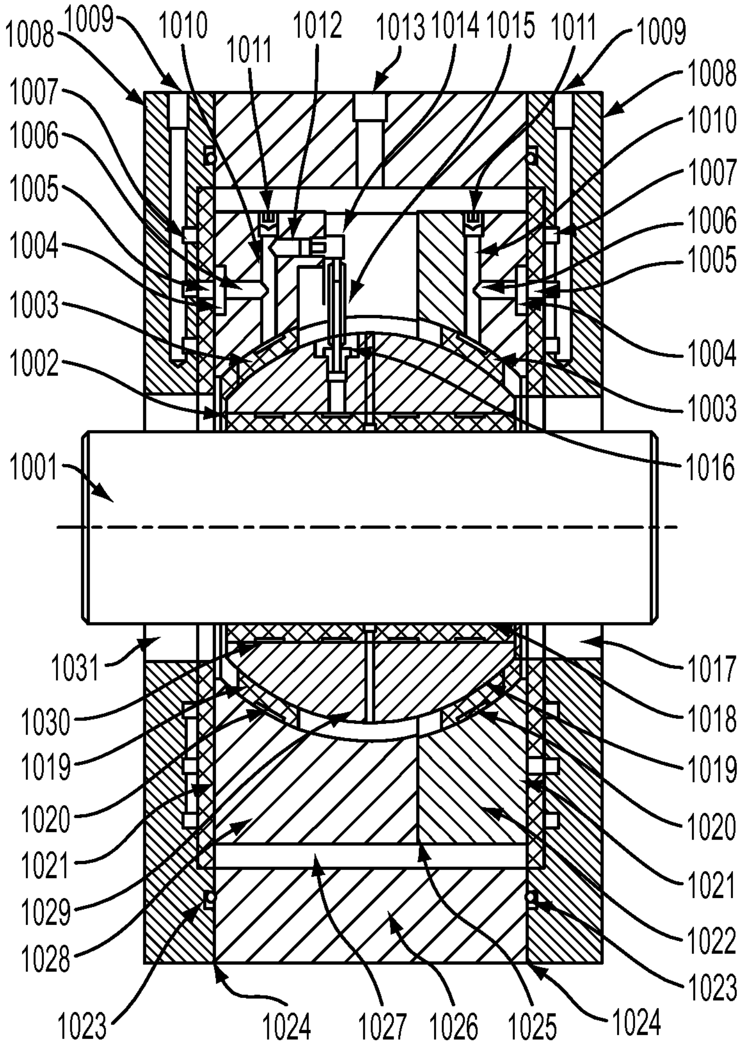
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FIG. 10A

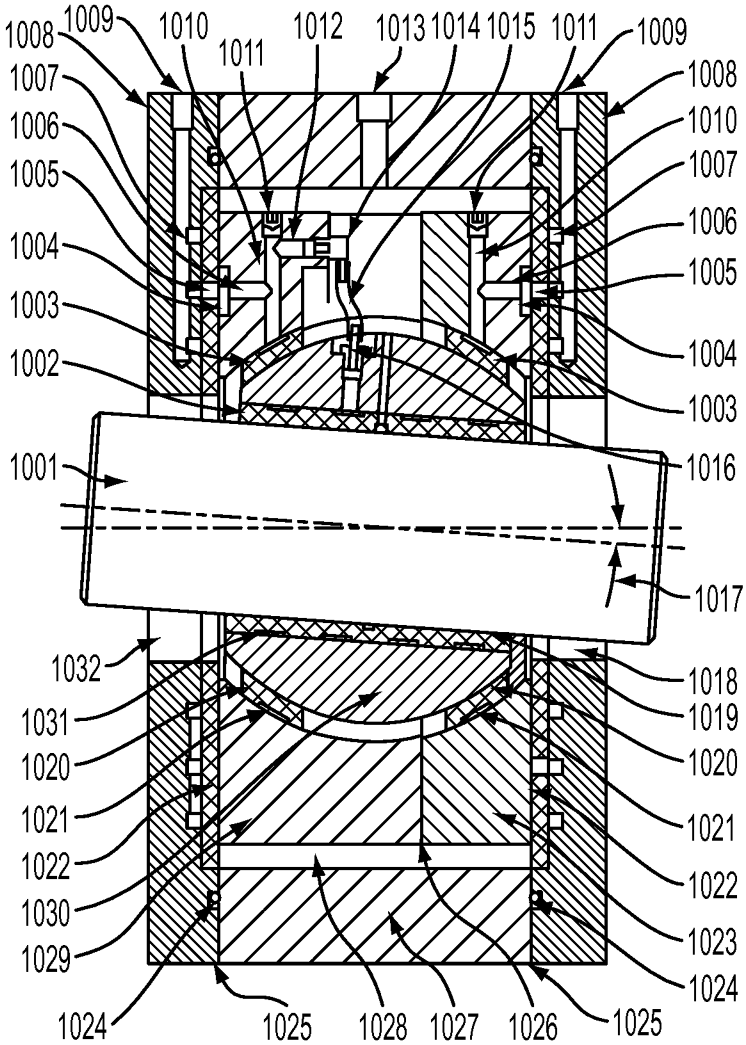
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FIG. 10B

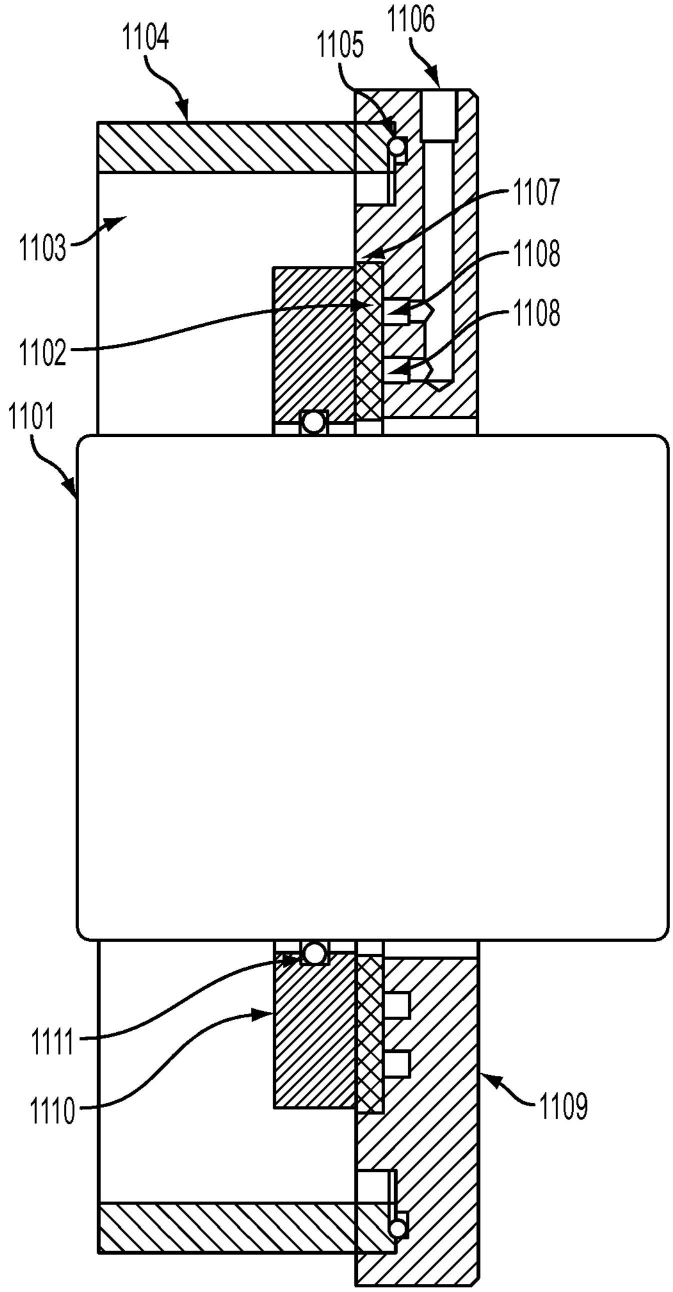

FIG. 11A

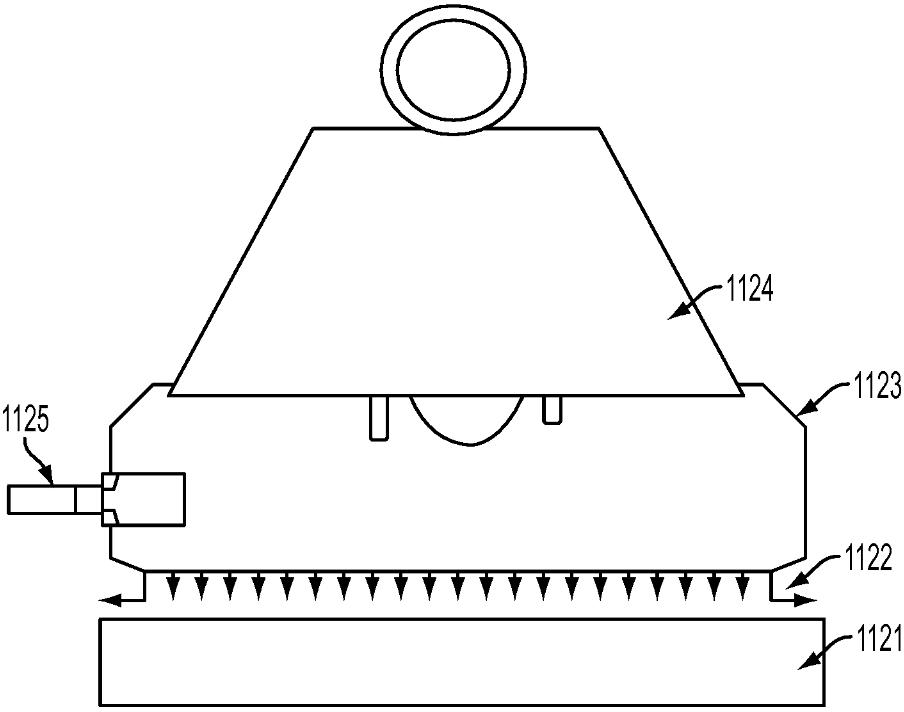

FIG. 11B

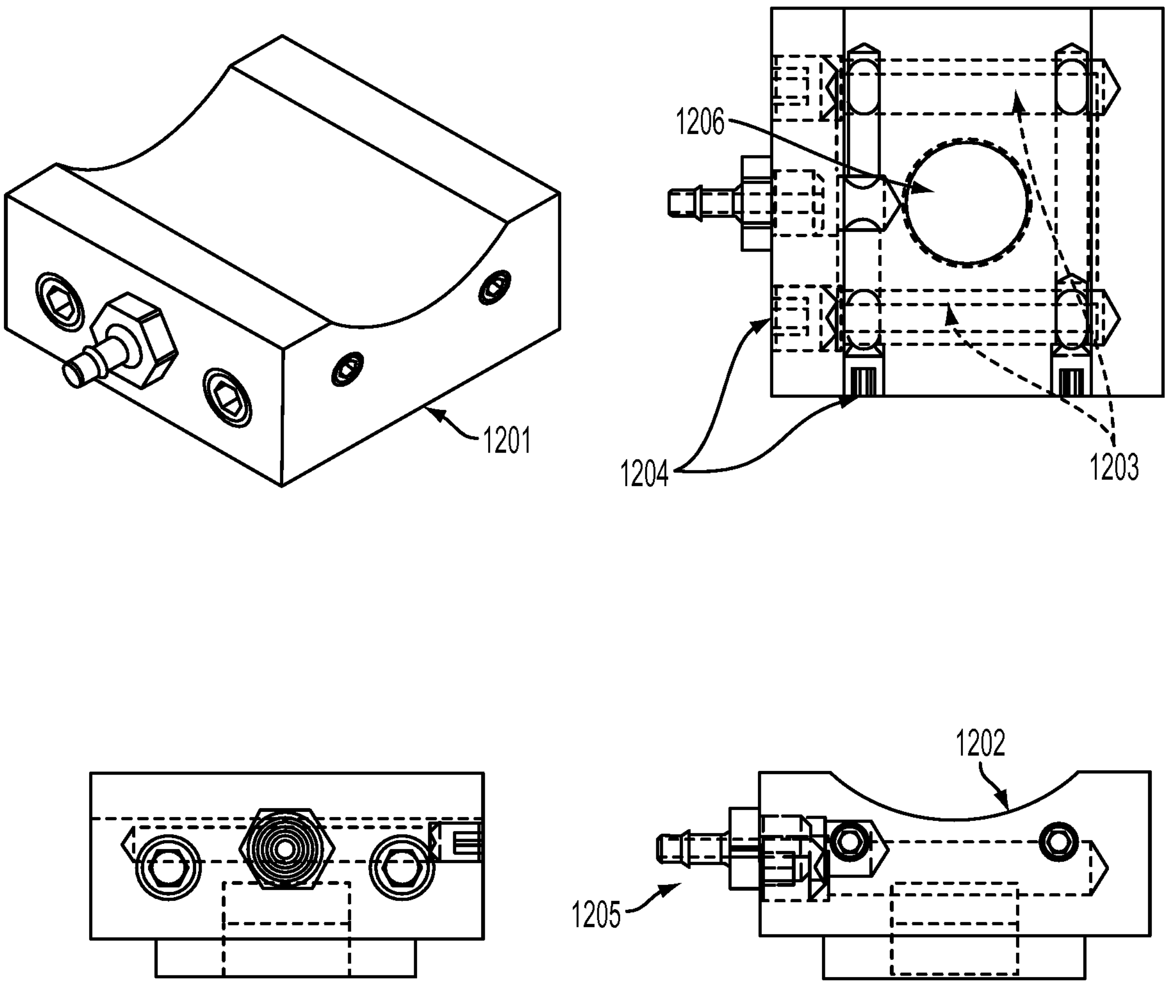

FIG. 12