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ABSTRACT

A piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, said piston assembly having a piston rod, a piston head, attached to said piston rod, forming an annular clearance space, having connected base and dome sections and a radially outer-most surface, said dome section having an interior cavity, a first sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said first sealing ring and said base section of said piston head forming a first ring passageway allowing working fluid to be exchanged between said first actuating cavity and said cylinder bore, a second sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said second sealing ring and said dome section of said piston head forming a second ring passageway allowing working fluid to be exchanged between said second actuating cavity and said cylinder bore, said piston assembly characterized by a sealing washer located in said annular clearance space, having first and second opposed flat surfaces and radially inner-most and outer-most surfaces, said first sealing ring located between said sealing washer and said base section of said piston head and positioned in direct engagement with said first opposed flat surface of said sealing washer, said radially inner-most surface of said first sealing ring separated from said sealing washer and said base section of said piston head by a first actuating cavity, and said second sealing ring located between said sealing washer and said dome section of said piston head and positioned in direct engagement with said second opposed flat surface of said sealing washer, said radially inner-most surface of said second sealing ring separated from said sealing washer and said dome section of said piston head by a second actuating cavity.

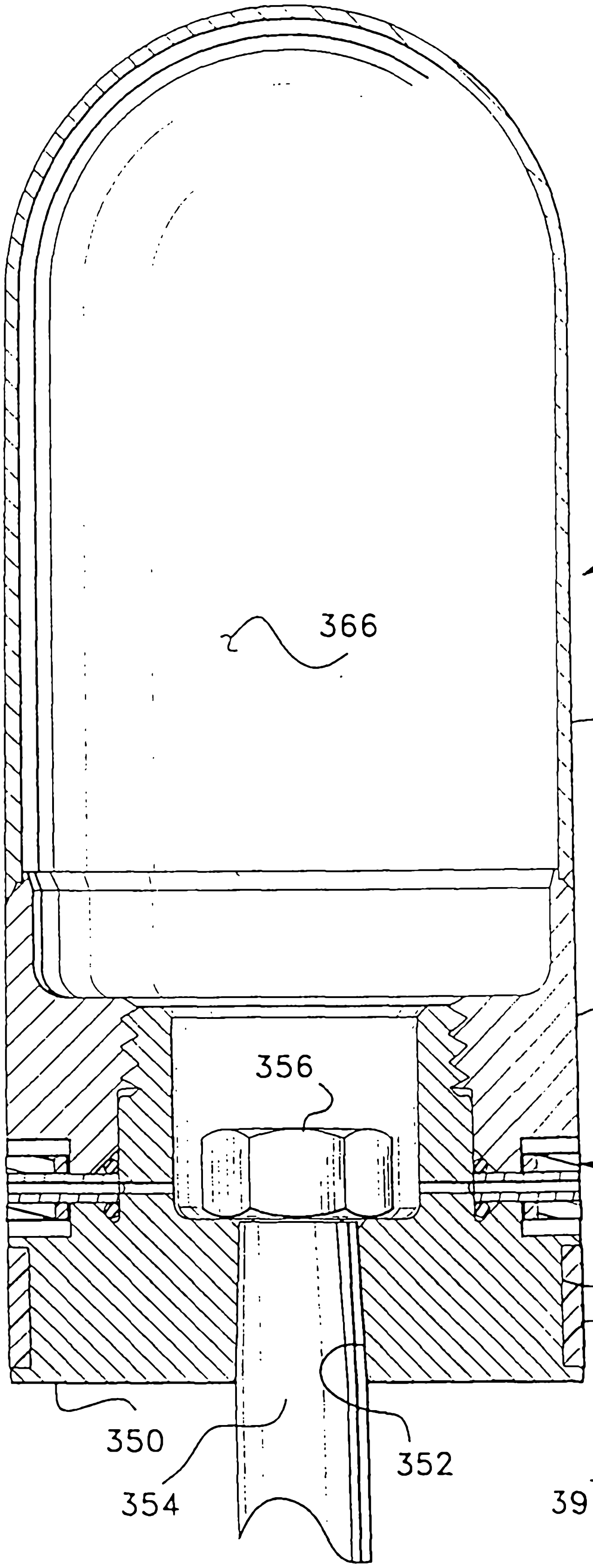
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Fig-11

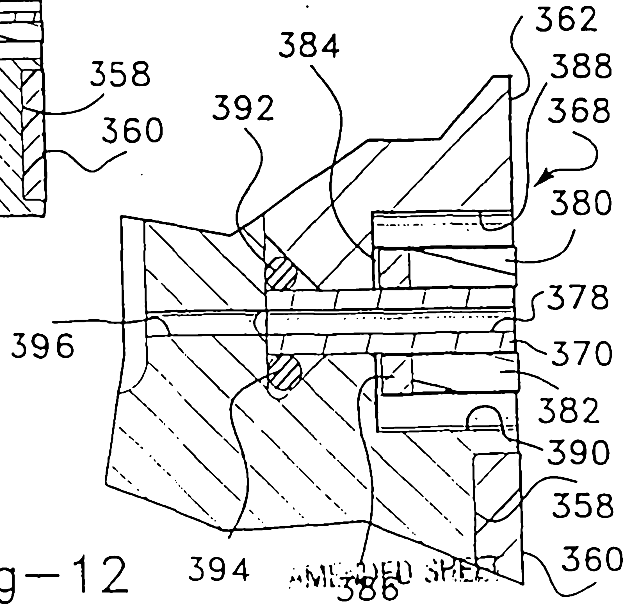
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Fig-12

Regulation 3.2

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COMPLETE SPECIFICATION

FOR A DIVISIONAL PATENT

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Invention Title:	**"Piston Assembly"**

The following statement is a full description of this invention, including the best method of performing it known to us:

PISTON ASSEMBLY

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention is related to a piston assembly heat engine. In particular the invention relates to an improved piston assembly for a Stirling cycle engine.

The basic concept of a Stirling engine dates back to a patent registered by Robert Stirling in 1817. Since that time, this engine has been the subject of intense scrutiny and evaluation. Various Stirling engine systems have been prototyped and put into limited operation throughout the world. One potential application area for Stirling engines is for automobiles as a prime mover or engine power unit for hybrid electric applications. Such applications place extreme demands on Stirling engine design. Due to the wide acceptance of spark ignition and Diesel engines, to gain acceptance, a Stirling engine must show significant advantages over those types, such as a dramatic enhancement in fuel efficiency or other advantages. In addition, reliability and the ability to manufacture such an engine at a low cost are of paramount importance in automotive applications. Similar demands are present in other fields of potential use of a Stirling engine such as stationary auxiliary power units, marine applications, solar energy conversion, etc.

Stirling engines have a reversible thermodynamic cycle and therefore can be used as a means of delivering mechanical output energy from a source of heat, or acting as a heat pump through the application of mechanical input energy. Using various heat sources such as combusted fossil fuels or concentrated solar energy, mechanical energy can be delivered by the engine. This energy can be used to generate electricity or be directly mechanically coupled to a load. In the case of a motor vehicle application, a Stirling engine could be used to directly drive traction wheels of the vehicle through a mechanical transmission. Another application in the automotive environmental is for use with a so-called "hybrid" vehicle in which the engine drives an alternator for generating electricity which charges storage batteries. The batteries drive the vehicle through electric motors coupled to the traction wheels. Perhaps other technologies for energy storage could be coupled to a Stirling engine in a hybrid vehicle such as flywheel or thermal storage systems, etc.

Significant advances have been made in the technology of Stirling machines, through a number of years. Examples of such innovations include development of a compact and efficient basic Stirling machine configuration employing a parallel cluster of double acting cylinders which are coupled mechanically through a rotating swashplate. In many applications, a swashplate actuator is implemented to enable the swashplate angle and therefore the piston stroke to be changed in accordance with operating requirements.

Although significant advances have been achieved in Stirling machine design, there is a constant need to further refine the machine, particularly if the intended application is in large volume production. For such applications, for example motor vehicles, great demands are placed on reliability and cost. It is well known that motor vehicle manufacturers around the world have made great strides in improving the reliability of their products. The importance of a vehicle engine continuing to operate reliably cannot be overstated. If a Stirling engine is to be seriously considered for motor vehicle applications, it must be cost competitive with other power plant technologies. This is a significant consideration given the mature technology of the spark ignition and Diesel internal combustion engines now predominately found in motor vehicles today.

In the past several decades significant improvements in exhaust pollution and fuel economy have been made for spark ignition and Diesel engines. However, there are fundamental limits to the improvements achievable for these types of internal combustion engines. Due to the high temperature intermittent combustion process which takes place in internal combustion engines, pollutants are a significant problem. Particularly significant are NO_x and CO emissions. Although catalytic converters, engine control, and exhaust treatment technologies significantly improve the quality of emissions, there remains room for improvement. Fuel efficiency is another area of concern for the future of motor vehicles which will require that alternative technologies be studied seriously. It is expected that the ultimate thermal efficiency achievable with the spark ignition internal combustion engines is on the order of 20%, with Diesel engines marginally exceeding this value. However, in the case of Stirling engines, particularly if advanced ceramic or other high temperature materials are implemented, thermal efficiencies in the neighborhood of 40% to 50% appear achievable. The external combustion process which could be implemented in an automotive Stirling engine would provide a steady state combustion process which allows precise control and clean combustion. Such a combustion system allows undesirable pollutants to be reduced.

In view of the foregoing, there is a need to provide design features for heat engines, such as a Stirling cycle engine, enabling it to be a viable candidate for incorporation into large scale mass production such as for automobiles and for other applications. The present invention relates to a piston assembly for a heat engine, such as a Stirling engine, which are directed toward these objects and goals.

Stirling engines are described in U.S. patent nos. 4,481,771; 4,532,855; 4,615,261; 4,579,046; 4,669,736; 4,836,094; 4,885,980; 4,707,990; 4,439,169; 4,994,004; 4,977,742; 4,074,114 and 4,966,841, which are hereby incorporated by reference. Basic features of many of the Stirling machines described in the above referenced patents are also implemented in connection with the present invention.

U.S. patent no. 4,885,980, which is equivalent to published European patent application EP-A-332272, discloses a Stirling engine having a swashplate that is journalled for rotation within a housing. The swashplate is driven by crossheads connected to reciprocating pistons and the swashplate in turn rotates a driveshaft through which mechanical power is delivered to a load coupled to the engine.

According to one aspect of the invention, there is provided a piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, said piston assembly having:

a piston rod,

a piston head, attached to said piston rod, forming an annular clearance space, having connected base and dome sections and a radially outer-most surface, said dome section having an interior cavity,

a first sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said first sealing ring and said base section of said piston head forming a first ring passageway allowing working fluid to be exchanged between said first actuating cavity and said cylinder bore,

a second sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said second sealing ring and said dome section of said piston head forming a second ring passageway allowing working fluid to be exchanged between said second actuating cavity and said cylinder bore,

said piston assembly characterized by:

a sealing washer located in said annular clearance space, having first and second opposed flat surfaces and radially inner-most and outer-most surfaces,

said first sealing ring located between said sealing washer and said base section of said piston head and positioned in direct engagement with said first opposed flat surface of said sealing washer, said radially inner-most surface of said first sealing ring separated from said sealing washer and said base section of said piston head by a first actuating cavity, and

said second sealing ring located between said sealing washer and said dome section of said piston head and positioned in direct engagement with said second opposed flat surface of said sealing washer, said radially inner-most surface of said second sealing ring separated from said sealing washer and said dome section of said piston head by a second actuating cavity.

There is provided according to another aspect of the invention a piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, said piston assembly having:

a piston rod,

a piston head, attached to said piston rod, forming an annular clearance space and having a radially outer-most surface,

first and second closed sealing rings,

said piston assembly characterised by:

a sealing washer, positioned within said annular clearance space, having first and second opposed flat faces and radially inner-most and outer-most surfaces,

said first sealing ring positioned between said sealing washer and said piston head, said first sealing ring, said sealing washer and said piston head forming a first actuating cavity, said first sealing ring positioned in a close mating relationship with said first opposed flat face of said sealing washer, said first sealing ring and said piston head forming a first ring passageway allowing working fluid to be exchanged between said first actuating cavity and said cylinder bore,

said second sealing ring positioned between said sealing washer and said piston head, said second sealing ring, said sealing washer and said piston head forming a second actuating cavity, said second sealing ring positioned in a close mating relationship with said second opposed flat face of said sealing washer, said second sealing ring and said piston head forming a second ring passageway allowing working fluid to be exchanged between said second actuating cavity and said cylinder bore.

The pistons include a sealing approach which implements easily machined elements which provide piston sealing. As the pair of sealing rings are used, they are subjected to fluid forces such that only one of the sealing rings is effective in a particular direction of reciprocation of the piston. This approach reduces friction, provides long ring life and enhances sealing performance.

Additional benefits and advantages of the present invention will become apparent to those skilled in the art to which this invention relates from the subsequent description of the preferred embodiments and the appended claims, taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross-sectional view through a Stirling engine incorporating a piston assembly in accordance with the invention;

FIG. 1A is a longitudinal cross-sectional view of the heater assembly of the engine of FIG. 1;

FIG. 1B is a partial cross-sectional view of a bellows rod seal incorporated into a modified form showing the bellows in an extended condition.

FIG. 1C is a view similar to FIG. 1B but showing the bellows compressed;

FIG. 2 is an end view of the drive case assembly taken from the output shaft end of the drive case, particularly showing the cross head components;

FIG. 3 is an enlarged cross-sectional view taken from FIG. 1 showing in greater detail the cross head assembly of the engine;

FIG. 4 is a partial cross-sectional view showing an electric swashplate actuator in accordance with a first embodiment;

FIG. 5 is a longitudinal cross-sectional view through the Stirling engine of FIG. 1. showing an alternate embodiment of a electric swashplate actuator;

FIG. 6 is a top view of the cross head body showing the guide rods in section;

FIG. 7 is a view partially in elevation and partially in section of the cross head body shown in FIG. 6;

FIG. 8 is a top view of the cross head adjuster sleeve;

FIG. 9 is a cross-sectional view taken along line 9-9 of FIG. 8;

FIG. 10 is an end view of the cylinder block component taken from the end of the drive case assembly;

FIG. 11 is a longitudinal cross-sectional view through the piston assembly;

FIG. 12 is an enlarged partial cross-sectional view particularly showing the piston ring assembly of this invention;

FIG. 13 is a top view of the cooler assembly;

FIG. 14 is a side view partially in section of the cooler assembly;

FIG. 15 is a plan view of the retainer plate;

FIG. 16 is a plan view of a cylinder extension locking C-ring;

FIG. 17 is a cross sectional view taken along line 17-17 from FIG. 16;

FIG. 18 is a plan view of a manifold segment of the heater head assembly;

FIG. 19 is a cross-sectional view taken along line 19-19 of FIG. 18;

FIG. 20 is a longitudinal cross-sectional view of a heater tube from the heater head assembly;

FIG. 21 is an enlarged partial cross-sectional view showing particularly the fin configuration of the heater tube;

FIG. 22 is a plan view of one of the fins of the heater tube shown in FIG. 20;

FIG. 23 is a plan view of an alternate configuration of a fin shape for the heater tube;

FIG. 24 is a cross-sectional view through the unloader valve;

FIG. 25 is a top view of the air preheater;

FIG. 26 shows a sheet of metal material from which the air preheater is formed;

FIG. 27 is a side view of the air preheater shown in FIG. 25;

FIG. 28 is an enlarged side view particularly showing the alternately welded configuration of the adjacent leaves of the preheater.

DETAILED DESCRIPTION OF THE INVENTION

A Stirling engine incorporating a piston assembly in accordance with this invention is shown in a completely assembled condition in FIG. 1 and is generally designated by reference number 10. Stirling engine 10 includes a number of primary components and assemblies including drive case assembly 12, cylinder block assembly 14, and heater assembly 16.

OVERALL CONSTRUCTION

Drive case assembly 12 includes a housing 18 having a pair of flat opposed mating surfaces 20 and 22 at opposite ends. Mating surface 20 is adapted to receive drive case output shaft housing 28 which is bolted to the drive case housing 18 using threaded fasteners 29. Mating surface 22 is adapted to be mounted to cylinder block assembly 14. Drive case housing 18 has a hollow interior and includes a journal 24 for mounting a drive shaft bearing. Arranged around the interior perimeter of drive case housing 18 is a series of cross head guide rods 26. A pair of adjacent guide rods 26 is provided for each of the four cross heads of the engine (which are described below). As will be evident from a further description of Stirling engine 10, the adjacent guide rods 26 should be parallel within extremely close tolerances.

One end of each guide rod 26 is mounted within bores 30 of drive case housing 18. The opposite ends of guide rods 26 are received in bores 32 of output shaft housing 28. The mounting arrangement for guide rods 26 is shown in FIGS. 1 and 3. One end of each guide rod 26 has a conical configuration bore 36 which terminates at a blind threaded bore. In addition, a series of slits are placed diametrically through the end of guide rods 26 at bore 36 so that guide rod end has limited hoop strength. Cone 34 is inserted within conical bore 36. A threaded fastener such as cap screw 38 is threaded into the threaded bore at the end of guide rod 26. By torquing threaded fastener 38, cone 34 is driven into bore 36 causing the end of guide rod 26 to expand into mechanical engagement with bore 32. This is achieved without altering the concentricity between the longitudinal axis of guide rod 26 and guide rod bores 30 and 32. Cap 40 seals and protects bore 32 and retains lubricating oil within the drive case.

Centrally located within output shaft housing 28 is journal 44 which provides an area for receiving spherical rolling bearing assembly 46 which is used for mounting drive shaft 50. At the opposite end of drive shaft 50 there is provided spherical roller bearing assembly 52 mounted in journal 24. Spherical bearing configurations are provided for bearing assemblies 46 and 52 to accommodate a limited degree of bending deflection which drive shaft 50 experiences during operation. Drive case housing 18 also provides a central cavity within which oil pump 56 is located. Oil pump 56 could be of various types but a gerotor type would be preferred. Through drilled passageways, high pressure lubricating oil is forced into spray nozzle 58 which sprays a film of lubricant onto the piston rods 260 (described below). In addition, lubricant is forced through internal passages within drive shaft 50, as will be explained in greater detail later.

Drive case 18 further defines a series of four counter-bored rod seal bores 60. At a position which would correspond with the lower portion of drive case 18, a sump port 62 is provided. The lubrication system of engine 10 can be characterized as a dry sump type with oil collecting in the interior cavity of drive case 18 being directed to oil pump and returned via suction of oil pump 56, where it is then pumped to various locations and sprayed as mentioned previously.

Drive shaft 50 is best described with reference to FIG. 1. Drive shaft 50 incorporates a variable angle swashplate mechanism. Drive shaft 50 includes an annular swashplate carrier 66 which is oriented along a plane tipped with respect to the longitudinal axis of drive shaft 50. Swashplate 68 in turn includes an annular interior cavity 70 enabling it to be mounted onto swashplate carrier 66. Bearings enable swashplate 68 to be rotated with respect to drive shaft swashplate carrier 66. Swashplate disc 72 is generally circular and planer but is oriented at an angle inclined with respect to that of swashplate cavity 70. By rotating swashplate 68 with respect to drive shaft 50, the angle defined by the plane of disc 72 and the longitudinal axis of drive shaft 50 can be changed from a position where they are perpendicular, to other angular orientations. Thus, rotation of drive shaft 50 causes disc 72 to rotate about an inclined axis. This basic swashplate configuration is a well known design implemented in prior Stirling engine configurations. Drive shaft 50 includes splined end 74 enabling it to be coupled to a load, which as previously stated, may be of various types. Two embodiments of actuators for changing the swashplate angle in a desired manner will be described later.

SWASHPLATE ACTUATOR

A first embodiment of an electric swashplate actuator is best shown with reference to FIG. 1 and 4, and is generally designated by reference number 110. Actuator 110 uses a DC torque motor, a planetary gear system, and bevelled gears to accomplish control over swashplate angle. With this embodiment of electric swashplate actuator 110, it is necessary to communicate electrical signals to rotating components. To achieve this, two pairs of slip ring assemblies 112 are provided. Two pairs are provided for redundancy since it is only necessary for one pair to apply electrical power. Each slip ring assembly 112 includes a pair of spring biased brushes 114 mounted to a carrier 116 attached to output shaft housing 28. Electrical signals are transmitted into slip rings 118 directly attached to drive shaft 50. Electrical conductors are connected to slip rings 118 and run through bearing mount 120 which is keyed to drive shaft 50 such that relative rotation is not possible between these two parts. Bearing mount 120 is connected with motor stator 122 having a number of permanent magnets (not shown) mounted thereto. The motor rotor 124 is journalled onto drive shaft 50 using needle bearing elements 126 such that they can rotate relative to one another. Electrical signals are transmitted to rotor 124 and its windings 128 via a second set of brushes 130. Accordingly, through the application of DC electrical signals through slip ring assemblies 112, electrical signals are transmitted to rotor windings 128 and thus the rotor can rotate relative to drive shaft 50. By applying voltage in the proper polarity, rotor 124 can be rotated in either direction as desired.

Actuator rotor 124 includes an extension defining sun gear 132. Three planet gears 134 mesh with sun gear 132 and also with teeth formed by stator extension 122 defining a ring gear which is fixed such that it does not rotate relative to shaft 50. Thus, as rotor 124 rotates relative to shaft 50, planet gears 134 orbit. Planet gears 134 feature two sections, the first section 138 meshing with sun gear 132, and a second section 139 having a fewer number of teeth meshing with ring gear 140. Revolution of the planet gear 134 causes rotation of ring gear 140 relative to drive shaft 50. Ring gear 140 is directly coupled to a bevel gear 142 which engages a bevel gear surface 144 of swashplate 68. As explained previously, relative rotation of swashplate 68 relative to drive shaft 50 causes an effective change in swashplate angle.

In normal operation, electric actuator 110 is not energized, therefore, sun gear 132 is stationary relative to drive shaft 50. Ring gear 140 is driven by swashplate 68 and both rotate at the same speed. Planet gears 134 carry the torque from ring gear 140 to sun gear 132 and stator ring gear 136. These then carry the torque to bearing mount 120 which in turn carries the torque to shaft 50. Therefore, except when actuated, there is no movement of the gears of electric actuator 110 relative to one another.

Now with reference to FIG. 5, a second embodiment of an electric swashplate actuator is shown and is generally designated by reference number 160. The primary distinction of electric actuator 160 as compared with electric actuator 110 is the use of a stationary motor which avoids the requirement of slip rings for communicating power to motor windings. Electric actuator 160 includes a stationary mounted driving electric motor (not shown) which drives worm gear 164 meshing with worm wheel 166. Worm wheel 166 can rotate freely relative to drive shaft 50 through a pair of anti-friction bearings 168. Worm wheel 166 is coupled to carrier arm 170. Shaft 172 is mounted to carrier arm 170 and drives planet gear 174 having a larger diameter toothed segment 176 and a smaller diameter toothed segment 178 which can rotate relative to shaft 172. Larger diameter planet gear segment 176 meshes with fixed gear 182 which is keyed or otherwise fixed to drive shaft 50 for rotation therewith. The smaller diameter planet gear segment 178 meshes with idler gear 184 which rotate relative to the shaft on bearings 186. Idler gear 184 engages with another planet gear set having planetary gears 188 having a smaller diameter segment 192 and a larger diameter segment 193. Planet gear 188 rotates about shaft 194. Shaft 194 is grounded to drive case housing 18. Larger diameter planet gear segment 193 meshes with sun gear 198 which is fixed to collar 200 which rotates relative to shaft 50 on bearings 202. Collar 200 is connected to bevel gear 204 which meshes with swashplate bevel gear 144.

In normal operation the actuator driving motor is not turning. Accordingly, worm 164 and worm wheel 166 are both stationary relative to drive case 18. Sun gear 198 is driven by the swashplate and both rotate at the same speed. Sun gear 198 causes the driven planet gear 188 to rotate about its axis which is held stationery to the drive case 18. This in turn causes idler gear 184 to rotate relative to shaft 50. The speed of idler gear 184 relative to the shaft is dependant on the sizes of the gears used. Fixed gear 182 meshes with the planetary gear 174. Because fixed gear 182 and sun gear 198 are the same size, planet gear 174 does not revolve around the drive shaft axis. However, when worm 164 is rotated, a gear reduction acting through the two planetary gear sets causes bevel gear 204 to rotate relative to drive shaft 50, thus changing the swashplate angle.

CROSS HEAD ASSEMBLY

Details of cross head assembly 220 are best shown with references to FIGS. 2, 3 and 6 through 9. Cross head body 222 forms a caliper with a pair of legs 224 and 226 connected by center bridge 228. Each of legs 224 and 226 define a pair of guide bores 230. Preferably, journal bearings are installed within guide bores 230 such as porous bronze graphite coated bushings 232. Bushings 232 enable cross head body 222 to move smoothly along guide rods 26. Cross head leg 224 also forms stepped cross head slider cup bore 234. Leg 226 forms slider cup bore 236 which also has a conical section 238. Within bores 234 and 236 are positioned slider cups 240 and 242, respectively. Slider cups 240 and 242 form semi-spherical surfaces 244 and 246. Slider elements 248 and 250 also define spherical outside surfaces 252 and 254, respectively, which are nested into slider cup surfaces 244 and 246, respectively. Opposing flat surfaces 256 and 258 are formed by the slider elements and engage swashplate disc 72. As will be explained in more detail below, a hydro-dynamic oil film is developed between spherical flat surfaces 256 and 258 as they bear against disc 72 to reduce friction at that interface. In a similar manner, a hydro-dynamic oil film is developed between slider cup spherical surfaces 244 and 246, and slider spherical outside surfaces 252 and 254.

Piston rods 260 extend between associated pistons and slider cup 242. Piston rod 260 has a threaded end 262 which meshes with slider cup threaded bore 264. The end of piston rod 260 adjacent threaded end 262 forms a conical outside surface 266 which is tightly received by cross head bore conical section 238. Thus, the relative position between slider cup 242 and cross head leg 224 is fixed. However, slider cup 240 is provided with means for adjusting its axial position within cross head body bore 234 such that precise adjustment of the clearances of the hydro-dynamic films is achievable. Slider cup 240 includes an extended threaded stud 270. In the annular space surrounded threaded stud 270 are adjuster sleeve 272 and cone 274. As best shown in FIGS. 8 and 9, sleeves 272 define an inside conical surface 276. Two perpendicular slits are formed diametrically across sleeve 272, one from the upper surface and one from the bottom surface and render the sleeve compliant in response to hoop stresses. Adjustment of the clearances for the hydro-dynamic films is provided by changing the axial position of slider cup 240 in bore 234. Once the gaps are adjusted properly, nut 278 is threaded onto stud 270 which forces cone 274 into engagement with sleeve conical surface 276, causing the sleeve to radially expand. This action forces the sleeve into tight engagement with cross head bore 234 thus fixing the position of cup 240.

ROD SEALS

As shown in FIG. 1, piston rod seal assembly 290 includes housing 292 mounted within rod seal bore 60. Rod seal assembly 290 further includes spring seal actuator 294 which urges an actuating collar 296 against sealing bushing 298. Seal actuator spring 294 is maintained within housing 292 through installation of an internal C-clip 300. Due to the conical surfaces formed on collar 296 and bushing 298, seal actuator spring 294 is able to cause the bushing to exert a radially inward squeezing force against piston rod 260, thus providing a fluid seal. Preferably, collar 296 is made of an elastomeric material such as a graphite filled Teflon™ material.

An alternate embodiment of a rod seal assembly is shown in FIGS. 1B and 1C. Bellows seal assembly 570 provides a hermetic rod seal. Bellows element 572 has its stationary end mounted to base 574, whereas the opposite end is mounted to ring 576. Bellows seal assembly 570 is carried by block 578 clamped between cylinder block assembly 14 and drive case assembly 12. FIG. 1B shows the bellows seal element in an extended position whereas FIG. 1C shows the element compressed. The design of engine 10 readily allows the sliding contact rod seal 290 to be replaced by bellows seal assembly 570 without substantial reworking of the engine design.

LUBRICATION SYSTEM

Oil lubrication of machine 10 takes place exclusively within drive case assembly 12. As mentioned previously, sump port 62 provides a collection point for lubrication oil within drive case housing 18. Through a sump pick-up (not shown), oil from sump port 62 enters oil pump 56 where it is forced at an outlet port through a number of lubrication pathways. Some of this oil sprays from nozzle 58 onto piston rods 260 and cross head guide rods 26. Another path for oil is through a center passage 310 within drive shaft 50. Through a series of radial passageways 312 in drive shaft 50, oil is distributed to the various bearings which support the drive shaft. Oil is also ported to swashplate 68 surfaces. The oil then splashed onto the sliding elements of the cross head assembly including slider cups 240 and 242, and slider elements 248 and 250. The exposed surfaces of these parts during their operation are coated with oil and thus generate a hydrodynamic oil film.

CYLINDER BLOCK

Cylinder block assembly 14, best shown in FIGS. 1 and 10, includes a cylinder block casting 320 having a pair of opposed parallel flat mating surfaces 322 and 324. Mating surface 322 enables cylinder block casting 320 to be mounted to drive case housing mating surface 22. Bolts 326 hold these two parts together. Stirling engine 10 is a four cylinder engine. Accordingly, cylinder block casting 320 defines four cylinder bores 328 which are mutually parallel. As shown in FIG. 1, cylinder bores 328 define a larger diameter segment through which piston assembly 330 reciprocates, as well as a reduced diameter clearance bore section for rod seal assembly 290. Four cooler bores 332 are also formed in cylinder block casting 320 and are mutually parallel as well as parallel to cylinder bores 328. Cylinder bores 328 are arranged in a square cluster near the longitudinal center of cylinder block casting 320. Cooler bores 332 are also arranged in a square cluster but lie on a circle outside that of cylinder bores 328, and are aligned with the cylinder bores such that radials through the center of cooler bores 332 pass between adjacent cylinder bores. In that Stirling engine 10 is a double acting type, cylinder block casting 320 including working gas passageways 334 which connect the bottom end of cooler bore 332 to the bottom end of an adjacent cylinder bore 328 as shown in FIG. 10. Cylinder block casting 320 further forms coolant passageways 336 which provide for a flow of liquid coolant through coolant bores 332 in a diametric transverse direction.

PISTON ASSEMBLY

Piston assembly 330 is best shown with reference to FIGS. 11 and 12. Piston base 350 forms a conical bore 352 which receives a conical end 354 of piston rod 260. Nut 356 combined with friction at the conical surfaces maintains the piston rod fixed to piston base 350. An outer perimeter groove 358 of the piston base receives bearing ring 360 which serves to provide a low friction surface engagement with the inside of cylinder bore 328. Bearing ring 360 is preferably made of an low friction elastomeric material such as "Rulon™" material. Dome base 362 is fastened onto piston base 350 through threaded engagement. Dome 364 is welded or otherwise attached to dome base 362. Dome 364 and dome base 362 define a hollow interior cavity 366 which is provided to thermally isolate opposing ends of piston assembly 330.

Located between piston base 350 and dome base 362 are a number of elements which comprise piston ring assembly 368 which provides a gas seal around the perimeter of piston assembly 330 as it reciprocates in its bore. Sealing washer 370 is clamped between piston base 350 and dome base 362 and is a flat with opposing parallel lapped surfaces. A number of radial passageways 378 are drilled through washer 370. On opposing sides of sealing washer 370 are provided sealing rings 380 and 382 preferably made of "Rulon™" type elastomeric low friction material. Sealing rings 380 and 382 contact cylinder bore 328 to provide gas sealing. Acting at the inside diameter of sealing rings 380 and 382 are spring rings 384 and 386 which are split to provide radial compliance. Spring rings 384 and 386 are provided to outwardly bias sealing rings 380 and 382, urging them into engagement with the cylinder bore.

At a number of circumferential locations, passageways 388 are drilled radially into dome base 362. In a similar manner, passageways 390 are formed within piston base 350. A pair of O-rings 392 and 394 are clamped against opposing face surfaces of sealing washer 370. At axial location aligned with sealing washer 370, piston base 350 defines one or more radial passageways 396 communicating with piston dome interior cavity 366 which functions as a gas accumulator.

As piston assembly 330 reciprocates within its bore the two sealing rings 380 and 382 provide a gas seal preventing cycle fluid from leaking across the piston assembly. Sealing rings 380 and 382 are pressure actuated such that only one of the two rings is providing a primary seal at any time. Specifically, sealing ring 380 provides a gas seal when the piston is moving downwardly (i.e. toward swash plate 68) whereas sealing ring 382 is pressure actuated when the piston is moved in an upward direction. Since Stirling engine 10 is of the double acting variety, piston assembly 330 is urged to move in both its reciprocating directions under the influence of a positive fluid pressure differential across the piston assembly. Thus, just after piston assembly 330 reaches its top dead center position, a positive pressure is urging the piston downwardly. This positive pressure acts on sealing ring 380 urging it into sealing contact with the upper surface of sealing washer 370. The lower sealing ring 382 however, is not fluid pressure actuated since it is urged away from sealing contact with sealing washer 370 since passageway 390 provides for equal pressure acting on the upper and lower sides of the ring. In the upward stroke of piston assembly 330 a positive pressure is urging the piston to move upwardly and thus sealing ring 382 seals and sealing ring 380 is not fluid pressure actuated as described previously. As this reciprocation occurs, piston cavity 366 is maintained at the minimum cycle pressure. This assures that the radial clearance space between sealing rings 380 and 382 is at a low pressure, thus providing a pressure differential for pressure actuating the seal rings into engagement with the inside diameter of the piston bores, thus providing a fluid seal.

COOLER ASSEMBLY

Cooler assembly 400 is best shown with reference to FIGS. 13 and 14 and is disposed within cylinder block cooler bores 332. Cooler assembly 400 compromises a "shell and tube" type heat exchanger. As shown, housing 402 includes pairs of perimeter grooves at its opposite ends which receive sealing rings 405 for sealing the assembly within cooler bore 332. Housing 402 also forms pairs of coolant apertures 408 within housing 402. A number of tubes 410 are arranged to extend between housing ends 412 and 414. Tubes 410 can be made of various materials and could be welded or brazed in place within bores in housing ends 410 and 414. As a means of reducing flow loses of the Stirling cycle working gas, the ends of the inside diameters of tubes 410 are counter bored or flared to form enlarged openings. The Stirling cycle working gas is shuttled back and forth between the ends 412 and 414 of the cooler housing and passes through the inside of tubes 410. A coolant, preferably a liquid is pumped in a cross flow manner through block coolant passages 336 and housing apertures 408 to remove heat from the working gas.

CYLINDER EXTENSIONS

Cylinder block assembly 14 further includes tubular cylinder tops or extensions 420 which form a continuation of the cylinder block bores 328. At their open ends, tubular cylinder extensions 420 form a skirt which allows them to be accurately aligned with cylinder bores 328 by piloting. O-ring seal 422 provides a fluid seal between cylinder block bores 328 and tubular cylinder extensions 420. Cylinder extensions 420 at their opposing end form a heater tube manifold 424 which will be described in more detail below.

REGENERATOR HOUSINGS

Cup shaped regenerator housings 430 are provided which are aligned co-axially with cooler bores 332. Regenerator housings 430 define an open end 432 and a closed top 434 having manifold 436 for communication with the heater assembly. Within regenerator housing 430 is disposed regenerator 444, which in accordance with known regenerator technology for Stirling engines, is comprised of a material having high gas flow permeably as well as high thermal conductivity and heat absorption characteristics. One type of regenerator uses wire gauze sheets which are stacked in a dense matrix.

RETAINER PLATE

Retainer plate 448 is best shown in FIG. 15 and provides a one-piece mounting structure for retaining tubular cylinder extensions 420 and regenerator housings 430 in position. Retainer plate 448 forms cylinder extension bores 450 and regenerator housing bores 452. Cylinder extension bores 450 have a diameter slightly larger than the largest diameter at the open end of tubular cylinder extension 420 and the bore is stepped as shown in FIG. 1. In a similar fashion, regenerator housing bores 452 are also enlarged with respect to the open end of regenerator housing 430 and are also stepped. Retainer plate 448 is designed so that the open ends of tubular cylinder extensions 420 and regenerator housings 430 can be inserted as an assembly through their associated plate bores. This is advantageous since the configuration of cylinder extension 420 and the heater assembly 16 attached to the cylinder extension and regenerator housing 430 would not permit top mounting. For assembly, retainer plate 448 is first positioned over cylinder extensions 420 and regenerator housings 430. Thereafter, semi-circular cylinder extension locking C-rings 454 shown in FIGS. 1, 16 and 17, and regenerator housings locking C-rings 456 are placed around the associated structure and allow retaining plate 448 to clamp these components against cylinder block mounting face 324, in a manner similar to that of an internal combustion engine valve stem retainer. Mounting bolts 458 fasten retainer plate 448 to cylinder block body 320. The use of a one-piece retaining plate provides rapid assembly and securely mounts the various components in an accurately aligned condition.

Cylinder extension 420 interact with cylinder block mating surface 324 to accurately pilot the center of the cylinder extensions with respect to cylinder block cylinder bores 328. However, the need for such accurate alignment does not exist for regenerator housings 430, and therefore, a face seal is provided allowing some degree of tolerance for misalignment between the regenerator housings and cooler bores 332. In this way, assembly is simplified by reducing the number of parts which must be simultaneously aligned.

HEATER ASSEMBLY

Heater assembly 16 provide a means of inputting thermal energy into the Stirling cycle working gas and is shown in FIG. 1A. A combustor (not shown) is used to burn a fossil fuel or other combustible material. Alternatively, heat can be input from another source such as concentrated solar energy, etc. In Stirling engine 10 combustion gases flow axially toward central heat dome 470 where it is deflected to flow in a radial direction. An array of heater tubes 478 is arranged to conduct heat from the hot gas as it flows radially out of the engine. Heater tubes 478 are arranged to form an inner band 480 and an outer band 482. The tubes of inner band 480 have one end which fits within cylinder extension manifold 424 and the opposite end fitting into heater tube manifold segment 484. As best shown in FIGS. 18 and 19, the tubes of inner bands 480 are arranged in a staggered relationship as are the tubes of outer band 482, thus enhancing heat transfer to the heater tubes. Manifold segment 484 has internally formed passageways such that the inner most tubes of inner band 480 are connected with the inner-most band of outer tubes 482 through passageways 486. In a similar manner the outer groups of inner and outer bands are connected via internal passageways 488. The tubes of the outer band 482 are connected with manifold segment 484 and the regenerator housing manifold 436.

Each of tubes 478 defining heater tube inner band 480 and outer band 482 are identical except the outer band tubes are longer. Tubes 478 are preferably made from a metal casting process which provides a number of benefits. The material which can be used for cast heater tubes can be selected to have higher temperature tolerance characteristics as compared with the deformable thin-walled tubes typically used. As shown in FIGS. 20 and 21, heater tubes 478 have projecting circular fins 492. The cross-section of the fins shown in FIG. 21 reveals that they can have a thickness which decreases along their length with rounded ends. Various other cross-sectional configurations for fins 492 can be provided to optimize heat transfer characteristics. In addition to optimizing the longitudinal cross-sectional shape of the fins, modifications of their perimeter shape can be provided. FIG. 22 shows a circular outside perimeter shape for fins 492. Using a casting process for forming heater tubes 478 other shapes to be provided. For example, FIG. 23 shows a generally dart shaped platform configuration. The configuration can be tailored to the gas flow dynamics which occur around the tubes. For example, it is known that for tubes arranged perpendicular to the gas flow direction, the upstream side surface 496 of the tubes tends to absorb more heat than the downstream or back side 498 of the tubes. For conventional tubes, this leads to significant thermal gradients which produce mechanical stresses on the heater tubes which can in turn lead to their failure over time. The platform provided in FIG. 23 may be advantageous to increase heat adsorption on the backside 498 to maintain more constant tube temperature for gas flowing in the direction of arrow 492 since more fin area is exposed on the downstream side where heat transfer is less efficient.

PRESSURE BALANCING

As in conventional Stirling cycle engines employing multiple double acting cylinders, in the case of the four cylinder engine shown four distinct isolated volumes of working gas such as hydrogen or helium are present in the engine. One of the volumes is defined by the expansion space above piston dome 364 which in turn flows through heater tubes 478, regenerator 444, cooler assembly 400, and cylinder block passageway 334 to the lower end of an adjacent cylinder bore 328. In a similar manner, three additional discrete volumes of gas are defined. Each of the gas volumes undergo shuttling between a compression space defined at the lower end of piston cylinder bore 328 in cylinder block casting 320, and an expansion space defined within tubular cylinder extension 420. Thus, the gases are shuttled between these spaces as occurs in all Stirling engines. Gases passing through heater assembly 16 absorb heat and expand in the expansion space and are cooled by cooler assembly 400 before passing into the compression space.

In order to minimize imbalances in the operation of engine 10, the mean pressure of the four distinct gas volumes needs to be equalized. This is achieved through the use of working fluid ports 500 positioned at the lower-most end of cylinder block cooler bore 332, best shown in FIG. 10, each of which are exposed to the separate gas volumes. Fitting 502 is installed in a port and from it are three separate tube elements. A first small capillary tube 504 communicates with pressure transducer block 506 having individual pressure transducers for each of the gas volumes, enabling those pressures to be measured. Capillary tube 508 communicates with manifold block

510 having an internal cavity which connects each of the individual capillary tubes 508 together. The function of manifold block 510 is to "leak" together the volumes for equalization of any mean pressure imbalances which may occur between them. A low restriction passageway connecting these cycle volumes together would unload the engine and would constitute an efficiency loss. Therefore, tubes 508 have a restricted inside diameter and thus the flow rate through these tubes is restricted. However, over time, pressure imbalances are permitted to equalize through fluid communication between the volumes.

UNLOADER VALVE

In the event of a mechanical failure or other condition which leads to a leakage of working gas from the engine, a severe imbalance condition can result. For example, if only one or more of the enclosed gas volumes leaks to atmosphere, potentially destructive loads would be placed on the mechanical components of engine 10. In order to preclude this from occurring, conduits 518 communicate with unloader valve 520 as shown with reference to FIG. 24. As shown, unloader valve includes housing 522 within internal stepped bore 524. A series of pipe fittings 526 are provided which communicate with individual diameter sections of stepped bore 524 via passageways 528. Each of fittings 526 communicates with the separate gas volumes via conduits 518. Spool 530 is positioned within stepped bore 524 and is maintained in the housing by cap 532. A series of grooves 534 are provided on the various diameter sections of spool 530 and retain O-rings 536. Spool 530 is urged in the right-hand direction as viewed in FIG. 24 by coil spring 538. An additional port is provided at fitting 540 which communicates with manifold block 510 via conduit 541 and is exposed to the engine mean pressure. This pressure signal passes through passageway 542 and acts on the full end area of spool 530. During normal engine operation, individual diameter sections of stepped bore 524 are exposed to the mean pressure of the four enclosed gas volumes. Each of these pressure signals produces a resultant net force on spool 530 urging it toward the right-hand direction which is assisted by the compliance of spring 538. In a normal operating condition, these pressures produce forces added to the spring compliance pushing shuttle spool 530 to the right-hand position as shown. However, in the event of the mechanical failure of engine 10 causing a leakage of working fluid, one (or more) of the passageways 528 experiences a loss in pressure. In this event, the net force acting to retains spool 530 in position is reduced and the equilibrium condition is unbalanced to move the shuttle in the left-hand direction under the influence of the engine mean cycle pressure through passageway 542. When this occurs, the various O-rings 536 unseat from their associated sealing surfaces and thus all of the gas volumes are vented together inside housing 522, rendering the engine incapable of producing mechanical output power and thus protecting the engine from destructive imbalance forces.

AIR PREHEATER

Combustion gases which pass through heater tube inner and outer banks 480 and 482 still are at an elevated temperature and have useful heat energy which can be recovered to enhance the thermal efficiency of engine 10. This is achieved through the use of air preheater 550 which has an annular ring configuration and surrounds heater tube outer bank 482. Air preheater 550 is formed from sheet metal stock having a high temperature capability. The stock first begins with a flat sheet 552 which may have local deformations as shown in FIG. 26 such as dimples 554, and is bent in an accordion-like fashion about fold lines 556. After sheet 552 is corrugated, its ends are welded to define the annular preheater configuration shown in FIGS. 25, 27, and 28. FIG. 28 shows that these corrugations are pinched together and welded at the axial ends of the preheater. Upper end 558 is formed with adjacent layers pinched together and welded as shown. Bottom end 560 has layers which are pinched together but alternate with those pinched together at top end 558. This arrangement provides the gas flow direction shown in FIG. 1A in which combustion gas flow is shown by cross-hatched arrows and fresh combustion air by clear arrows. Combustion gases passing through heater assembly 16 are deflected by baffle 562. The hot gases then enter the inside diameter of air preheater 550. Since the upper end 558 of these wraps are sealed, the gas is forced to flow downwardly as shown by the arrows. After passing through air preheater 550 these gases are vented or are further treated downstream. Fresh combustion air enters at the radially outer side of air preheater 550 and is constrained to flow in an axial direction through baffle 564. Combustion inlet air travels upwardly in an axial direction as shown by the upward directed arrows and is thereafter conveyed to a fuel combustor (not shown). Heat is transferred through the thin sheet metal forming air heater 550.

As a means of further enhancing thermal efficiency of engine 10, the inside surface of air preheater 550 exposed to combustion gases can be coated with a catalyst material such as platinum or palladium, or other catalyst materials. This thin layer 566 encourages further combustion of hydro-carbons within the combustion gases which has the two-fold benefits of reducing emissions as well as increasing the combustion gas temperature thereby increasing combustor inlet air temperature and efficiency.

While the above description constitutes the preferred embodiments of the present invention, it will be appreciated that the invention is susceptible of modification, variation and change without departing from the proper scope and fair meaning to the accompanying claims.

Throughout this specification and the claims which follow, unless the context requires otherwise, the word "comprise", and variations such as "comprises" and "comprising", will be understood to imply the inclusion of a stated integer or step or group of integers or steps but not the exclusion of any other integer or step or group of integers or steps.

The reference to any prior art in this specification is not, and should not be taken as, an acknowledgement or any form of suggestion that that prior art forms part of the common general knowledge in Australia.

The claims defining the invention are as follows:

1. A piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, said piston assembly having:

a piston rod,

a piston head, attached to said piston rod, forming an annular clearance space, having connected base and dome sections and a radially outer-most surface, said dome section having an interior cavity,

a first sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said first sealing ring and said base section of said piston head forming a first ring passageway allowing working fluid to be exchanged between said first actuating cavity and said cylinder bore,

a second sealing ring, formed of elastic low friction material, having radially inner-most and outer-most surfaces, located in said annular clearance space said second sealing ring and said dome section of said piston head forming a second ring passageway allowing working fluid to be exchanged between said second actuating cavity and said cylinder bore,

said piston assembly characterized by:

a sealing washer located in said annular clearance space, having first and second opposed flat surfaces and radially inner-most and outer-most surfaces,

said first sealing ring located between said sealing washer and said base section of said piston head and positioned in direct engagement with said first opposed flat surface of said sealing washer, said radially inner-most surface of said first sealing ring separated from said sealing washer and said base section of said piston head by a first actuating cavity, and

said second sealing ring located between said sealing washer and said dome section of said piston head and positioned in direct engagement with said second opposed flat surface of said sealing washer, said radially inner-most surface of said second sealing ring separated from said sealing washer and said dome section of said piston head by a second actuating cavity.

2. A piston assembly according to claim 1 wherein said first ring passageway comprises a channel in said base section of said piston head open to said annular clearance space and extending between said first actuating cavity and said radially outer-most surface of said piston head and said second ring passageway comprises a channel in said dome section of said piston

head open to said annular clearance space and extending between said second actuating cavity and said radially outer-most surface of said piston head.

3. A piston assembly according to claim 1 further including means for biasing said first sealing ring radially outward and means for biasing said second sealing ring radially outward.

4. A piston assembly according to claim 3 wherein said means for biasing said first sealing ring comprises a first split spring ring located within said first actuating cavity, contacting said radially inner-most surface of said first sealing ring and urging said radially inner-most surface of said first sealing ring radially outward and said means for biasing said second sealing ring comprises a second split spring ring located within said second actuating cavity, contacting said radially inner-most surface of said second sealing ring and urging said radially inner-most surface of said second sealing ring radially outward.

5. A piston assembly according to claim 1 wherein said sealing rings are closed rings with rectangular radial cross sections.

6. A piston assembly according to claim 1 wherein said sealing rings are manufactured from tetrafluoroethylene fluorocarbon.

7. A piston assembly according to claim 1 wherein said interior cavity and said radially outer-most surface of said sealing washer are connected by a cavity passageway.

8. A piston assembly according to claim 1 further including a first gasket between said sealing washer and said base section of said piston head and a second gasket between said sealing washer and said dome section of said piston head.

9. A piston assembly according to claim 1 further including a bearing ring located in an outer perimeter groove in said piston head, said bearing ring formed of low friction material.

10. A piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, said piston assembly having:

a piston rod,

a piston head, attached to said piston rod, forming an annular clearance space and having a radially outer-most surface,

first and second closed sealing rings,

said piston assembly characterised by:

a sealing washer, positioned within said annular clearance space, having first and second opposed flat faces and radially inner-most and outer-most surfaces, said first sealing ring positioned between said sealing washer and said piston head, said first sealing ring, said sealing washer and said piston head forming a first actuating cavity, said first sealing ring positioned in a close mating relationship with said first opposed flat face of said sealing washer, said first sealing ring and said piston head forming a first ring passageway allowing working fluid to be exchanged between said first actuating cavity and said cylinder bore,

said second sealing ring positioned between said sealing washer and said piston head, said second sealing ring, said sealing washer and said piston head forming a second actuating cavity, said second sealing ring positioned in a close mating relationship with said second opposed flat face of said sealing washer, said second sealing ring and said piston head forming a second ring passageway allowing working fluid to be exchanged between said second actuating cavity and said cylinder bore.

11. A piston assembly according to claim 10 wherein said piston head has an interior cavity and said sealing washer has a cavity passageway connecting said interior cavity and said radially outer-most surface of said sealing washer.

12. A piston assembly according to claim 10 wherein said sealing washer is a closed ring, said piston head comprises joined base and dome sections and said base and dome sections are joined after said base and dome sections are placed adjacent to different opposing flat surfaces of said sealing washer.

13. A piston assembly according to claim 10 wherein said first ring passageway comprises a channel in said piston head open to said annular clearance space and extending between said first actuating cavity and said radially outer-most surface of said piston head and said second ring passageway comprises a channel in said piston head open to said annular clearance space and extending between said second actuating cavity and said radially outer-most surface of said piston head.

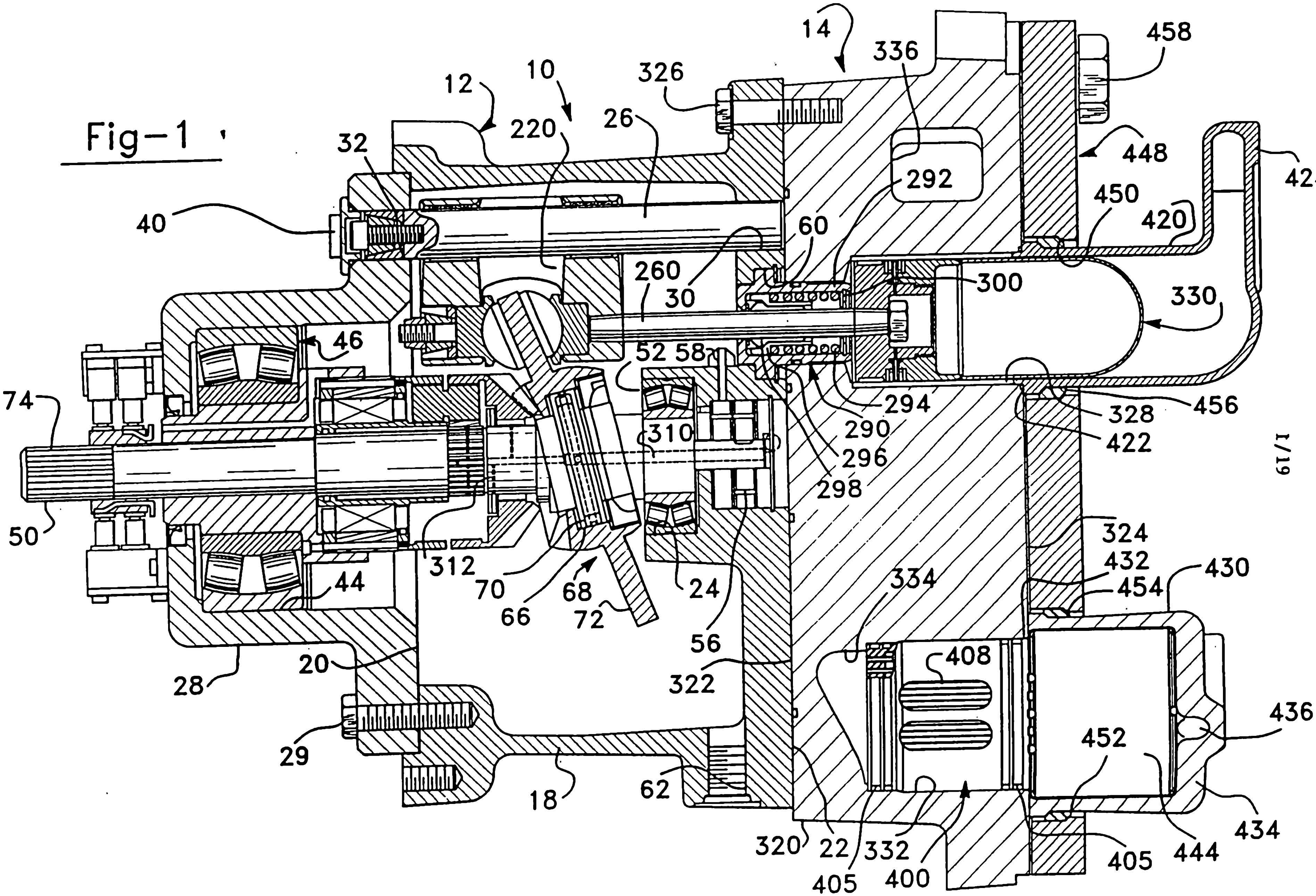
14. A piston assembly for reciprocation within a cylinder bore of a double-acting hot gas engine, such as a double-acting Stirling engine, substantially as described with reference to the drawings.

DATED this 21st day of December, 2000.

STM Corporation

by DAVIES COLLISON CAVE
Patent Attorneys for the Applicant

Fig-1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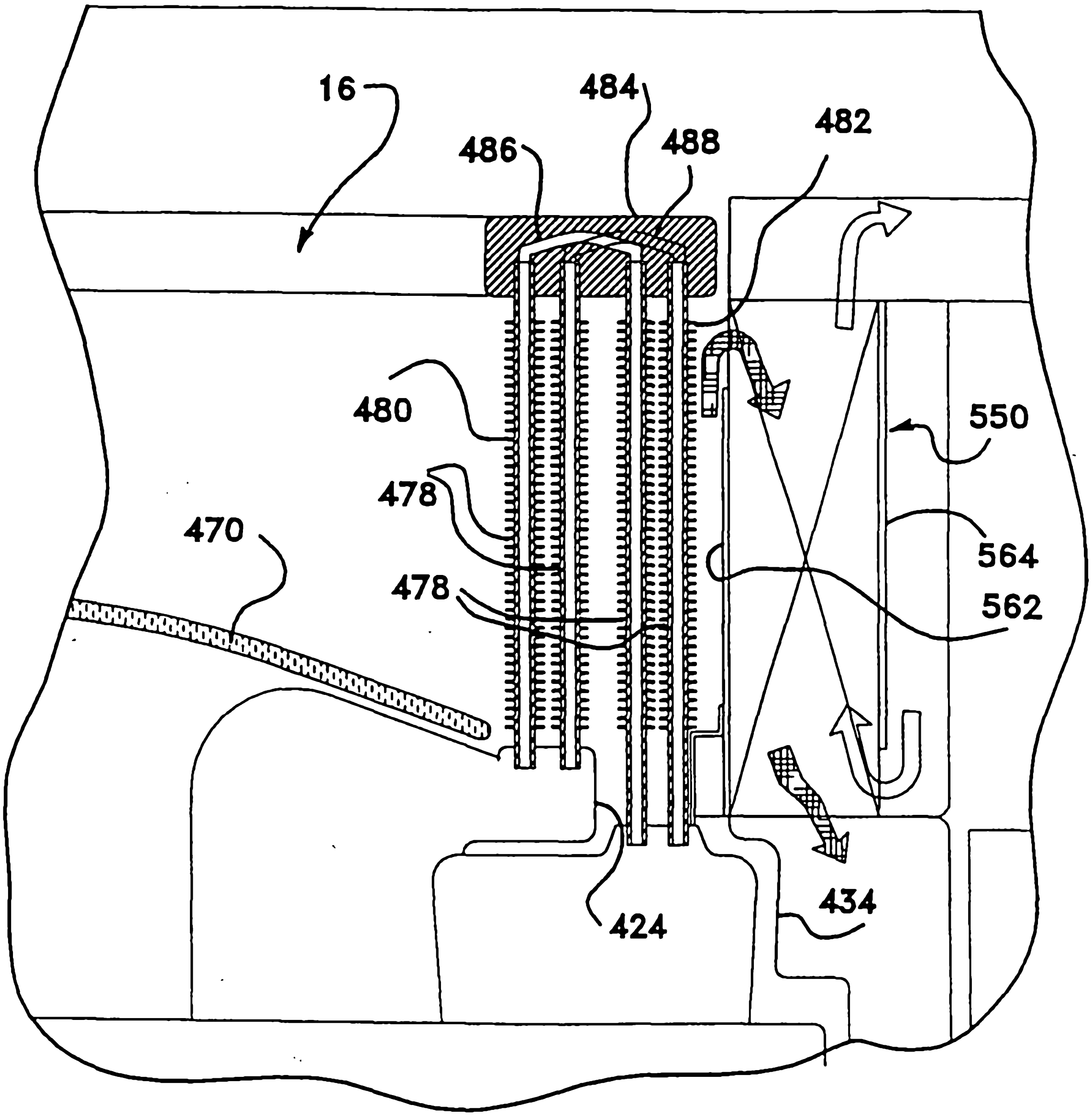


Fig—1A

Fig-1B

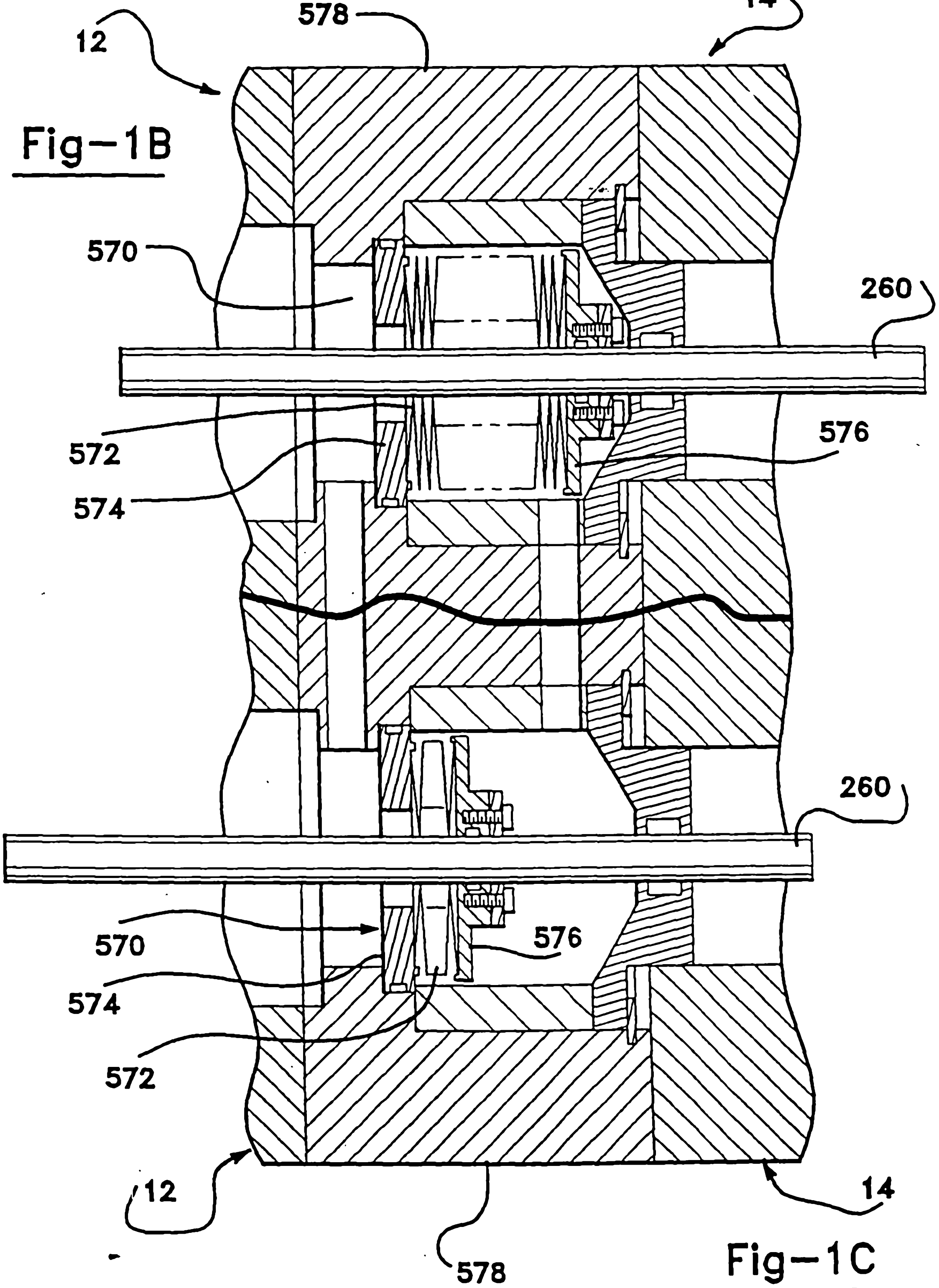
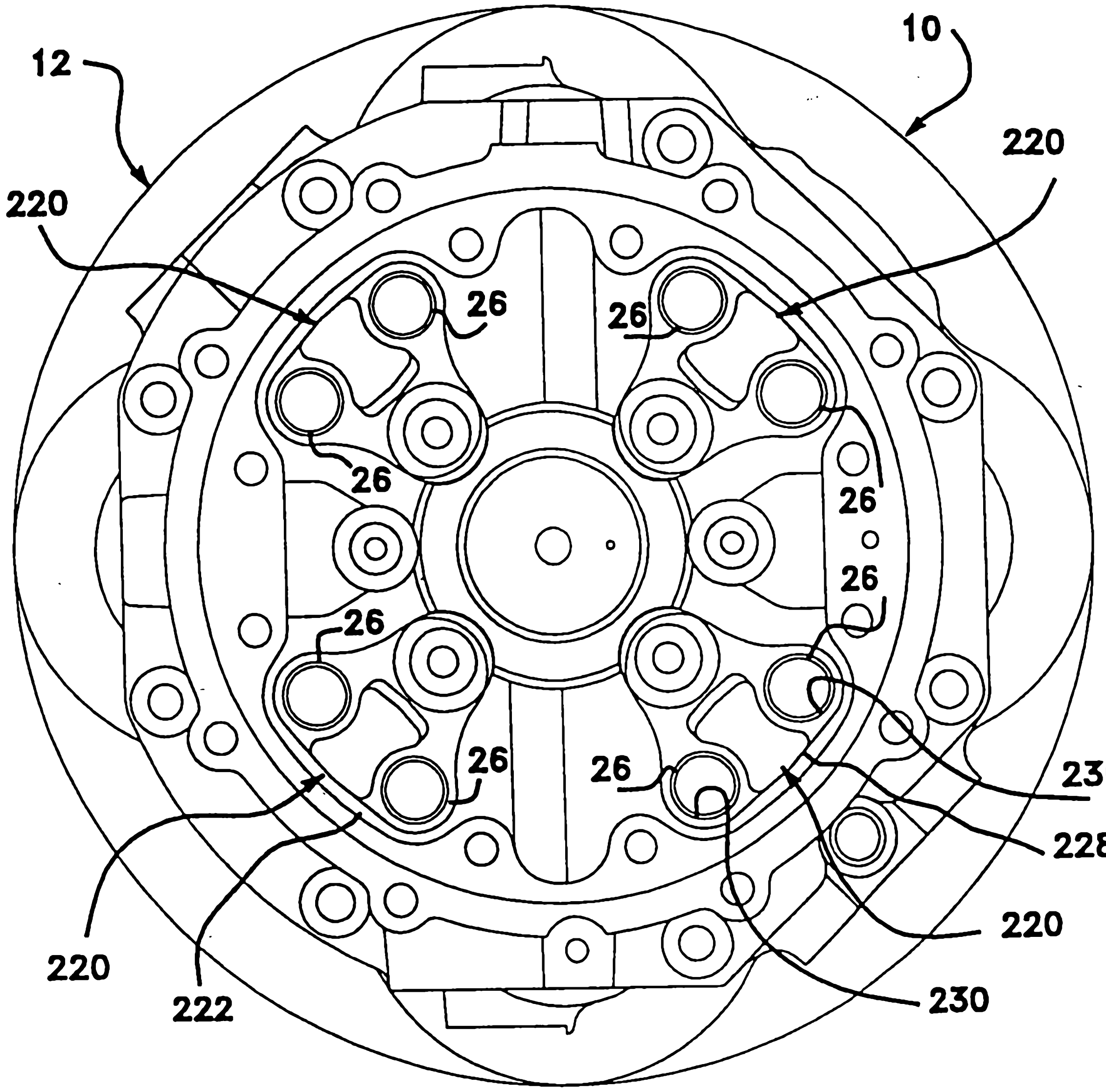


Fig-1C



Fig–2

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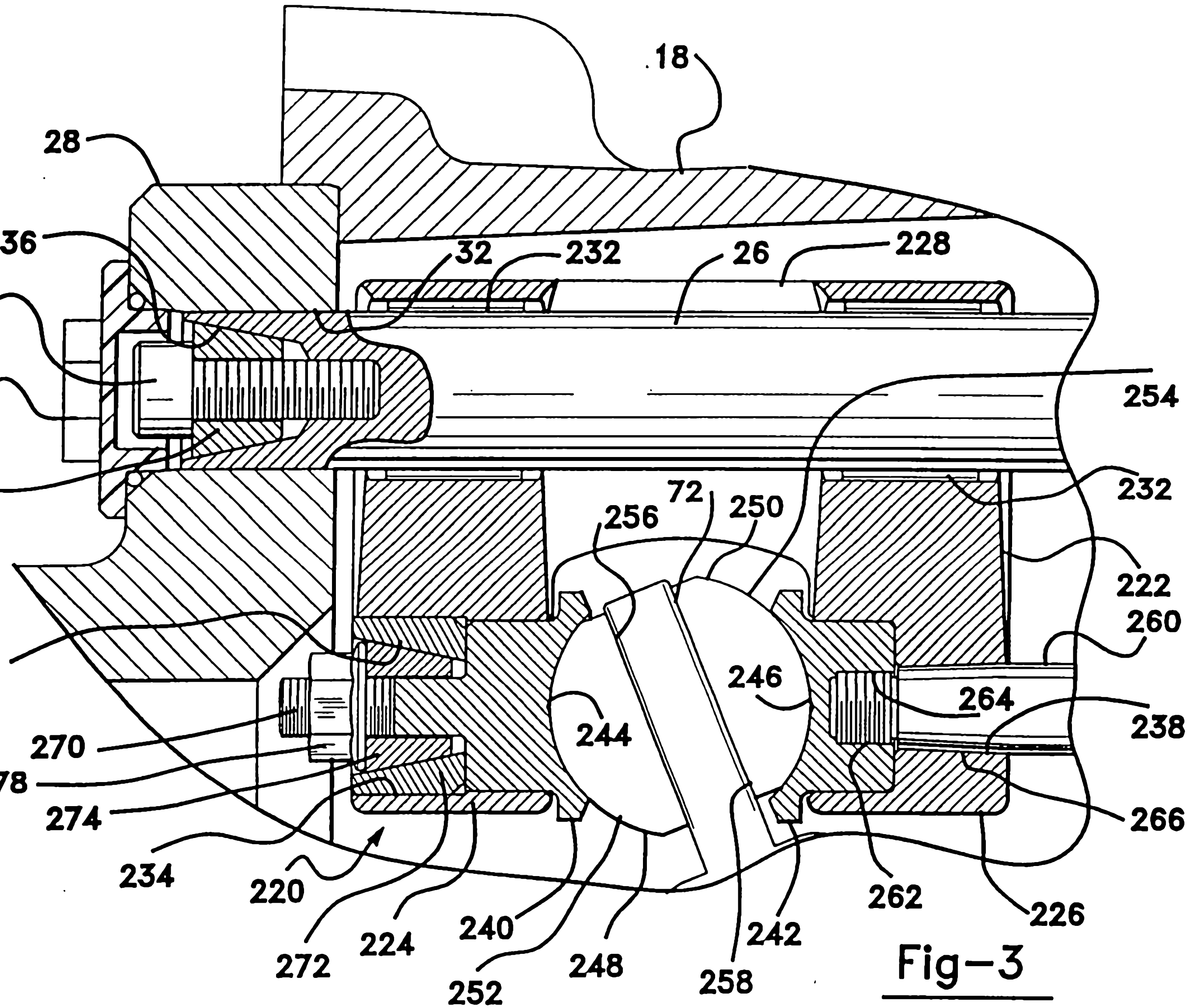

Fig-3

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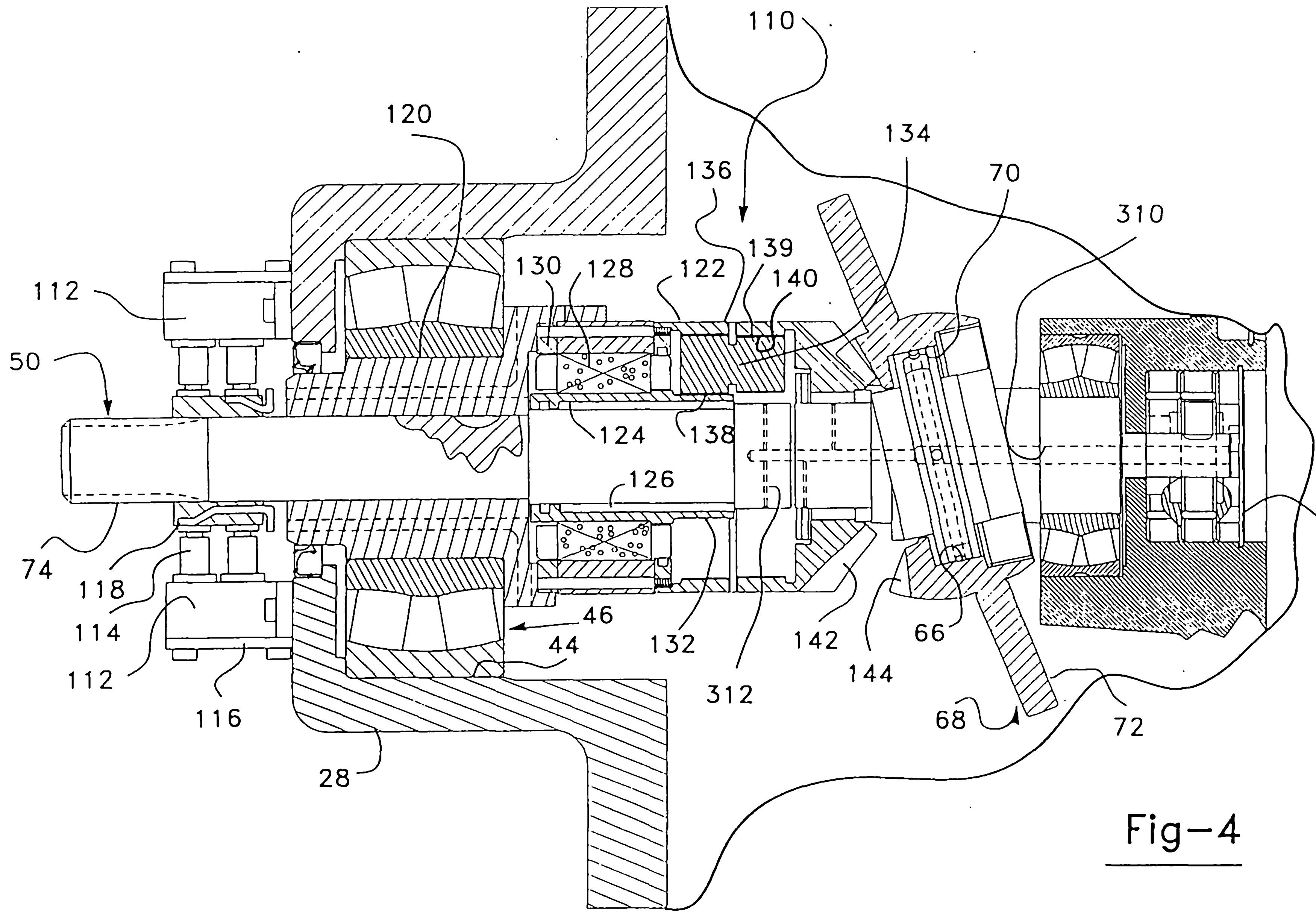

Fig—4

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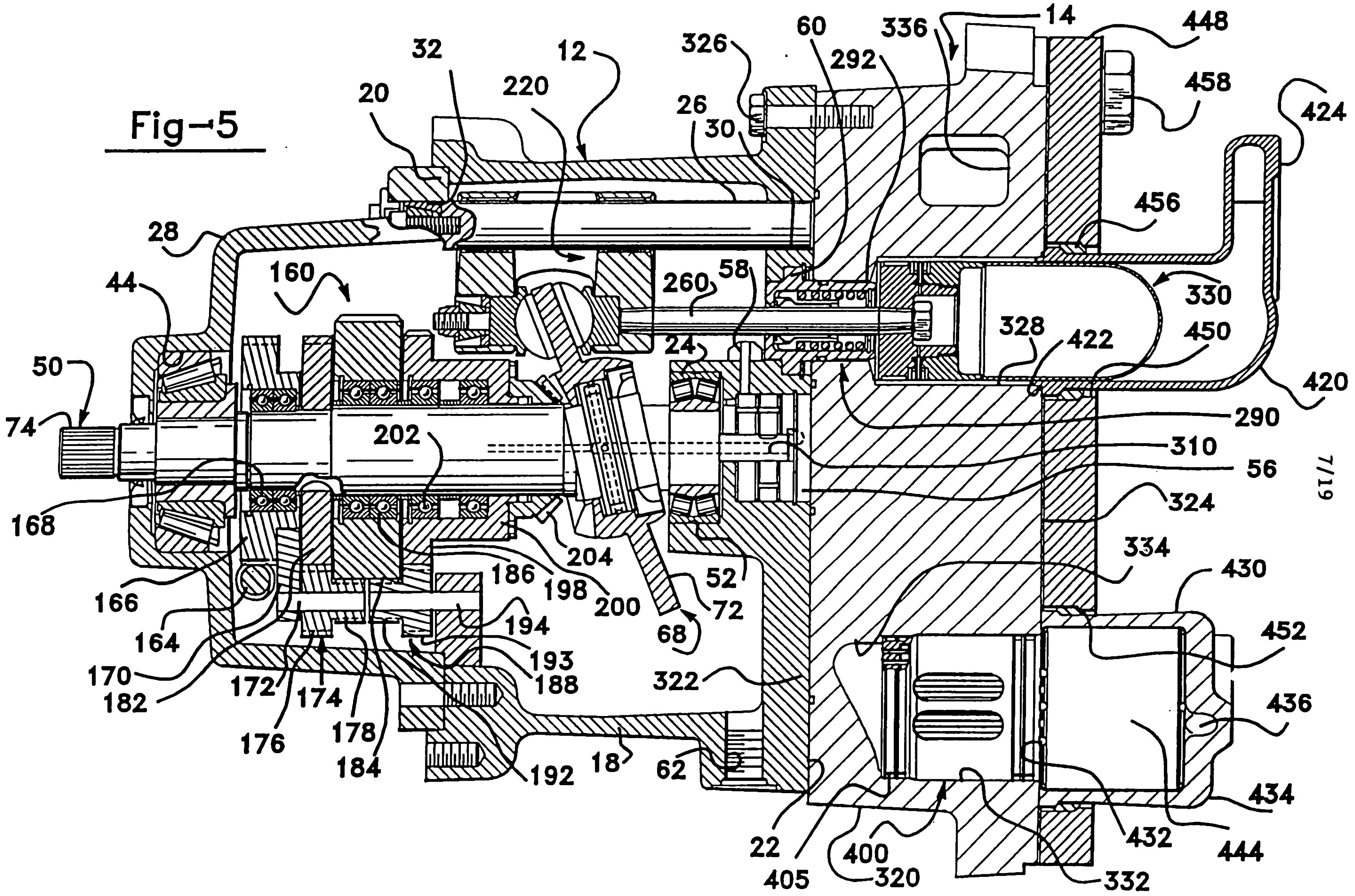
Fig-5



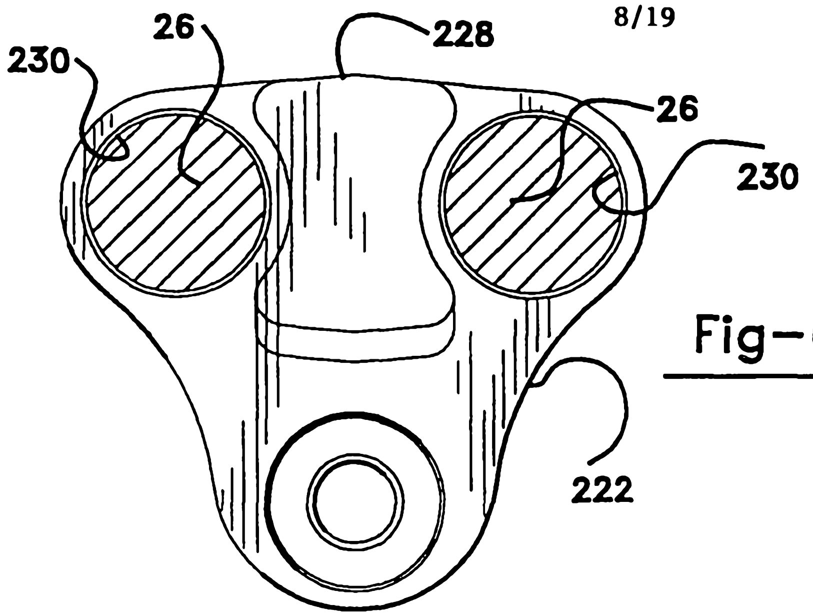


Fig-6

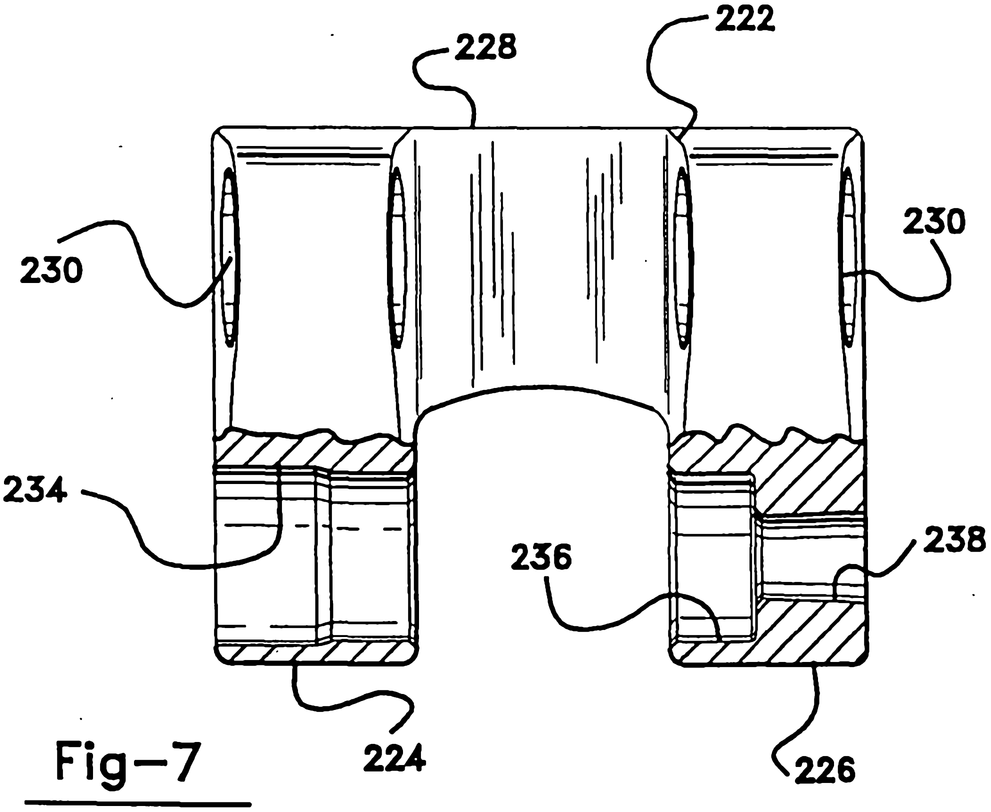


Fig-7

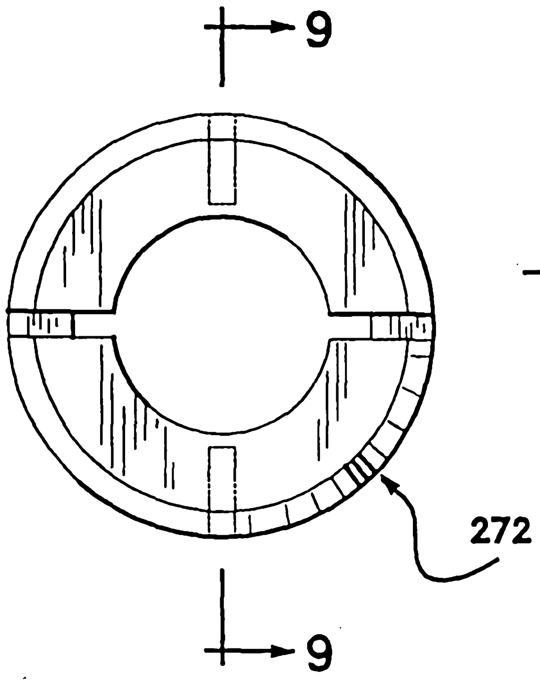


Fig-8

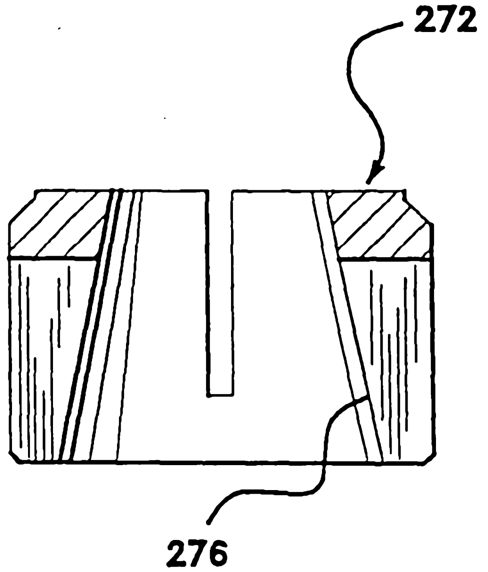


Fig-9

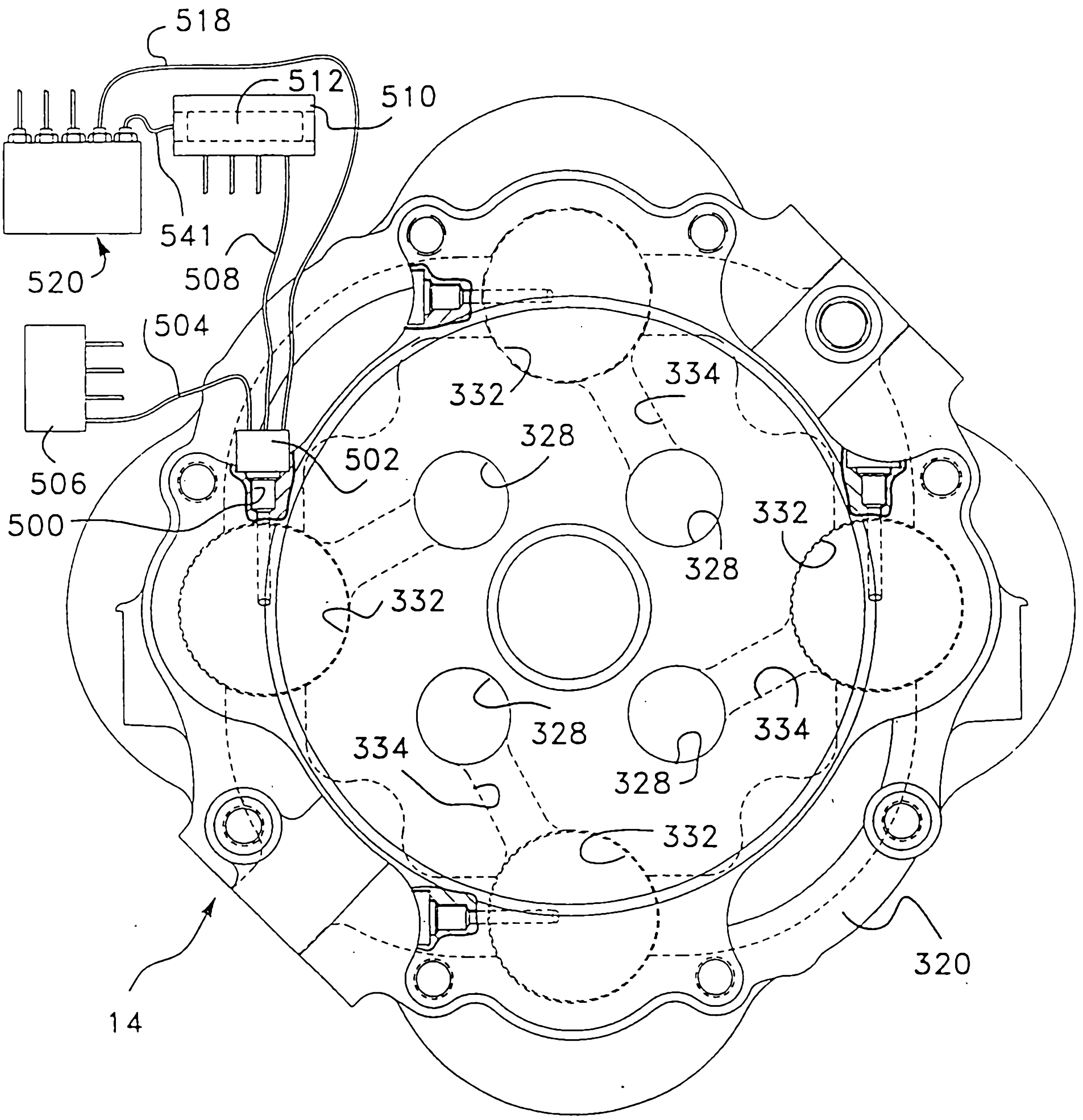

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Fig-10

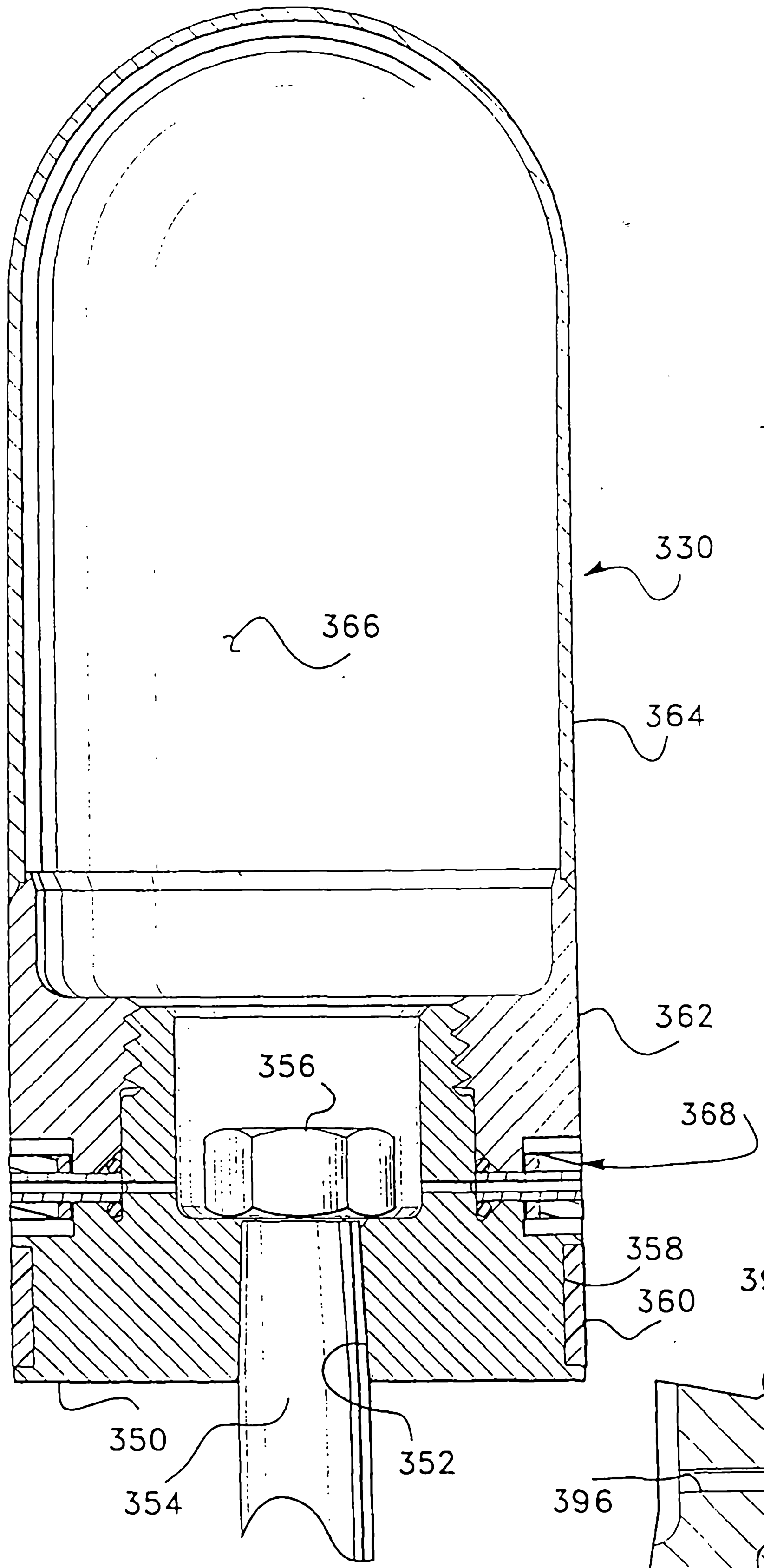
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Fig-11

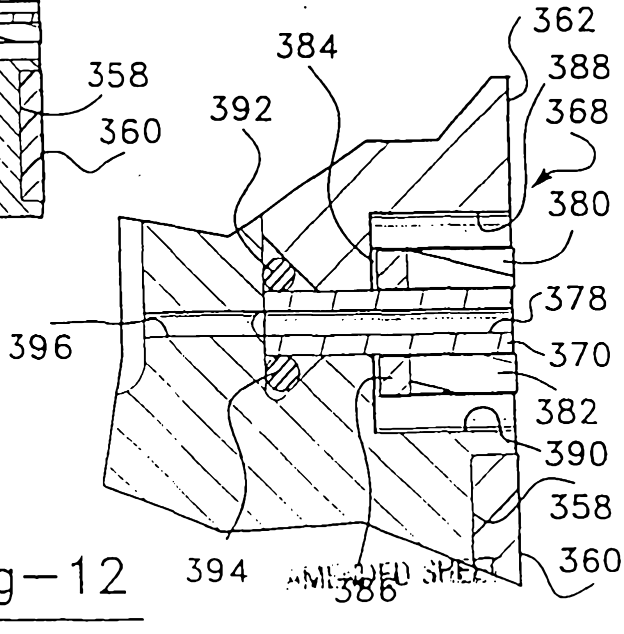
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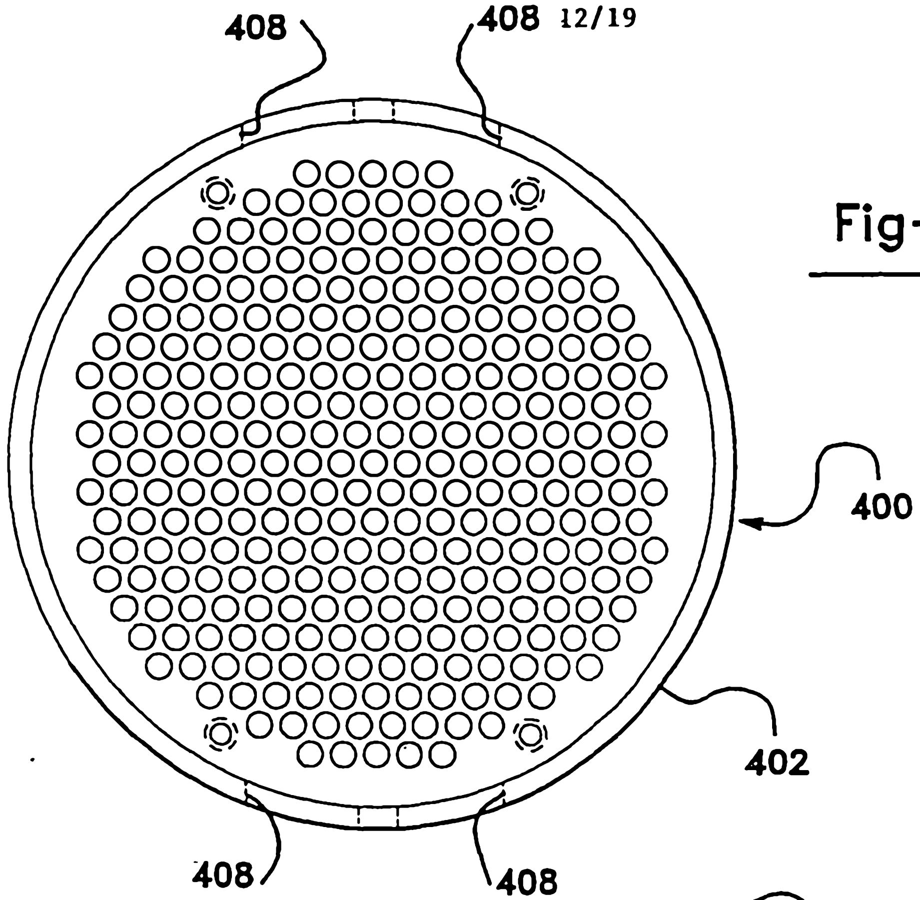
Fig-12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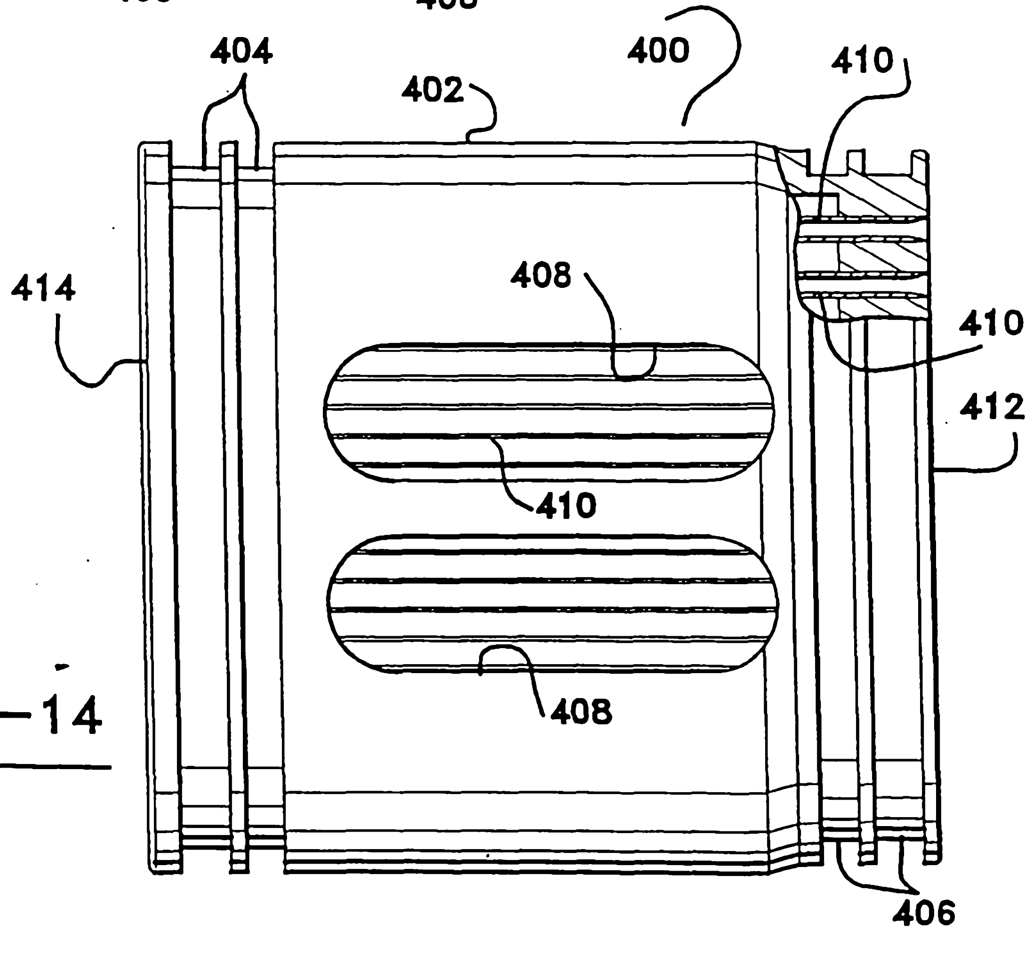
Fig—13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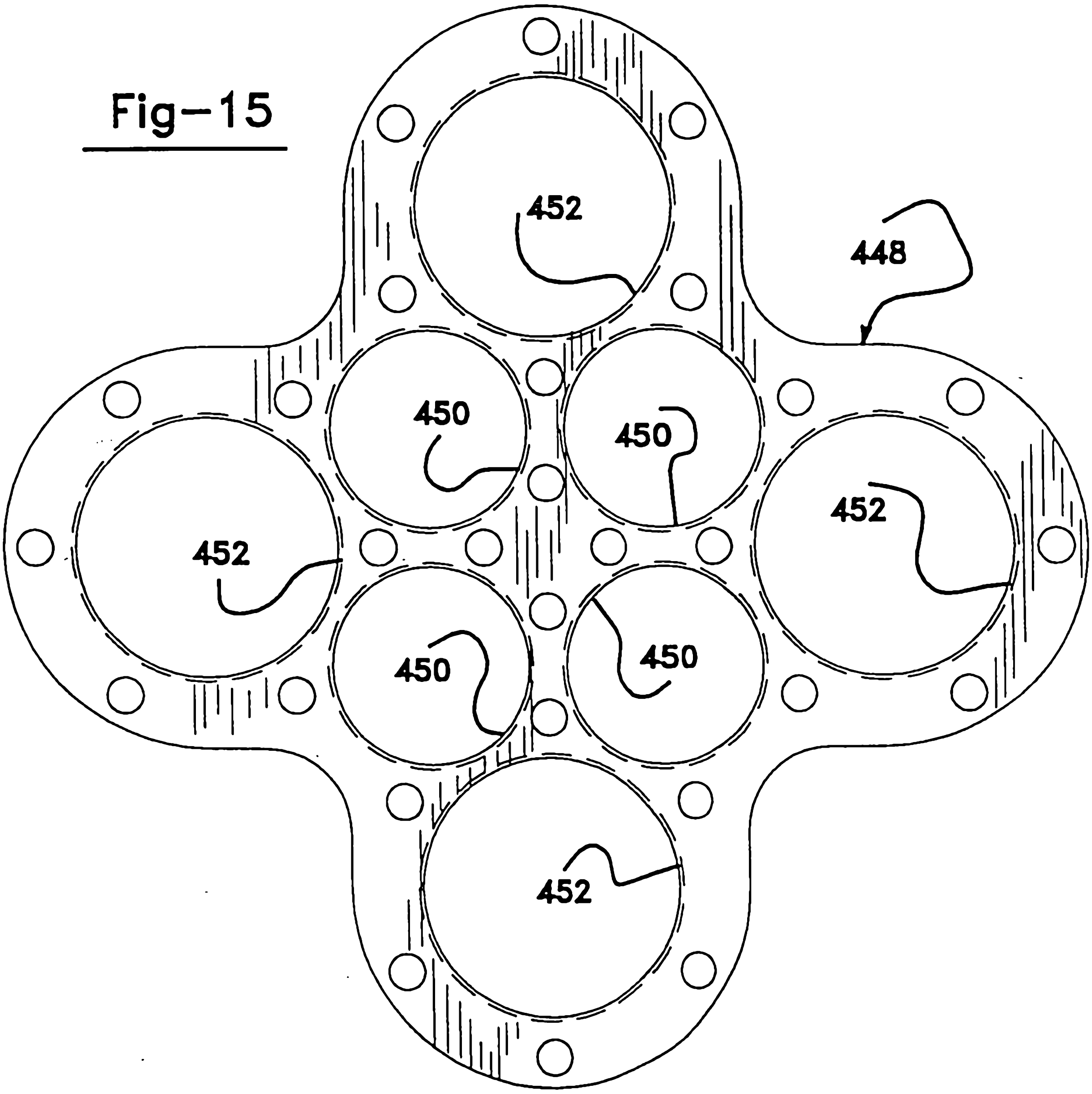
Fig—14

Fig-15



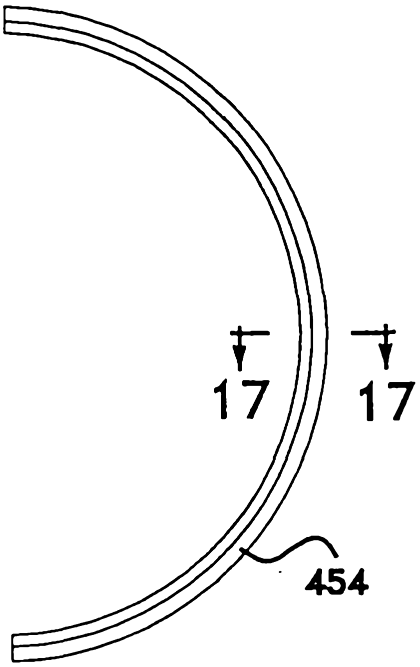


Fig-16

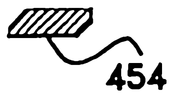
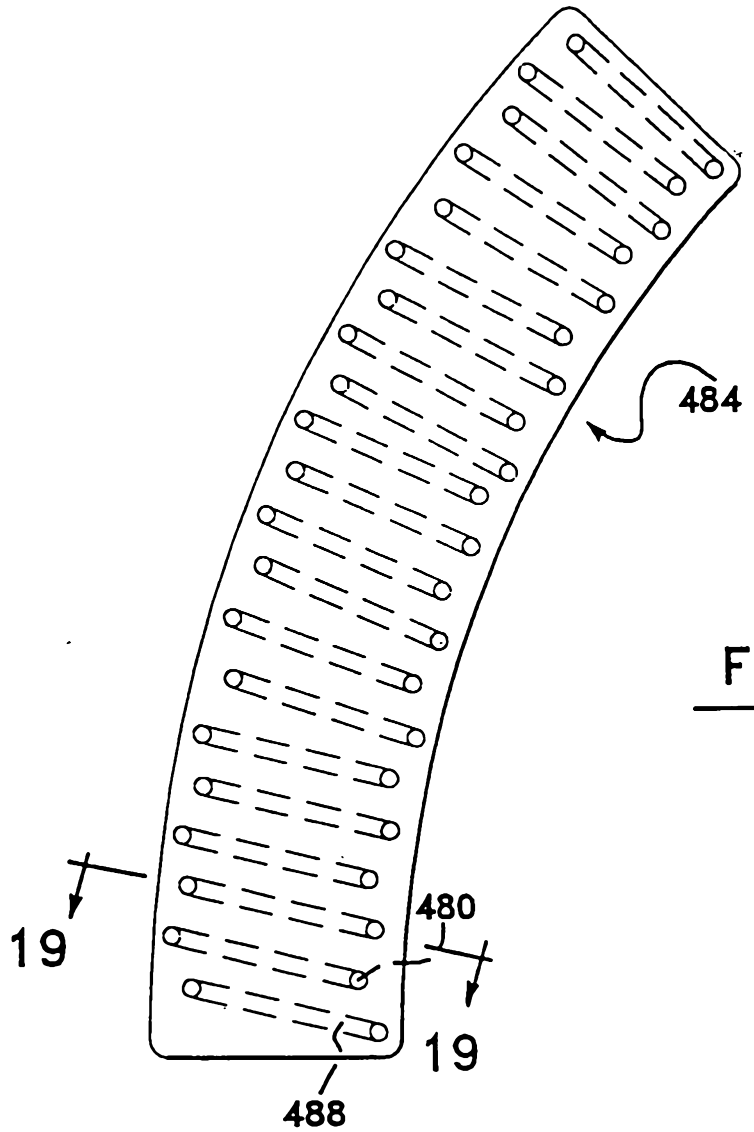
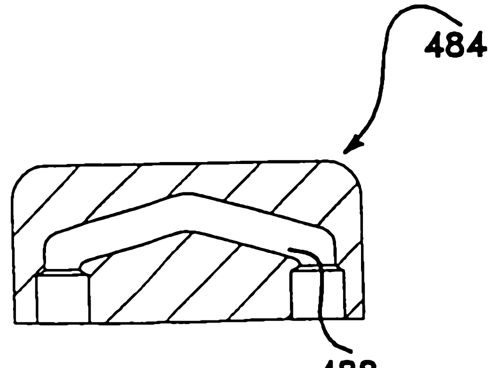


Fig-17



Fig–18



Fig–19

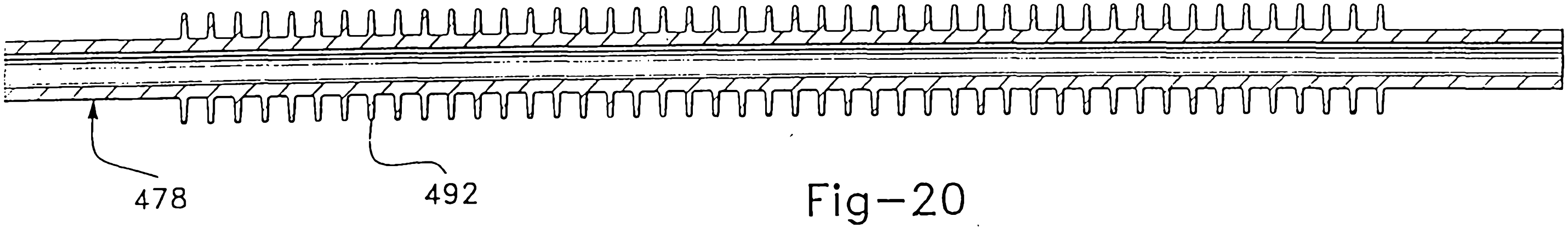


Fig-20

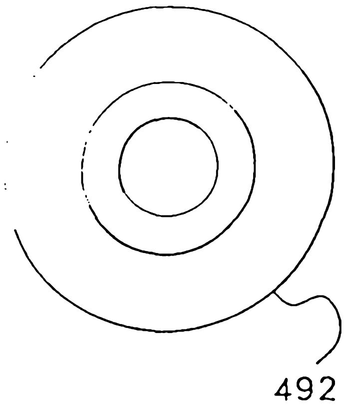


Fig-22

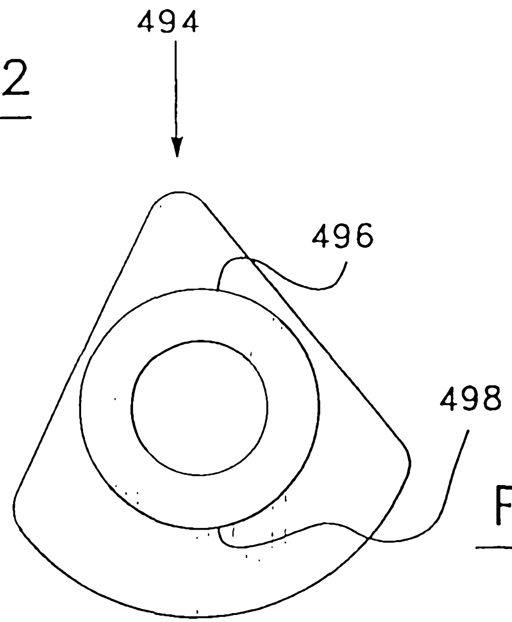


Fig-23

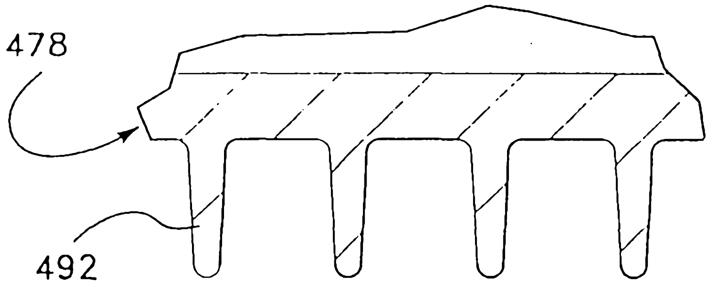


Fig-21

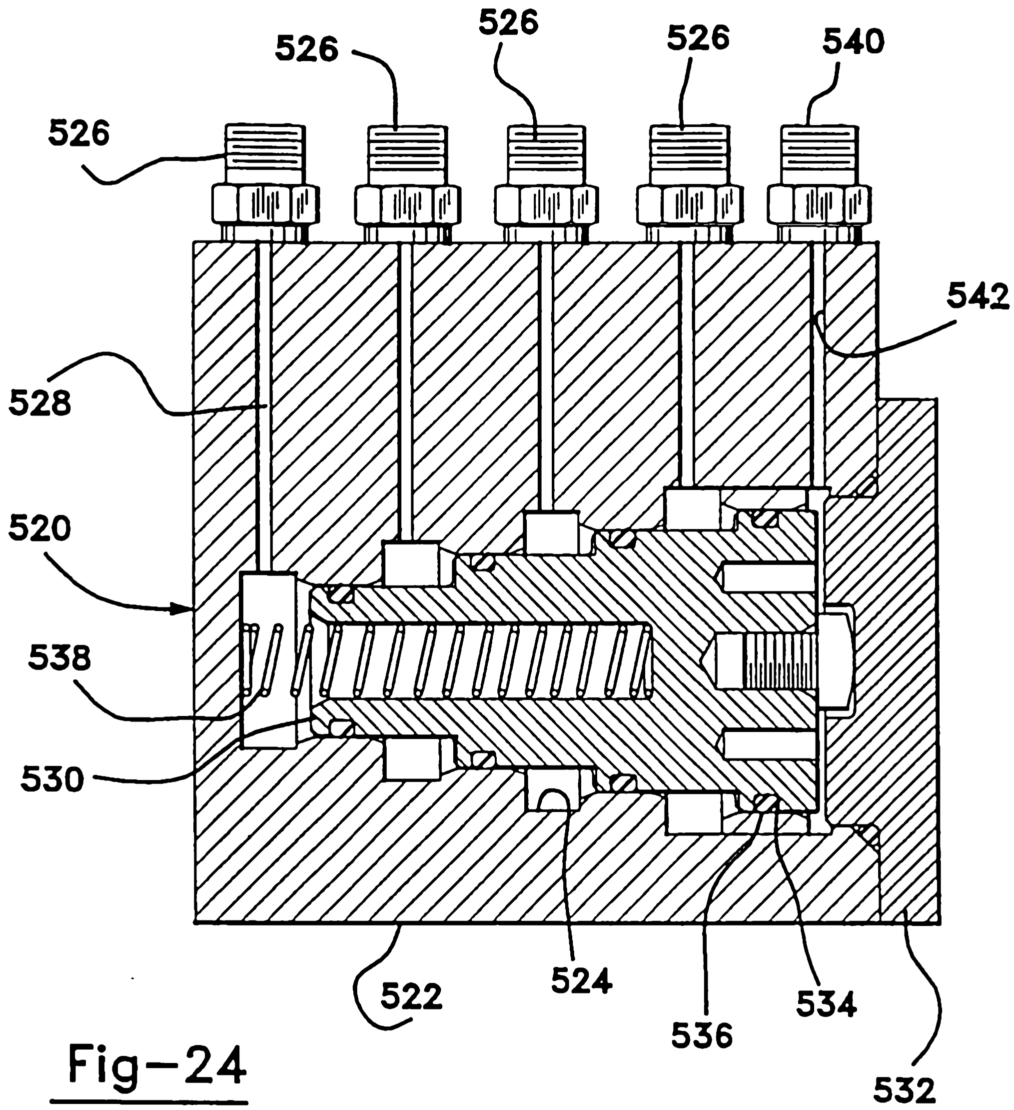


Fig-24

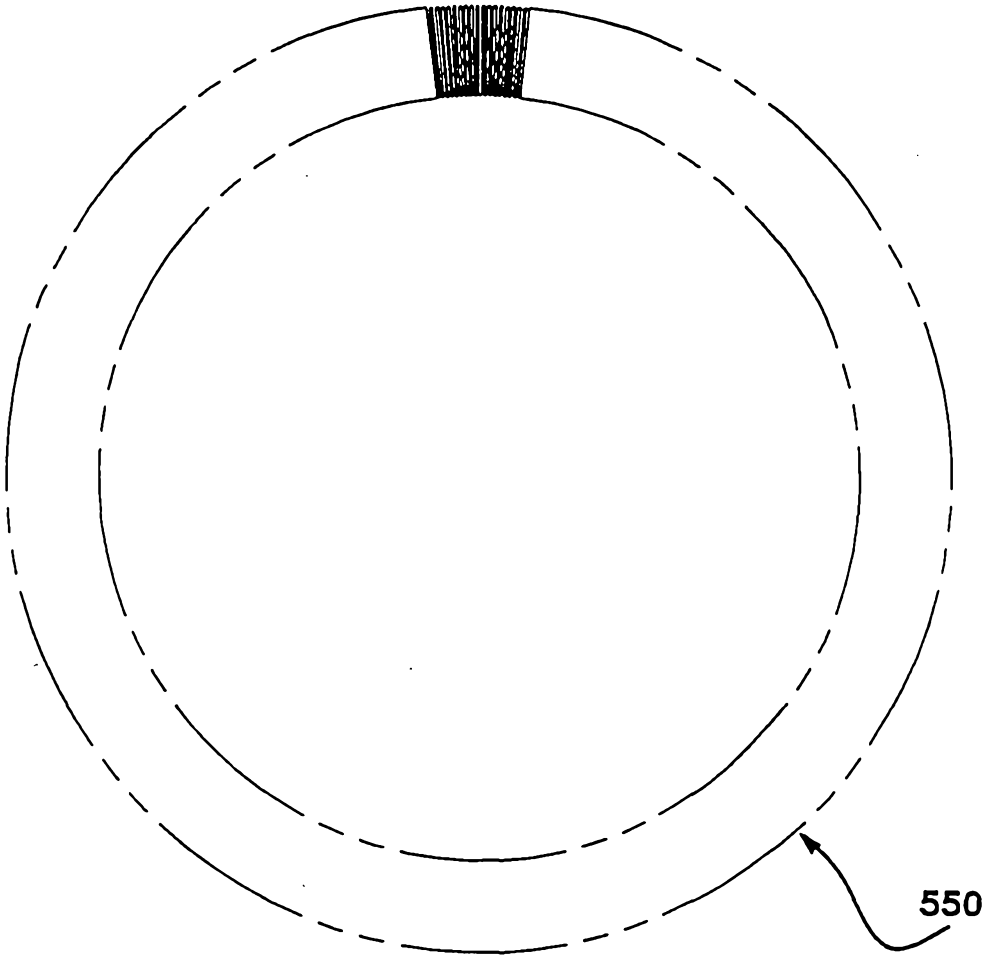


Fig-25

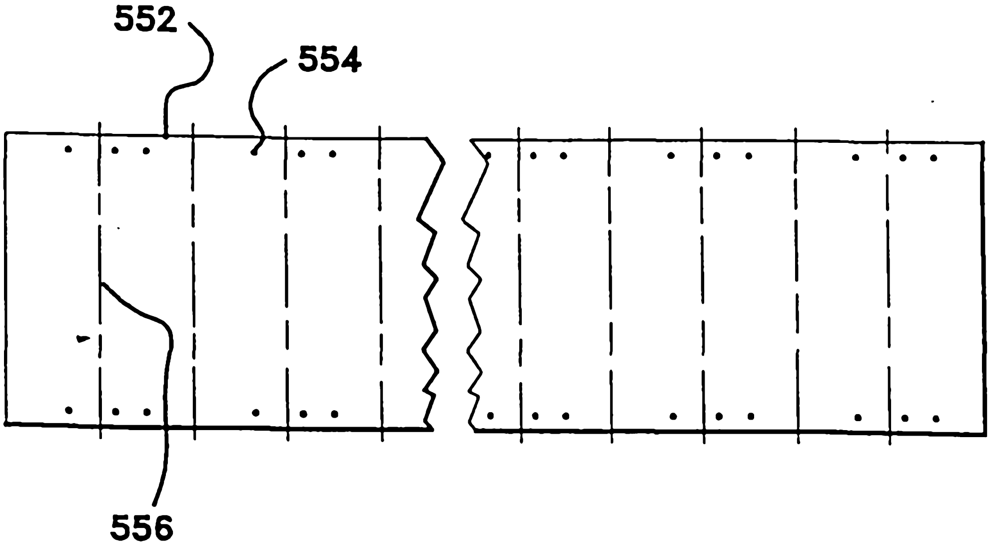


Fig-26

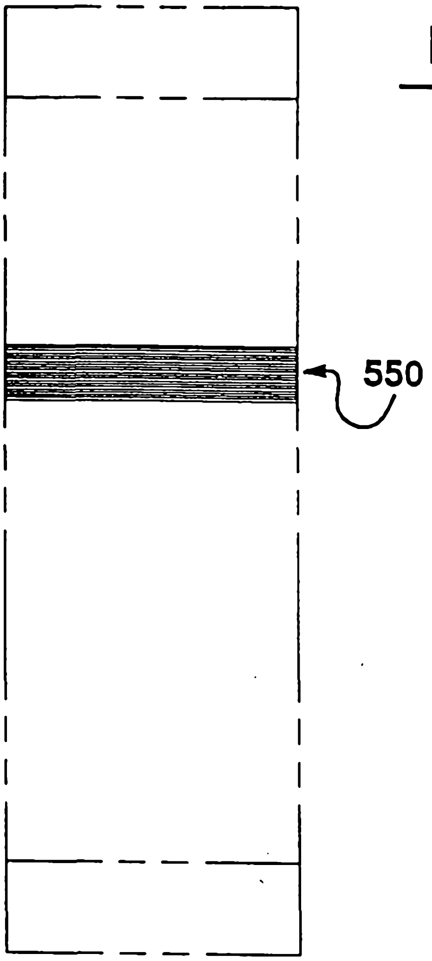


Fig-27

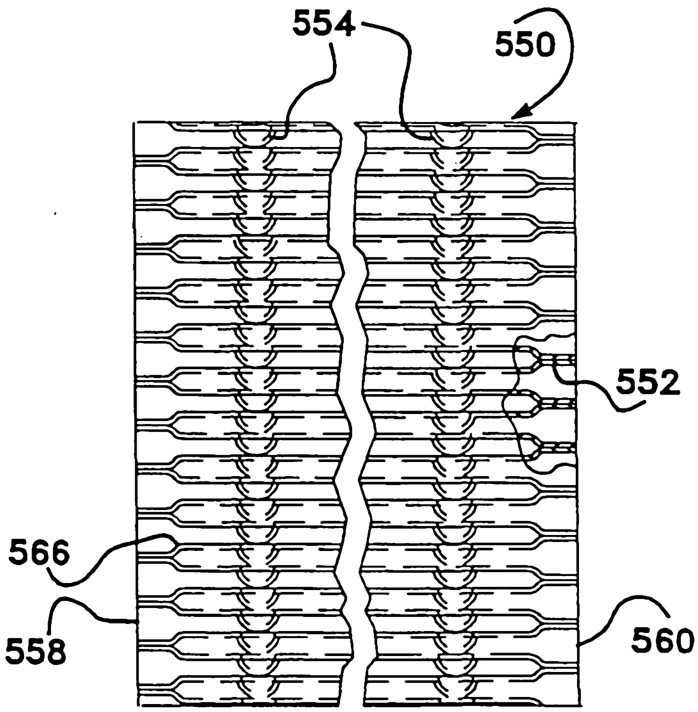


Fig-28