

(19) World Intellectual Property Organization
International Bureau



(43) International Publication Date
9 October 2003 (09.10.2003)

PCT

(10) International Publication Number
WO 03/083311 A2

(51) International Patent Classification⁷: F04D 13/06,
F02M 37/08, F04C 29/02, 29/04, 15/00

(21) International Application Number: PCT/US03/05818

(22) International Filing Date: 26 February 2003 (26.02.2003)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
60/360,693 28 February 2002 (28.02.2002) US
10/286,083 1 November 2002 (01.11.2002) US
10/286,084 1 November 2002 (01.11.2002) US
10/290,705 8 November 2002 (08.11.2002) US
10/292,226 12 November 2002 (12.11.2002) US
10/322,969 18 December 2002 (18.12.2002) US
10/340,930 13 January 2003 (13.01.2003) US

(71) Applicant: STANDEX INTERNATIONAL CORP.
[US/US]; 6 Manor Parkway, Salem, NH 03079 (US).

(72) Inventors: KIMBERLIN, Robert, R.; 217 Lehman
Trail, Murfreesboro, TN 37128 (US). HARVEY, Robert,
E., III; 121 White Bridge Road, Nashville, TN 37209

(US). JIANG, Jie; 4705 Chaepstow Drive, Nashville, TN
37211 (US). LARSON, William, M.; 411 Archer Road,
Murfreesboro, TN 37129 (US). RAWNSLEY, William,
C., Jr.; 3110 Throne Street, Murfreesboro, TN 37129
(US).

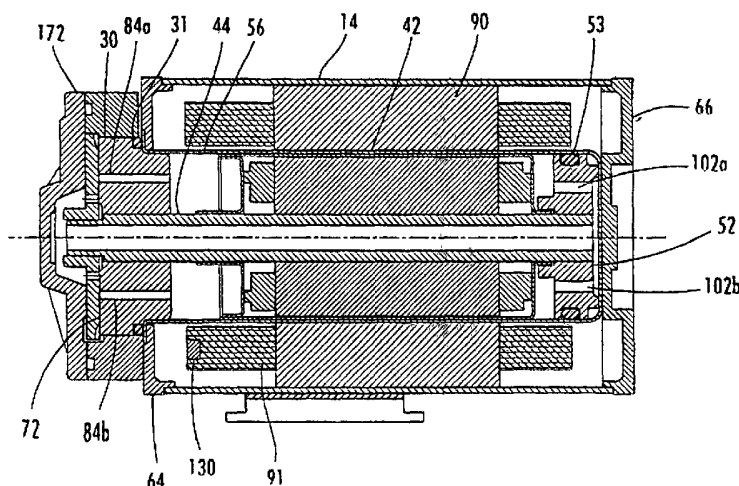
(74) Agents: SELLERS, Michael, E. et al.; Luedeka, Neely &
Graham, P.C., P.O. Box 1871, Knoxville, TN 37901 (US).

(81) Designated States (national): AE, AG, AL, AM, AT, AU,
AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU,
CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH,
GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC,
LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW,
MX, MZ, NO, NZ, OM, PH, PL, PT, RO, RU, SC, SD, SE,
SG, SK, SL, TJ, TM, TN, TR, TT, TZ, UA, UG, UZ, VC,
VN, YU, ZA, ZM, ZW.

(84) Designated States (regional): ARIPO patent (GH, GM,
KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZM, ZW),
Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM),
European patent (AT, BE, BG, CH, CY, CZ, DE, DK, EE,
ES, FI, FR, GB, GR, HU, IE, IT, LU, MC, NL, PT, SE, SI,
SK, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN,
GQ, GW, ML, MR, NE, SN, TD, TG).

[Continued on next page]

(54) Title: MOTOR PUMP



(57) Abstract: A motor pump includes a pump head for pressurizing a fluid and an electric motor for driving the pump head. Within the pump head, radial forces produced in two symmetrical pumping zones are balanced against each other to reduce stress and increase efficiency. High pressure fluid exits the pump and follow a flowpath through the motor air gap, through an end bearing carrying a distal end of a hollow pump rotor shaft, and returning to the pump head by way of the hollow rotor shaft. The motor rotor shaft is axially movable with respect to the pump head, which effectively floats the motor rotor in high pressure fluid. A motor stator barrier can inhibits fluid flow into the motor stator, and a rotor barrier can inhibits fluid flow into the motor rotor. The rotor barrier can is efficiently attached to the rotor shaft in such a way as to stiffen the shaft and increase its bending resistance. The walls of the stator barrier can are thin and expand under fluid pressure to make contact with and find structural support from the stator laminations and motor endshield. Overheat conditions within the motor, as well as the pump head, are detected by positioning a heat sensing switch within the motor adjacent the pump head to shut down the motor when an overheat condition is detected.



WO 03/083311 A2



Published:

— *without international search report and to be republished upon receipt of that report*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

MOTOR PUMP

Technical Field

The present invention relates generally to motor pumps. More particularly, the present invention relates to a motor pump having a variety of unique features for improving pump operation and performance.

5

Background

Pumps for moving fluids are generally available in a variety of types and configurations. However, many pump types are used only for low pressure applications because high-pressure is generally difficult to achieve. Although pumps known generally as positive displacement pumps have been employed in both high and low pressure applications, they are particularly useful in high-pressure applications because of their somewhat unique ability to generate the higher operating pressures which some pump types are unable to achieve.

An example of an application which often calls for high-pressure is when one needs to move a potable fluid such as water in a beverage dispensing machine. Positive displacement pumps, such as vane, piston or gear pumps, are typically used for such applications. Such positive displacement pumps achieve their pumping action by pressing volumes of water into smaller spaces. Due to high-pressure requirements, tight tolerances are required, typically .0005 inch running clearance or less, in order to maintain proper hydrodynamic performance and efficiency and to prevent leak paths. The high-pressure requirements can limit the choice of materials available for manufacturing the pump. Choice of materials can also be affected by the particular type of fluid being pumped. Additionally, these tight tolerance applications usually require precision grinding and other manufacturing methods that prevent multiple features from being incorporated into a single part. Therefore, many parts are typically needed in the fabrication of the pump.

In addition to the high-pressure requirements, positive displacement pumps exhibit other operational and manufacturing challenges including pressure/load imbalance, pump cavitation, pump overheating, rotor shaft flexing, drive coupling wear, and mechanical seal leaks. Unfortunately, solutions to these challenges as provided by prior art pumps have been less than optimal and in some cases, completely ineffective.

What is needed, therefore, is an improved pump that overcomes the difficulties and disadvantages of prior art pumps.

Brief Description of the Drawings

Preferred embodiments of the invention will now be described in further detail. Other features, aspects, and advantages of the present invention will become better understood with regard to the following detailed description, appended claims, and accompanying drawings (which are not to scale) where:

FIG. 1 is a side view of a canned motor pump according to the present invention;

FIG. 2 is an end view of the pump head for the canned motor pump of FIG. 1;

FIG. 3 is an exploded view of a canned motor pump with a rotary vane pump head according to the invention;

FIG. 4 is a cross-section view of a canned motor pump with a rotary vane pump head according to the invention with the pump in a low-pressure state;

FIG. 5 is an open view of the rotary vane pump head of FIG. 4;

FIG. 6 is an isometric view of an end bearing according to the invention showing contoured inlet passages;

FIG. 7 is an isometric view of a front bearing according to the invention showing contoured fluid pockets;

FIG. 8 is a cross-section view of a canned motor pump according to the invention with the pump in a high-pressure state;

FIG. 9 is a cross-section view of a stator barrier can according to the invention;

FIG. 10 is a cross-section view of a canned rotor with hollow shaft according to the invention;

FIG. 11 is a cross-section view of a pump head according to the invention showing the pressure relief valve;

FIG. 12 is a cross-section view of a pump head subassembly according to the invention with the air gap clearance between the stator and rotor barrier cans slightly exaggerated for purposes of clarity;

FIG. 13 is a cross-section view of a canned motor pump with a closed shroud centrifical pump head according to the invention; and

FIG. 14 is a cross-section view of a canned motor pump with a turbine pump head according to the invention.

Detailed Description of the Preferred Embodiment(s)

Preferred embodiments of the present invention will now be described with reference to the accompanying drawings, wherein like reference characters designate like or similar parts throughout. The terminology used herein is intended to be interpreted in its broadest reasonable manner, even though it is being utilized in conjunction with a detailed description of certain specific preferred embodiments of the present invention. This is further emphasized below with respect to some particular terms used herein. Any terminology intended to be interpreted by the reader in any restricted manner will be overtly and specifically defined as such in this specification.

FIGS. 1 and 2 illustrate, respectively, side and end views of a canned motor pump according to a preferred embodiment of the invention. The pump 10 includes a pump head housing 12 which houses internal components of the pump attached to a motor housing 14 for housing electric motor components. The pump head housing 12, which functions to receive a fluid at a first pressure and output the fluid at a second pressure greater than the first pressure, is attached to the motor housing 14 as by bolts 16a-d or other suitable attachment or fastening devices. A mounting bracket 18 is attached to the motor housing 14 to facilitate installation of the motor pump 10. The electric motor contained in motor housing 14 operates to drive pump head components contained in the pump head housing 12 to produce flow and pressure of a pumped fluid, such as potable water for use in beverage machines. Although other types of motors may be used in the practice of the invention, the particular motor type illustrated in FIGS. 1 and 2 is a 1/2 horsepower, permanent split capacitor (PSC) motor with a capacitor housing 21.

As shown in the exploded view of FIG. 3, the pump head 22 includes the pump head housing 12, front pump head O-ring 23, front bearing 24, cam ring 26, pump rotor 28 with sliding vanes 40, pin 32, and rear bearing 30. A total of eight (8) of the vanes 40 are preferably employed for the particular pump rotor 28 shown in FIG. 3, although more or less vanes 40 may be employed if desired. For purposes of clarity, only two (2) of the vanes 40 are depicted in FIG. 3. The front bearing 24, cam ring 26, and rear bearing 30 are preferably fabricated from a suitable material (such as carbon, silicon, graphite, polymer, glass, and steel), and these parts are preferably locked to one another by use of the pin 32 positioned in grooves 34, 36 provided in the outer surfaces of the front bearing 24 and the cam ring 26, respectively, and in a detent 38 formed in the rear bearing 30. The pump rotor 28 is preferably fabricated from stainless steel and includes a plurality of slots each of which contains a vane 40. A rear pump head O-ring 31 provides a seal between the pump head 22 and motor 14.

With continued reference to FIG. 3, a motor rotor 42 (preferably a canned rotor) includes a shaft 44 (preferably a hollow shaft) having a keyed end 46 which penetrates through openings 48, 50 formed in the rear bearing 30 and pump rotor 28, respectively. The pump rotor opening 50 is keyed to the rotor shaft 44 so that rotation of the rotor shaft 44 in response to a magnetic field produced by operation of the motor stator drives the pump head 22 by rotating the pump rotor 28 and its vanes 40. The distal end of the rotor shaft 44 is supported by motor bearing 52 which is attached to the closed end 54 of a stator barrier can 56. A motor bearing O-ring 53 is positioned between the motor bearing 52 and the inner surface of the stator barrier can 56. At least a portion of the stator barrier can 56 is positioned in the air gap of electric motor 58 and functions to protect one or more magnetic field producing elements of the stator (including the stator laminations and associate field windings) from contact with the pumped fluid. The stator barrier can 56 also preferably includes a flange 60 adjacent to the open end of the can 56 which is received by an annular seat 62 formed on one side of the front motor endshield 64. Preferably, a rear motor endshield 66, or end cap, is provided for closure of the rear portion of the motor 58.

With reference now to FIGS. 4-8, inlet fluid enters the pump head housing 12 at a low pressure inlet 20. The fluid moves from low pressure inlet 20 into an intake plenum chamber 70 defined by an annular space surrounding the cam ring 26 and bounded on one side by the pump head housing 12 with the other side bounded by a front surface 72 of the rear bearing 30. Opposed fluid intake channels 72a, 72b formed on the rear bearing front surface 72 provide flow channels for fluid to flow from the intake plenum chamber 70 to suction zones 74a, 74b of opposed fluid pumping chambers 76a, 76b formed between the pump rotor 28 and the cam ring 26. Fluid flowing from pockets 73a, 73b (FIG. 7) formed in the front bearing 24 flows into the suction zones 74a, 74b. The fluid intake channels 72a, 72b of the rear bearing 30 are illustrated in FIG. 6, and the fluid pockets 73a, 73b of the front bearing 24 are illustrated in FIG. 7. As can be seen in FIG. 6, upstream edges 78a, 78b of the fluid intake channels 72a, 72b are each preferably contoured to enhance flow characteristics, reduce noise, and reduce cavitation. As similarly shown in FIG. 7, upstream edges 79a, 79b of the pockets 73a, 73b formed in the front bearing 24 are also preferably contoured to enhance flow characteristics and to reduce noise and cavitation.

Referring again to FIG. 5, the pumping chambers 76a, 76b may be formed by making the pump rotor 28 substantially round and making the inner surface of the cam ring 26 substantially oval or oblong. Within the pumping chambers 76a, 76b, fluid is moved by the sweeping action of the vanes 40 through pumping zones 80a, 80b and then into pumping zones 82a, 82b. The

vanes 40 slide along the inner surface of the cam ring 26 as the vanes 40 are moved by the rotor 28. The increasing volume within the suction zones 74a, 74b functions to draw fluid by suction force from the intake plenum chamber 70 into the pumping chambers 76a, 76b. Likewise, the decreasing volume within the pumping zones 82a, 82b functions to force the flow out of the pumping zones 82a, 82b through one or more rear bearing discharge passages 84a, 84b and through one or more front bearing discharge passages 85a, 85b. The flow of pressurized fluid through passages 84a, 84b and 85a, 85b produces thrust which acts inwardly on the front and rear bearings 24, 30, and the opposed relation of the rear bearing discharge passages 84a, 84b and front bearing discharge passages 85a, 85b provides advantageous balancing of this thrust which acts to hold the bearings 24, 30, rotor 28, and cam ring 26 in proper alignment and to inhibit fluid leakage between these pumping elements.

Pressure loads generated in the pumping zones 82a, 82b exert forces generally in a radial direction on the pump rotor 28 and cam ring 26. Since the pumping zones 82a, 82b are in opposition to one another, pressure loads are transmitted substantially symmetrically to the pump rotor 28 and cam ring 26. This substantially even distribution of loads tends to balance the loads transmitted to the motor shaft 44, thereby reducing bearing loads acting upon the rear bearing 30. The load balancing also tends to inhibit separation of pump head parts. It has been observed that unbalanced loads transmitted to the rear bearing can require power levels 20 - 30% greater to deliver the same amount of pump output. Evenly distributing the loads generated in the pumping zones 82a, 82b also has the advantageous effect of centering the cam ring 26 within the pump head housing 12.

Referring to FIG. 8, and as explained above, pressurized fluid exits the pump head 22 in balanced fashion through one or more front bearing discharge passages 85a, 85b and one or more rear bearing discharge passages 84a, 84b. Pressurized fluid exiting passages 85a, 85b is directed to the pump head outlet 111 (as shown by flow arrow 93) without the pressurized fluid leaving the pump head 22. Pressurized fluid exiting passages 84a, 84b is directed to a fluid circulation path, at least a portion of which is external to the pump head 22, with the fluid circulation path terminating adjacent the pump head outlet 111. In a preferred embodiment, the fluid circulation path (shown by flow arrows 100) includes at least a portion of the motor 14 commonly referred to as the air gap 88. After passing through that portion of the motor air gap 88 separating the motor rotor 42 and the stator barrier can 56, pressurized fluid flow continues through motor bearing passages 102a, 102b into aft fluid chamber 104 and returning to a discharge plenum chamber 106 in the pump head 22 adjacent outlet 111 by way of hollow rotor shaft 44. The discharge plenum chamber 106 communicates directly with the pump head outlet 111.

Circulation of fluid through the pump 10 in this manner functions to remove heat from the motor 14 during operation. It also circulates fluid to the motor bearing 52 to keep it cool and clean.

5 The stator barrier can 56, or liner, provides a seal to prevent or inhibit pressurized fluid from contacting the stator laminations 90 and stator windings 91. The can 56 is defined by an open end 92 in opposed relation to a closed end 54 with a liner body between the open and closed ends. The open end 92 preferably includes an annular flange 60 which serves as the only point of attachment of the can 56 to the motor 14. Alternatively, the can 56 is secured to or within the motor 14 by other suitable structure. The stator barrier can 56, shown also in FIG. 9, is preferably fabricated as a single metal part using a deep drawing process which produces a
10 liner body with a substantially uniform cross-section between its open and closed ends. So, in a preferred embodiment, the can 56 requires no welding. Alternatively, the can 56 is fabricated from two or more parts attached to one another, such as by welding.

In the interest of minimizing the air gap 88 to enhance motor performance, at least that portion of the can walls 94 which occupy the motor air gap 88 are preferably relatively thin. In a
15 preferred embodiment, the can walls 94 are made to be substantially thin so that they deform when loaded by pressurized fluid. This tendency to deform is preferably opposed/supported by utilization of existing motor structure. In operation, additional structural support for that portion of the can walls 94 which occupy the air gap 88 is largely provided by the motor laminations 90. Additional structural support may also be provided by the rear motor endshield 66, or end cap,
20 when the wall of closed end 54 is made thin enough to deform as a result of loading imparted by the pressurized fluid on the closed end 54.

For high pressure applications during which the pumped fluid can reach pressures of about 300 psi, the stator barrier can 56 is preferably made of stainless steel drawn to a wall thickness of about 0.015 inches. For lower operating pressures, the walls 94 of the can 56 can be
25 made even thinner.

In FIG. 4, the can 56 is shown in relaxed state with no internal fluid pressures acting upon the can 56. When loaded by sufficiently pressurized fluid acting upon inner surface 96 of the can 56, the can walls 94 will tend to deform and bear against the stator laminations 90 and, in a preferred embodiment, the rear motor endshield 66 as shown in FIG. 8. However, the endshield 66 does not have to carry the full load. Loads may also be carried by the flange 60 and its
30 supporting front motor endshield 64 as well as the stator laminations 90.

The can 56 is preferably configured to have a small clearance fit between the can 56 and the stator laminations 90 for ease of assembly. The particular application of the pump 10 can be used to determine how much clearance should be provided. For example, in applications where

the pumped fluid is to be used to remove heat from the motor 14, the clearance between the can 56 and stator laminations 90 should be sufficiently small so as to increase the amount of contact and thermal conductivity between the can 56 and the laminations 90. Conversely, for applications where it is desirable to thermally isolate the pumped fluid from the motor 14, the clearance between the can 56 and laminations 90 should be greater.

During fabrication of the can 56, it has been observed that the roundness of the can is somewhat difficult to control due to the stresses experienced in the material as the can 56 is formed. Thus, the middle portion of the can 56 is typically out-of-round. Typically, the can 56 has well controlled diameters at both ends. For applications where it is desirable to use the pumped fluid to remove heat from the motor 14, the resultant tight tolerances necessitate application of a moderate amount of force during insertion of the can 56 into the stator bore. As the can 56 is inserted into the stator bore, it will typically come to rest with one or more portions of the outer surface of the can 56 in contact with the stator laminations 90. As the pump 10 generates pressure, the can 56 expands/deforms and more fully contacts the motor laminations 90 and more fully conforms to the shape of the stator bore. Making the can 56 thin and expandable in this manner provides numerous advantages. First, since the can 56 is conformable, manufacturing imperfections in the can 56 are easily compensated. Second, the thinness of the can 56 minimizes the overall size of the motor air gap, which improves motor performance. Third, the can 56 is able to take advantage of existing motor structure to improve pump performance. Fourth, allowing the can 56 to expand against and conform to the motor laminations 90 provides for a more efficient transfer of heat from the motor stator to the fluid for purposes of motor cooling.

Typical vane pumps employ a pressure relief valve positioned in or near the low pressure inlet to prevent overpressure conditions from damaging the pump. Such a relief valve configuration tends to undesirably impede the flow of fluid into the pump. As shown in FIG. 11, the present invention eliminates the difficulties associated with known relief valve configurations by moving the pressure relief valve 142 away from the low pressure inlet 20 and high pressure outlet 111 with no portion of the relief valve 142 positioned in the inlet 20 or outlet 111. The relief valve is bi-directional, allowing the high and low pressure regions of the pump 10 to communicate with each other through the relief valve 142. This is accomplished by providing a relief valve cavity 144 in the pump head housing 12. Adjacent upper end 150 of the relief valve cavity 144 is an upper relief valve cavity passage 152 in fluid communication with the high pressure discharge plenum chamber 106. A lower relief valve cavity passage 154 is in fluid communication with the low pressure intake plenum chamber 70. A removable cap 146 provides

external access to the relief valve cavity 144, and a cap O-ring 148 provides a fluid seal between the cap 146 and pump head housing 12.

5 The relief valve 142 includes a poppet 156 with an annular seating surface held against an inner annular seat 157 of flow tube 159 by poppet spring 158 (or other suitable actuator). The flow tube 159 also includes an outer annular seat 161 which engages an annular shoulder 163 of the relief valve cavity 144 to form a seal. Flow tube spring 165 (or other suitable actuator) acts to force the seat 161 against the shoulder 163 as shown. The force applied by spring 165 is adjustable by a threaded connector 167 (or other suitable force adjustment device) which also functions to secure the relief valve 142 within its cavity 144. To prevent overtravel of the poppet 10 156, the poppet 156 is positioned adjacent the upper end 150 of the relief valve cavity 144 so that upper end 150 functions as a barrier that engages the poppet 156 when the poppet 156 moves a maximum allowed distance. Configuring the poppet 156 and relief valve cavity 144 in this manner advantageously prevents the poppet 156 from being pulled into the discharge passage 152 where the poppet 156 could cause damage to the pump 10 or otherwise cause an anomalous operating condition.

15 The relief valve 142 is configured to allow flow through the relief valve cavity 144 in either direction. In low pressure relief mode, fluid flows from the inlet 20 to the outlet 111 when fluid pressure at the inlet 20 exceeds fluid pressure at the outlet 111 by a threshold amount, said threshold amount being greater than the force of poppet spring 158. In other words, when fluid pressure exerted on the valve poppet 156 is sufficient to overcome the force of spring 158, low pressure fluid in the low pressure intake plenum chamber 70 enters the relief valve cavity 144 and flow tube 159 through passage 154, moves the poppet 156 upwardly away from annular seat 157, and exits the cavity 144 through passage 152. In high pressure relief mode, fluid flows from the outlet 111 to the inlet 20 when fluid pressure at the outlet 111 exceeds fluid pressure at the inlet 20 by a threshold amount, said threshold amount being greater than the force of flow tube spring 165. In other words, when fluid pressure exerted on the valve poppet 156 and flow tube 159 is sufficient to overcome the force of spring 165, the upper flow tube 159 moves downwardly from its seated/closed position with high pressure fluid entering the cavity 144 through passage 152 and exiting through passage 154. The threshold amount of fluid pressure needed to overcome the force of flow tube spring 165 is greater than the threshold amount of fluid pressure needed to overcome the force of poppet spring 158.

25 Typical pump motor combinations use a pump head and a separate motor. In extreme thermal loading conditions within the pump head, the pump head will overheat. While most motors include a thermal sensor/switch to shut down the motor when the motor overheats,

overheat conditions within the pump head are not detected by the motor sensor. As a result, the motor will continue driving the overheated pump head until catastrophic failure occurs.

In accordance with one aspect of the present invention, the motor overheat sensor/switch is reconfigured in such a way as to sense overheat conditions within the pump head 22 as well as the motor 14. In a preferred embodiment, this is accomplished by positioning an overheat sensor switch 130 within the motor stator windings 91 in an area adjacent the pump head 22. Preferably, the overheat sensor/switch 130 is of a bi-metal type and integral with the stator windings 91 so as to open the circuit of the stator windings 91 and prevent motor operation when a threshold overheat temperature, preferably about 320 degrees F., is sensed. When the overheat condition is no longer present, the bi-metal switch closes and pump operation may resume. (In an alternate switch embodiment, the bi-metal switch must be manually reset before the pump resumes operation.) The bi-metal switch 130 includes a closed position providing electrical continuity through the stator windings 91 during normal operation of the motor 14, and an open position preventing electrical continuity through the stator windings 91 when an overheat condition is sensed by the bi-metal switch 130.

It will be understood that other configurations of the sensor/switch 130 may be employed in the practice of the invention. For example, instead of an integral sensor/switch, a temperature sensor may be positioned adjacent the pump head 22 in the motor windings 91 or in some other location where the sensor can sense the temperature of the pump head 22 as well as the motor 14. The output of the temperature sensor is provided to a processor or relay which activates a switch to prevent motor operation when a threshold temperature is sensed. Thus, the present invention contemplates a single sensing device that can sense the presence of an overheat condition in either the pump head 22 or motor 14 so that action can be taken to shut down or otherwise prevent operation of the pump 10 when an overheat condition is present in the pump 10. This helps prevent damage to the pump 10 in dry running conditions as well as severe overload conditions.

It will be appreciated that while prior art pumps inefficiently feed high pressure fluid through the motor and then discharge the high pressure fluid to a low pressure region of the pump (typically the inlet), the pump 10 of FIG. 8 does not suffer such a disadvantage. Instead, the pump 10 of FIG. 8 connects the pumping zones 82a, 82b of both pumping chambers 76a, 76b by splitting the flow and providing what is essentially two discrete paths of flow within the pump 10 with both portions of the flow recombining in the same discharge plenum chamber 106. One discrete flow path is from the pumping zones 82a, 82b, through front bearing discharge passages 85a, 85b, and into the discharge plenum chamber 106. The other discrete flow path is from the

pumping zones 82a, 82b, through rear bearing discharge passages 84a, 84b, into that portion of the motor air gap 88 separating the motor rotor 42 and the stator barrier can 56, through motor bearing passages 102a, 102b into aft fluid chamber 104, through the hollow rotor shaft 44, and into the discharge plenum chamber 106, as previously described.

5 In operation, flow is discharged through the pump from both sides of the pump rotor 28 through discharge passages 84a, 84b, 85a, 85b. The initial pressures in discharge passages 84a, 84b, 85a, 85b are equal, which balances the thrust loading of the pump rotor 28. Also, the discharge plenum chamber 106 is able to communicate to both sides of the pump rotor 28 through vane slot passages 108 formed when vanes 40 slide outwardly into a pumping chamber
10 76a and 76b. This allows further thrust balancing by supplying both sides of the pump rotor 28 with fluid pressure equal to that found in chamber 106. The flow path from discharge passages 85a, 85b to chamber 106 is much shorter than from discharge passages 84a, 84b to chamber 104. As fluid passes from chamber 107 to chambers 109 and 104, there is a pressure differential created by the additional distance and restricted flow area formed by the motor rotor 42 and the
15 barrier can 56 as illustrated by fluid flow arrows 100. This tends to create a slight thrust load on the pump head side of the motor rotor 42, which tends to move the motor rotor 42 away from the pump rotor 28. In addition, the motor stator laminations 90 and rotor laminations tend to align on the electrical center of the motor during operation causing further movement between the two rotors. Manufacturing tolerances and process variations create further movement of the relative
20 position of both rotors. These relative movements between the two rotors require additional torque from the motor as the two rotors thrust against their respective thrusting surfaces. To reduce this tendency, the motor rotor 42 and pump rotor 28 are connected by a sliding keyed end joint 46 to allow free axial movement of the two parts which minimizes parasitic loads in the system. Thus, rotor shaft 44 is essentially floated in high pressure fluid and axially movable with
25 respect to the pump head 22 and its pressure generating elements. This has the desirable effect of enhancing the balance of forces acting on the motor rotor 42 which in turn enhances motor operation and efficiency. It is also noted that the flow of pressurized fluid through the air gap 88 is independent of axial movement of the rotor shaft 44.

The motor rotor 42, as shown in greater detail in FIG. 10, includes the hollow shaft 44
30 onto which the rotor laminations 110 are attached. The rotor laminations 110 are sealed from and protected against the pumped fluid by a thin rotor barrier, or can 112. In a preferred embodiment, the can 112 includes no more than two parts assembled about the rotor laminations 110 using no more than three areas of attachment. As illustrated in FIG. 10, such a can 112 includes a body portion 114 disposed about the rotor laminations 110. The body portion 114

includes an insertion end 115 with an opening for receiving the laminations 110. At the other end of the body portion 114 is an integrally formed end wall 117 forming a single part with the body portion 114. The end wall 117 includes an end wall opening 119 which is attached to the rotor shaft 44 in a way that prevents fluid from entering the barrier between the shaft 44 and the end wall opening 119. The can 112 also includes a cap 116 having an outer perimeter 121 attached to the large opening of the body portion 112 in a way that prevents fluid from entering the barrier between the cap outer perimeter 121 and the body portion opening. The cap 116 includes a central cap opening 123 which is attached to the rotor shaft 44 in a way that prevents fluid from entering the barrier between the shaft 44 and the cap opening 123.

If desired, spacers 118a-d interconnecting the laminations 110 and can 112 may be employed to provide added structural support. The can 112 may be assembled to the shaft 44 by applying a positive mechanical attachment method (such as a weld, pressure fit, or adhesive) at the three attachment areas 120.

Since the shaft 44 is hollow, it has the potential for greater deflection during operation. Electric motors exhibit a characteristic commonly referred to as "side pull" which results in the bending or movement of the rotor 42 during startup. Bending of the rotor 42 as a result of side pull is particularly noticeable when the running center of the rotor 42 is not coincident with the stator centerline as a result of manufacturing imperfections. Application of the rotor barrier can 112 to the rotor 42, in combination with positive mechanical attachments 120, provides additional stiffness which significantly improves the overall bending resistance of the rotor 42. For example, a 0.5 inch diameter shaft has a polar moment of inertia of about 0.00306 (in⁴). The addition of a 0.25 inch diameter hole through the shaft 44 reduces the polar moment of inertia by about 6% to 0.00287 (in⁴). The addition of a rotor barrier can 112 with a 0.014 inch wall thickness increases the moment of inertia to a value of 0.597 (in⁴) even with the hollow shaft 44. This is an increase of nearly 15 times the original shaft stiffness. This has been observed to effectively reduce rotor deflection, during startup, by about the same ratio.

Another advantage of the two-piece rotor can 112 configuration is that the end wall 117 provides a structure that can be used to push the body portion 114 onto the rotor laminations without crushing the body portion 114. Prior art cans having no such end wall provide significantly less surface area to which force can be applied during assembly. The resultant high force per square inch of such cans can easily cause crumpling or other destruction of the can. With the end wall 117, force can be applied over a greater surface area to reduce the amount of force per square inch, and, consequently, the likelihood that the can will be damaged during

assembly. It also allows for the can 112 to be made with thinner walls, which reduces the motor air gap 88 and improves motor performance.

As discussed above, fluid flow through the pump 10 can cause slight pressure differentials within the pump 10. Such pressure differentials will be experienced in the forward and rearward motor chambers 107, 109. Both ends of the motor rotor 42 are subjected to pressures within the respective chambers 107, 109. This causes a thrust load that reduces the load applied to the pump rotor 28 and the motor bearing 52 when the pump 10 is mounted in a vertical position with the pump head below the motor.

The pump 10 enables the stator barrier can 56 to be assembled to the entire pump head 22 and motor rotor 42 by clamping the assembly 160 together with the front motor end shield 64 as shown in FIG. 12. The assembly 160 can be tested individually and treated for use in food applications without having to be fully assembled to the motor 14. This is also beneficial from a logistics standpoint because it allows for the production of pump head assemblies 160 ahead of schedule and the assemblies 160 can be stocked without committing to a particular motor type. It also allows for greater flexibility in the assembly methods employed, and it helps to reduce inventory and more easily meet customer demands.

It will be appreciated that a canned motor pump of the above described configuration is not limited to a rotary vane pump head. FIGS. 13 and 14 illustrate two alternate types of pump heads that may be employed in place of a rotary vane pump head. FIG. 13 shows a closed shroud centrifical pump head 170 driven by the motor 14, and FIG. 14 shows a turbine pump head 172 driven by the motor 14. Flow of fluid out of the pump heads 170, 172 through bearing discharge passages 84a, 84b, between the stator barrier can 56 and canned rotor 42, through motor bearing passages 102a, 102b, and returning to the pump heads 170, 172 by way of a hollow motor shaft 44 is the same as that described above with reference to a rotary vane pump head.

The foregoing description details one or more preferred embodiments of the present invention and describes the best mode contemplated. It will be appreciated, however, that no matter how detailed the foregoing description appears, the invention can be practiced in many ways without departing from the spirit of the invention. Therefore, the above mentioned description is to be considered exemplary rather than limiting.

Claims

What is claimed is:

1. A fluid pump comprising:

a pump head for receiving a fluid at a first pressure and pressurizing the fluid to a second pressure greater than said first pressure, said pump head including:

an inlet for receiving fluid at said first pressure;

5 an outlet for outputting fluid at said second pressure;

a first passage for directing fluid at said second pressure to said outlet without the pressurized fluid leaving the pump head prior to reaching the outlet; and

10 a second passage for directing fluid at said second pressure to a fluid circulation path at least a portion of which is external to the pump head with the fluid circulation path terminating adjacent the outlet; and

an electric motor for driving the pump head to increase the pressure of the fluid to said second pressure;

wherein pressurized fluid passing through said first and second passages is directed to the outlet without being re-pressurized by the pump head.

2. The pump of claim 1 wherein said electric motor includes:

a stator with one or more magnetic field producing elements for producing a magnetic field in response to a flow of electrical current through the one or more magnetic field producing elements;

5 a rotor having a rotor shaft for driving the pump head in response to the magnetic field produced by the stator; and

an air gap between the stator and the rotor, the air gap in flow communication with pressurized fluid from the pump head; and

wherein said fluid circulation path includes at least a portion of the air gap.

3. The pump of claim 2 wherein said rotor shaft is axially movable with respect to pressure generating elements within the pump head such that the rotor shaft is free to move in an axial direction independently of said pressure generating elements.

4. The pump of claim 2 wherein said rotor shaft is hollow and configured to receive pressurized fluid, wherein said fluid circulation path includes the hollow rotor shaft.

5. The pump of claim 1 wherein said pump head further includes:

a relief valve cavity having a first portion in fluid communication with said inlet and a second portion in fluid communication with said outlet; and

5 a relief valve disposed in said relief valve cavity with no portion of the relief valve positioned in the inlet or the outlet, said relief valve configured to include:

a low pressure relief valve mode wherein fluid flows from said inlet to said outlet when fluid pressure at the inlet exceeds fluid pressure at the outlet by a first threshold amount; and

10 a high pressure relief valve mode wherein fluid flows from said outlet to said inlet when fluid pressure at the outlet exceeds fluid pressure at the inlet by a second threshold amount greater than said first threshold amount.

6. The pump of claim 1 wherein said motor further includes a liner positioned in said air gap to inhibit pressurized fluid which enters the air gap from contacting the one or more magnetic field producing elements of the stator, said rotor further including a rotor barrier can to prevent pressurized fluid entering the air gap from contacting the rotor laminations.

7. The pump of claim 6 wherein said liner includes a thin liner wall that deforms during operation of the apparatus to bear against said one or more magnetic field producing elements for structural support as a result of loading imparted by the pressurized fluid on the liner wall.

8. The pump of claim 1 wherein said first passage is in opposed relation to said second passage so as to provide balancing of thrust produced by flow of pressurized fluid through the first and second passages.

9. The pump of claim 1 wherein said pump head further includes:

a front bearing having said first passage disposed therein;

a rear bearing adjacent said electric motor and having said second passage disposed therein;

5 a pump rotor with a plurality of sliding vanes and disposed between said front and rear bearings; and

a cam ring disposed about said pump rotor and sliding vanes, said cam ring having an inner surface along which the sliding vanes are moved by the rotor to produce fluid pressurized to said second pressure;

10 wherein fluid pressurized by movement of the sliding vanes is directed through said first and second passages of the bearings.

10. The pump of claim 1 wherein said pump head is a rotary vane pump head.

11. The pump of claim 1 wherein said pump head is a turbine pump head.

12. The pump of claim 1 wherein said pump head is a centrifical pump head.

13. An apparatus for detecting the presence of an overheat condition in a fluid pump, the apparatus comprising:

a pump head for receiving a fluid at a first pressure and outputting the fluid at a second pressure that is greater than said first pressure;

5 a motor positioned adjacent said pump head for driving the pump head to pressurize the fluid; and

an overheat sensor operatively associated with both the pump head and the motor for sensing overheat conditions.

14. The apparatus of claim 1 wherein the overheat sensor is located within the motor.

15. The apparatus of claim 1 wherein said overheat sensor is further operable to prevent operation of the motor when an overheat condition is sensed.

16. The apparatus of claim 1 wherein said overheat sensor includes a bi-metal switch formed in the stator windings of the motor, said bi-metal switch including:

a closed position providing electrical continuity through the stator windings during normal operation of the motor; and

5 an open position preventing electrical continuity through the stator windings when an overheat condition is sensed by the bi-metal switch.

17. The apparatus of claim 1, further comprising a switch for preventing operation of the motor when an overheat condition is sensed.

18. The apparatus of claim 1 wherein said overheat sensor is configured to detect the presence of an overheat condition when a threshold temperature of about 320 degrees F. is sensed by the overheat sensor.

19. The apparatus of claim 1 wherein said pump head is a rotary vane pump head.

20. The apparatus of claim 1 wherein said pump head is a turbine pump head.

21. The apparatus of claim 1 wherein said pump head is a centrifical pump head.

22. A rotor for an electric motor, said rotor comprising:
a shaft;

5 laminations attached to said shaft such that a first shaft portion extends beyond one end of the laminations and a second shaft portion extends beyond an opposed end of the laminations;
and

a fluid impermeable barrier for protecting the rotor laminations, said barrier including no more than two parts assembled about the laminations using no more than three areas of attachment including:

10 a first attachment area for attaching a first barrier part to a second barrier part for preventing fluid from entering the barrier at the first attachment area;

a second attachment area for attaching the first barrier part to the first shaft portion for preventing fluid from entering the barrier at the second attachment area; and

a third attachment area for attaching the second barrier part to the second shaft portion for preventing fluid from entering the barrier at the third attachment area.

23. The rotor of claim 1, further including one or more spacers interconnecting the laminations and the barrier.

24. The rotor of claim 1 wherein said barrier is assembled by welding.

25. The rotor of claim 1 wherein said barrier is assembled by pressure fit.

26. The rotor of claim 1 wherein said barrier is assembled by adhesive.

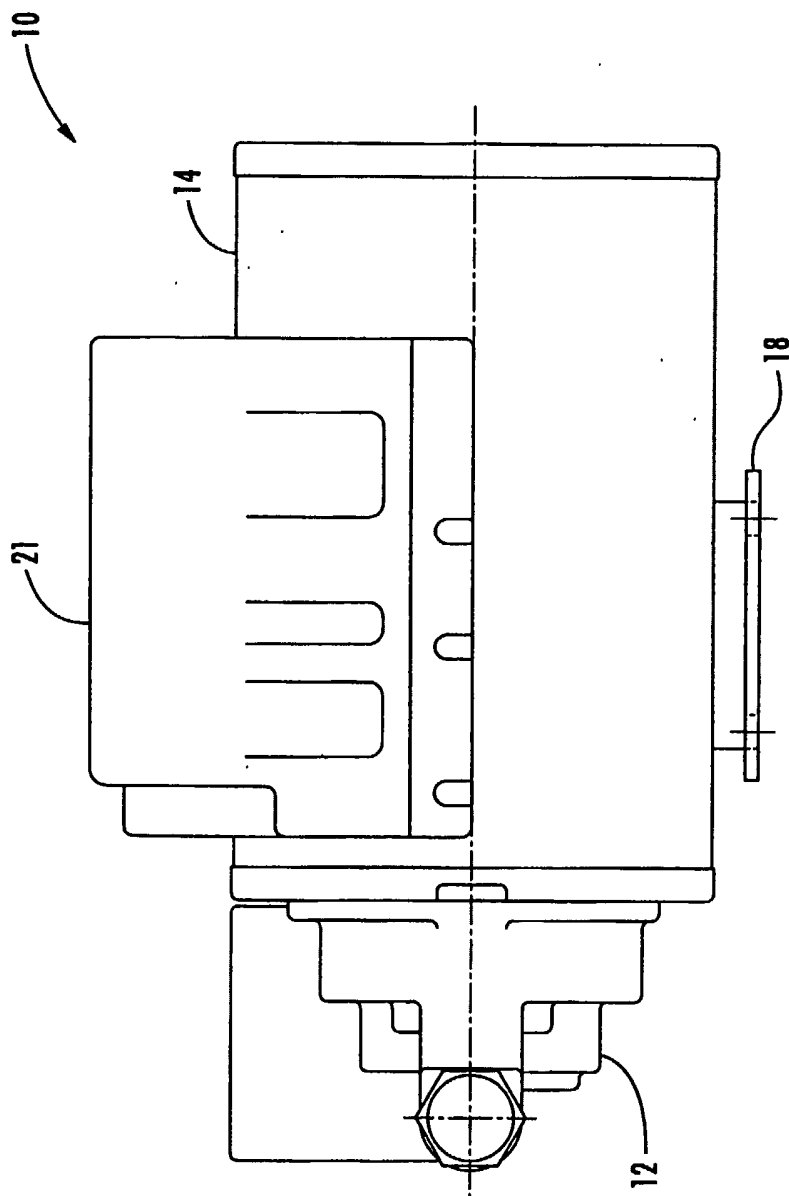
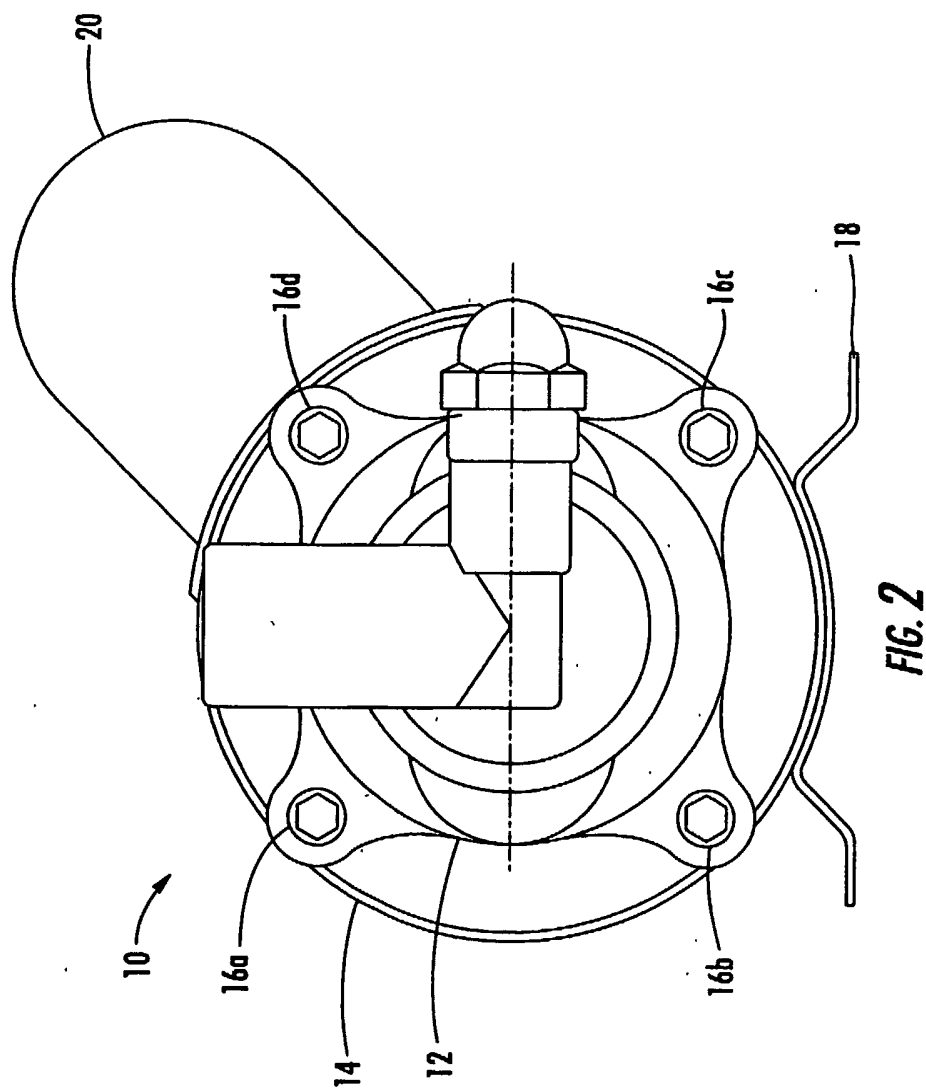


FIG. 7



3/14

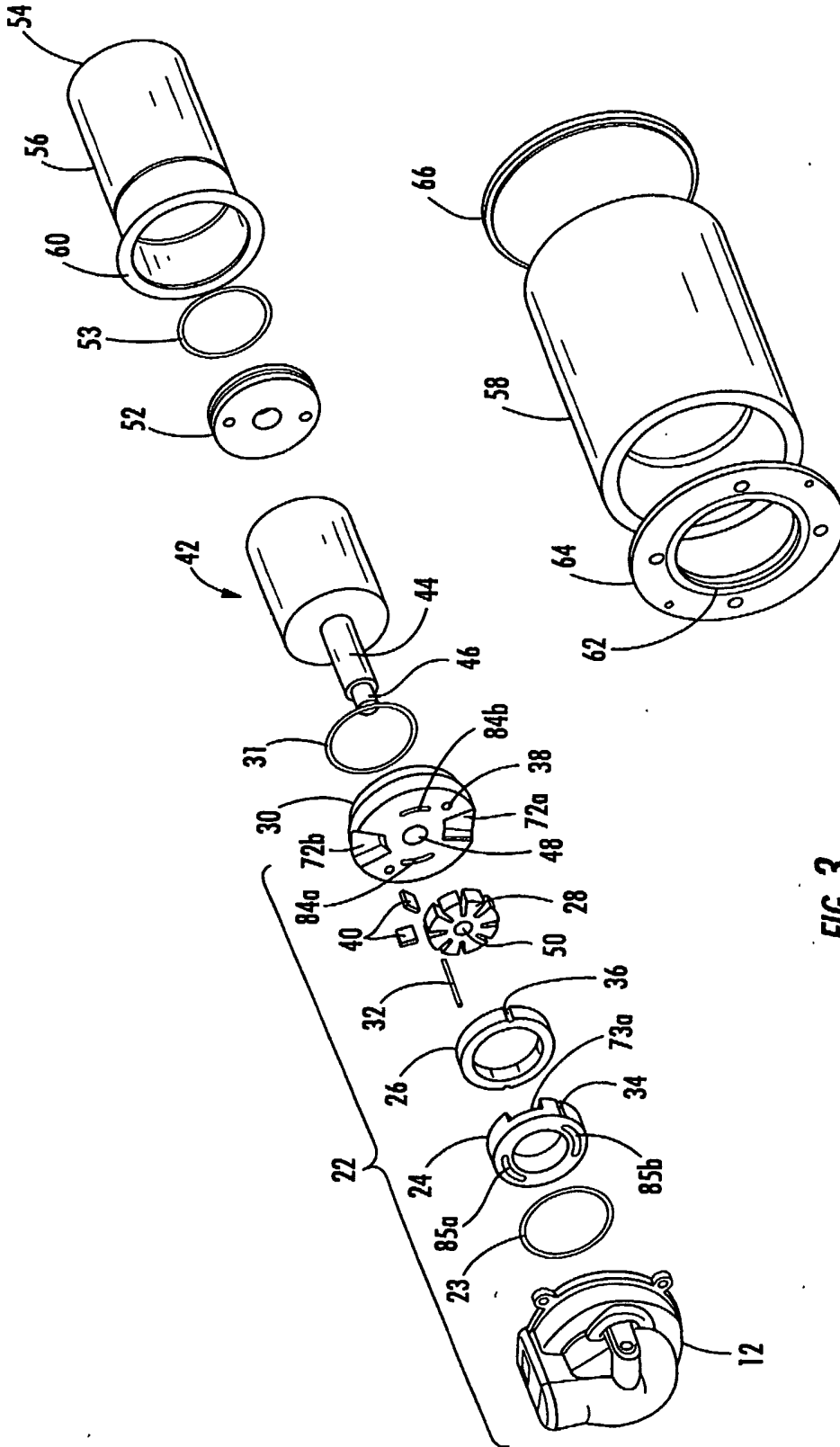


FIG. 3

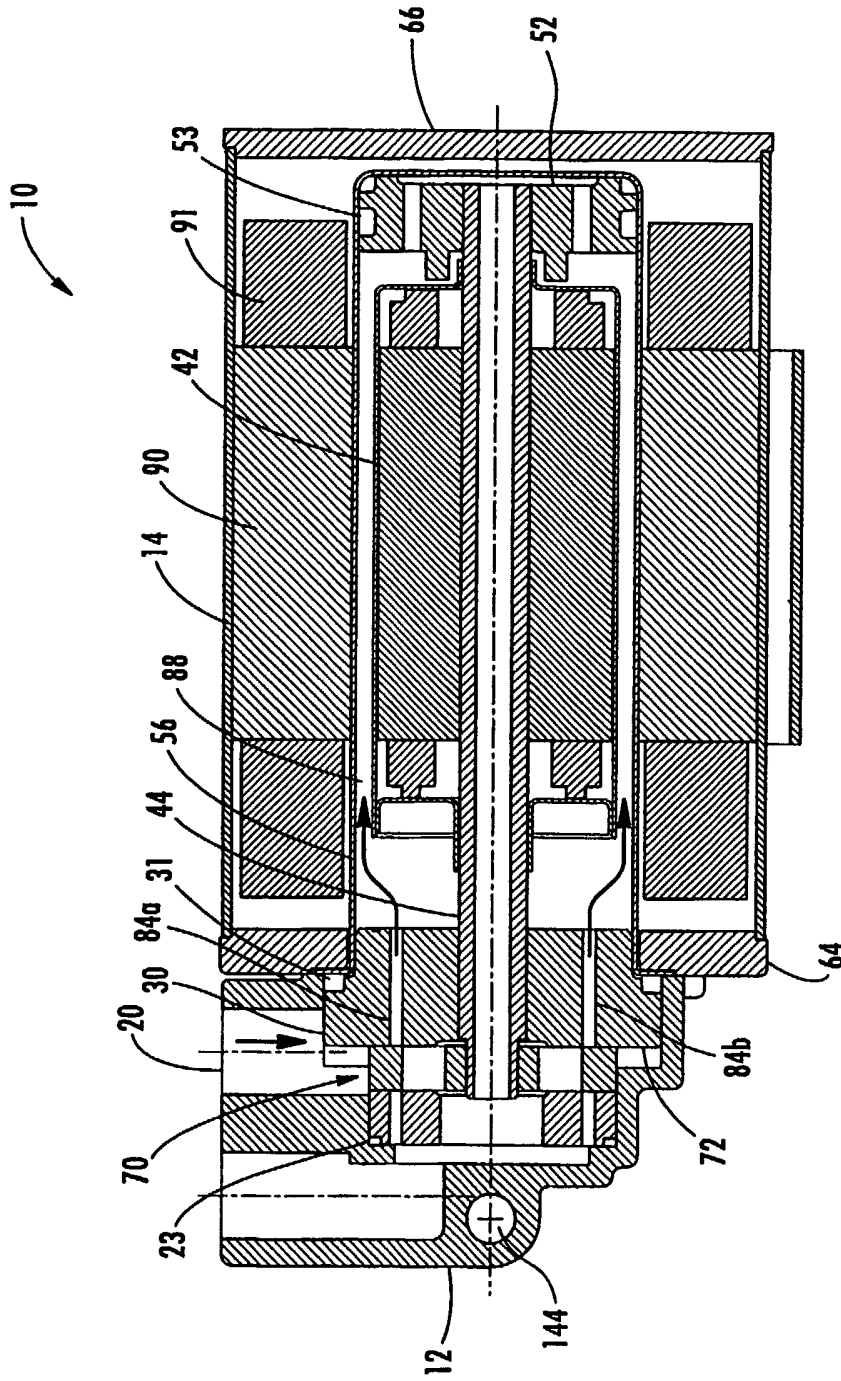


FIG. 4

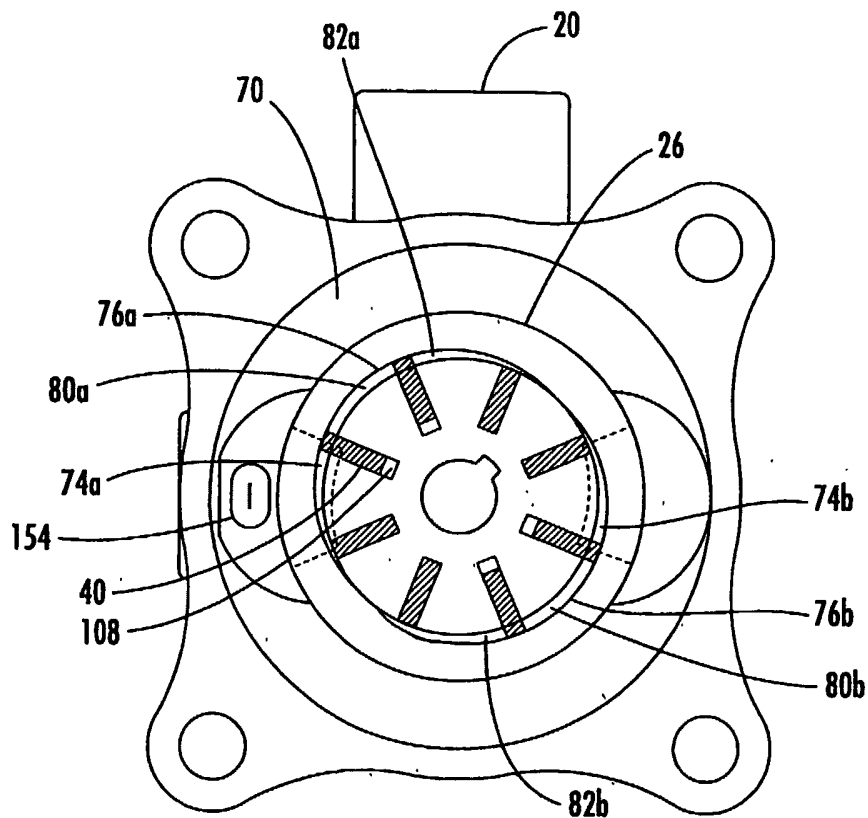


FIG. 5

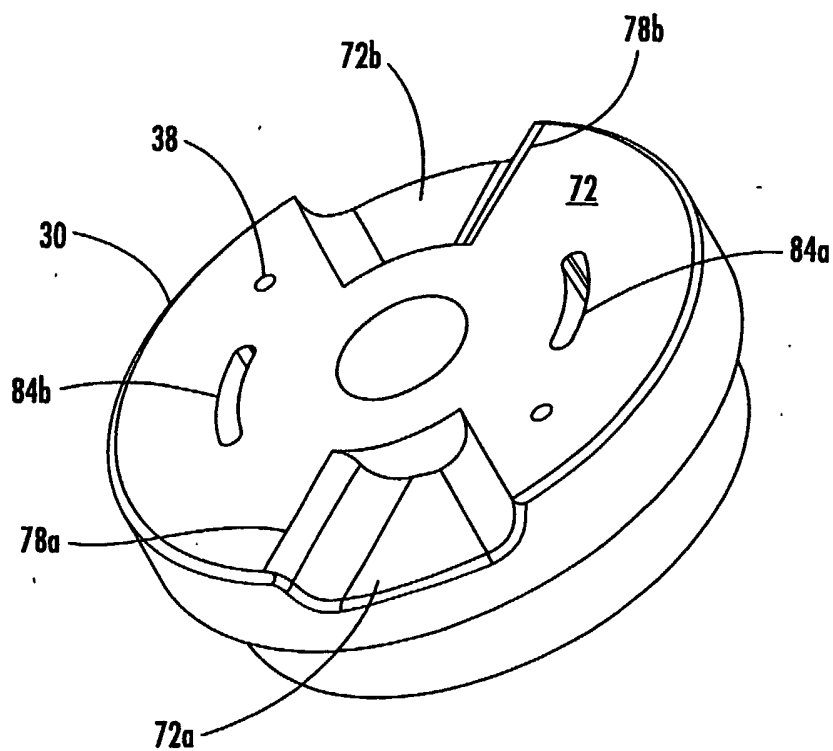


FIG. 6

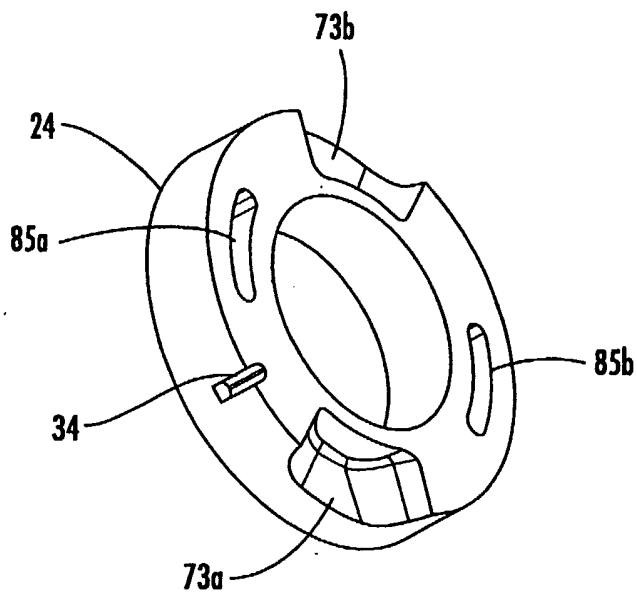


FIG. 7

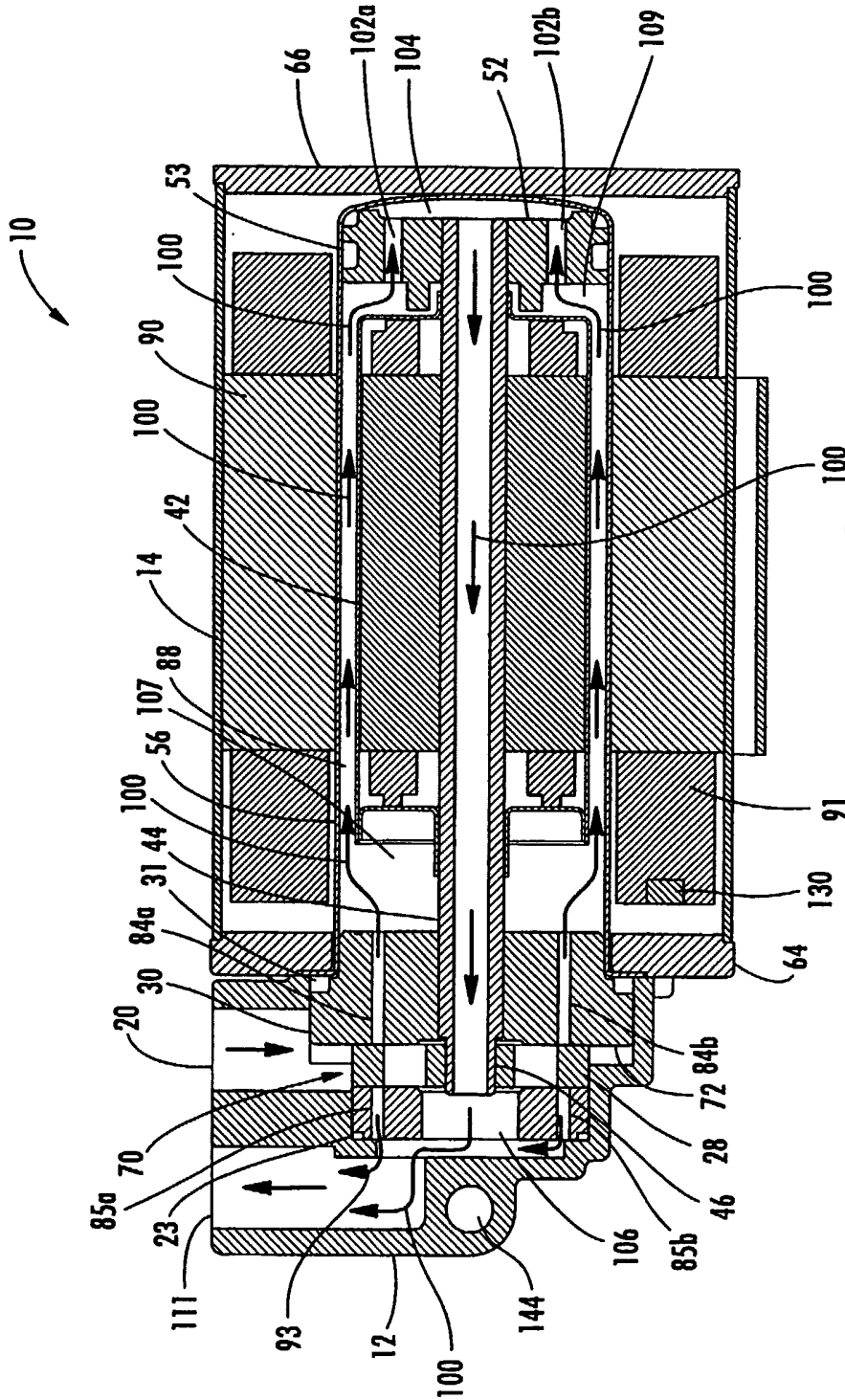


FIG. 8

9/14

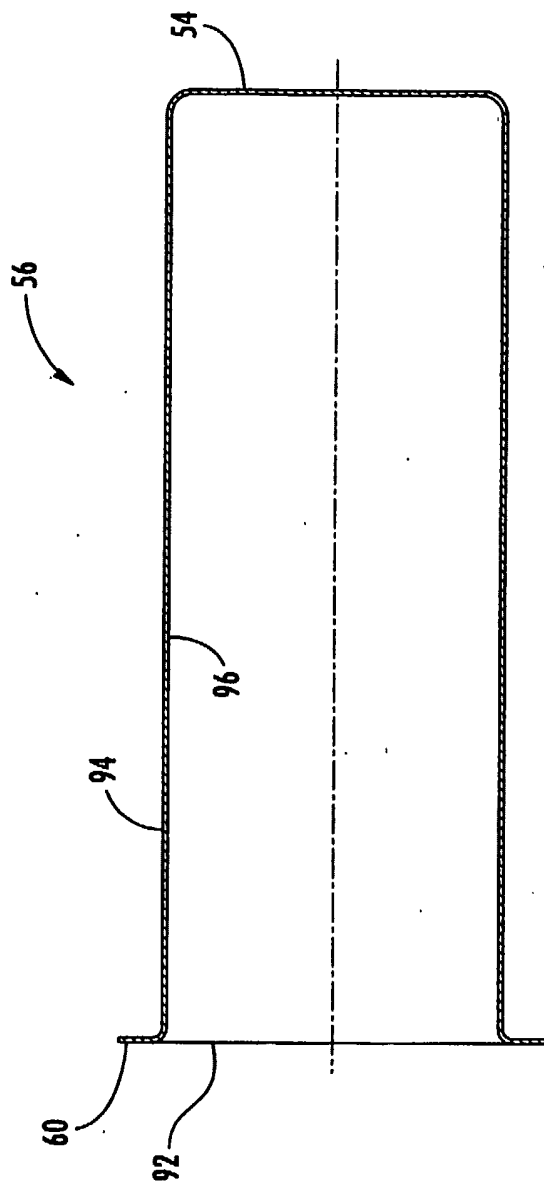


FIG. 9

10/14

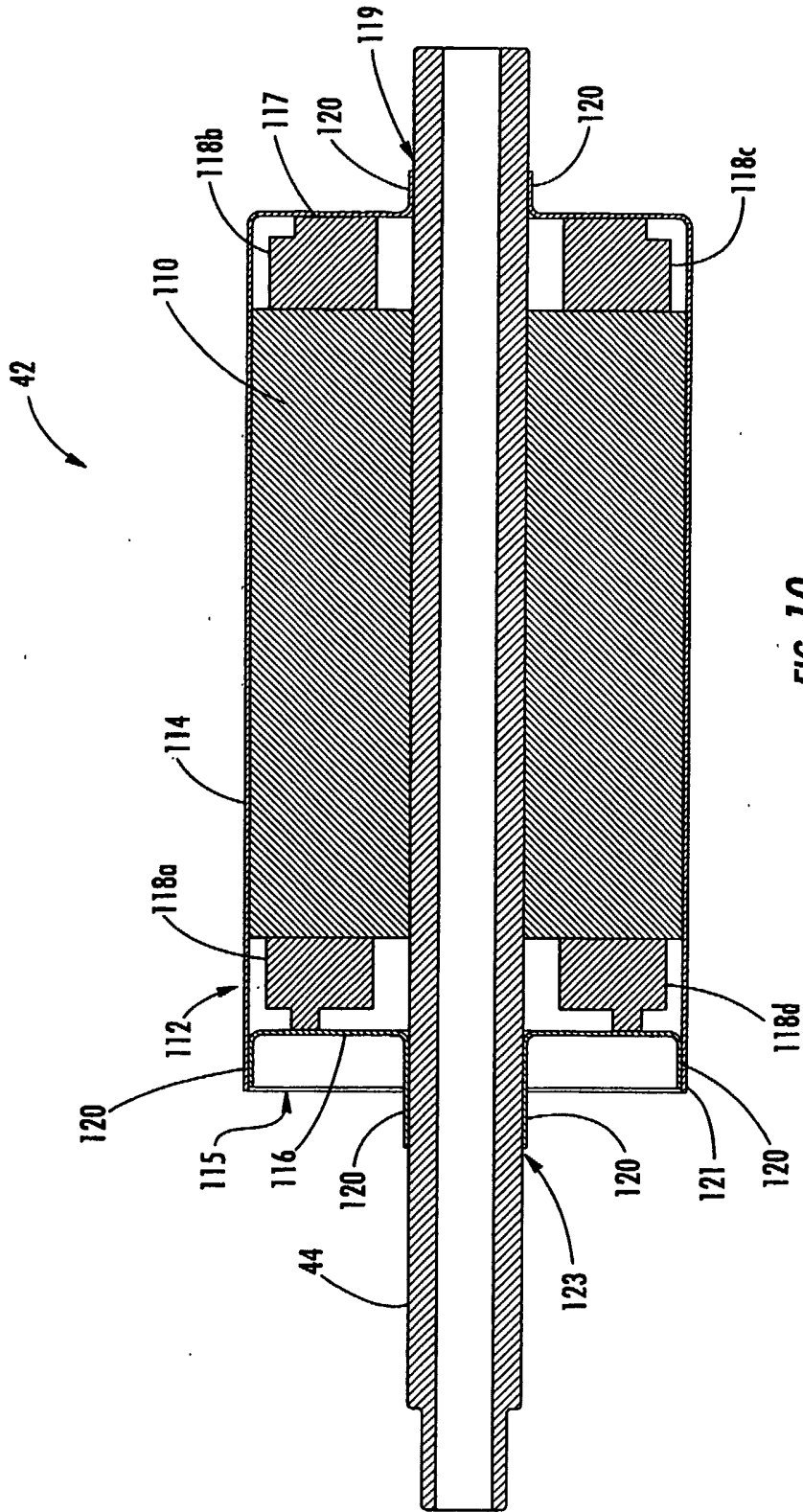


FIG. 10

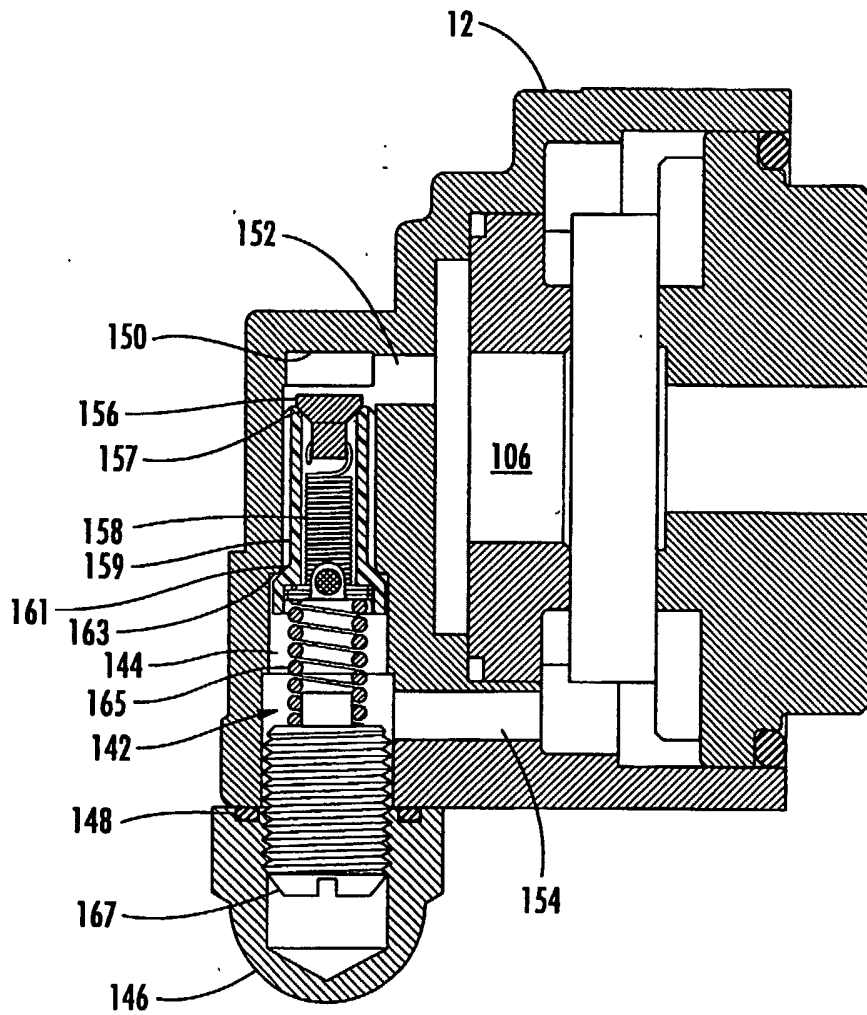


FIG. 11

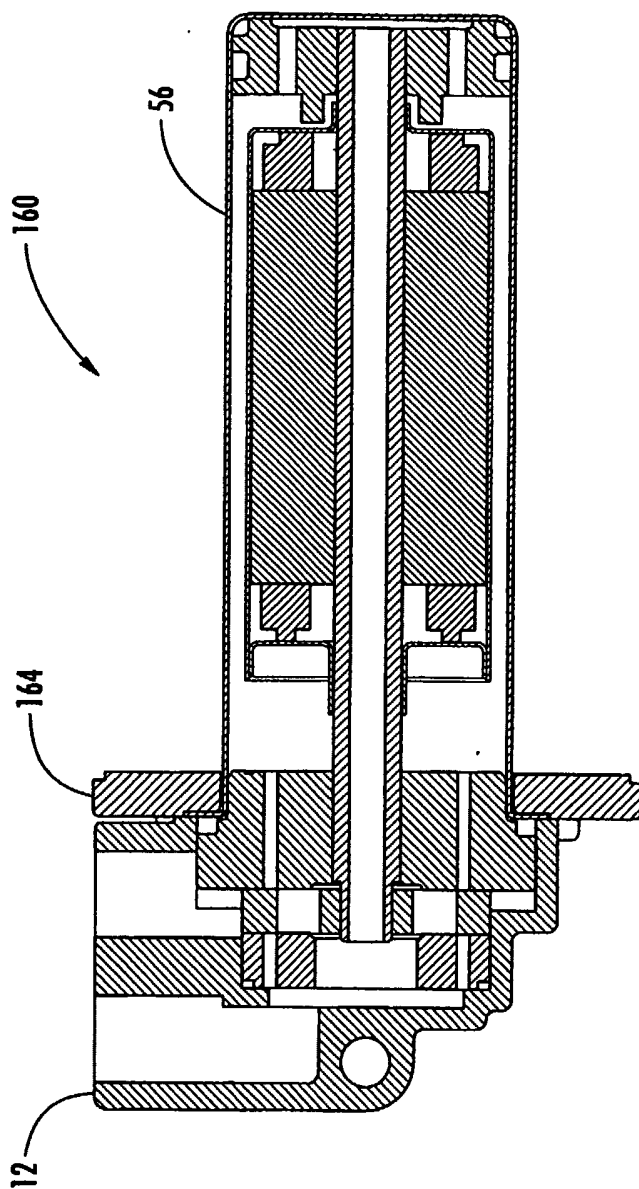


FIG. 12

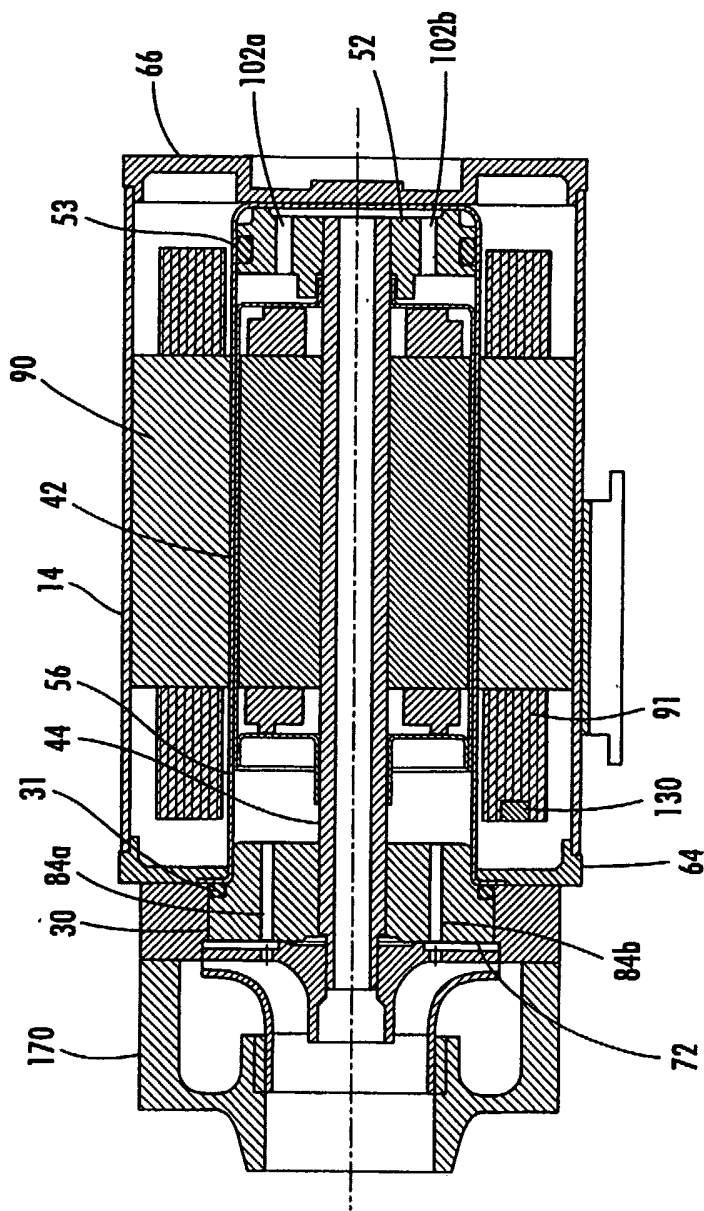


FIG. 13

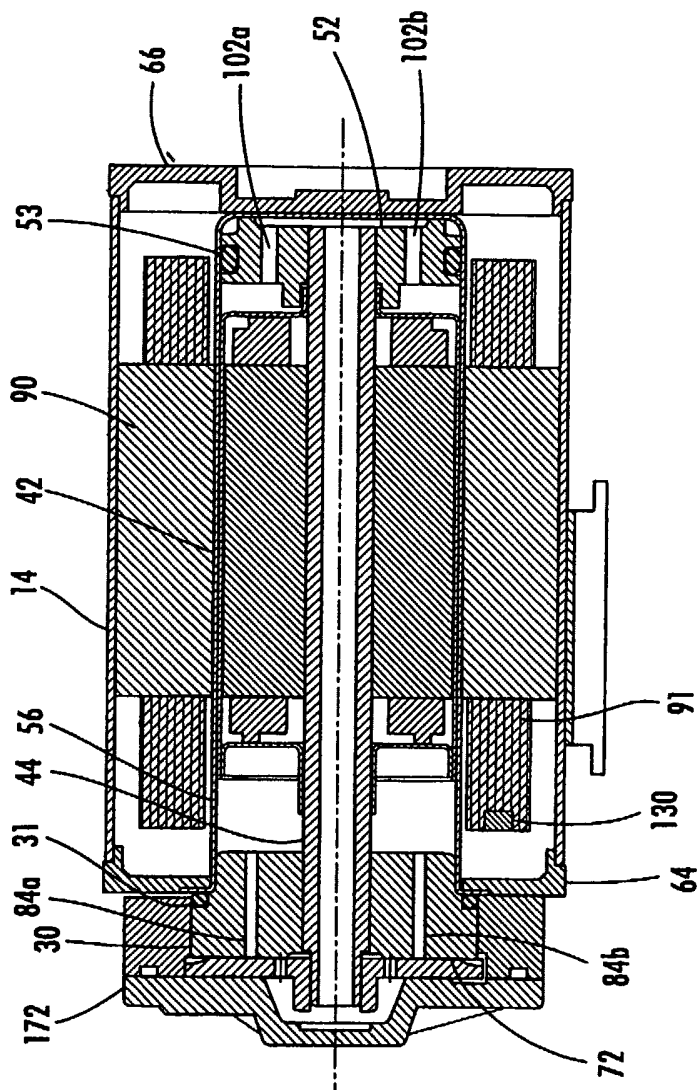


FIG. 14