US 20150030490A1

(19) **United States**

(12) **Patent Application Publication**

(10) **Pub. No.: US 2015/0030490 A1**

(43) **Pub. Date: Jan. 29, 2015**

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(54) **BEARING HOUSING AND ASSEMBLY OF A SCREW COMPRESSOR**

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(21) Appl. No.: **14/513,009**

(22) Filed: **Oct. 13, 2014**

Related U.S. Application Data

(63) Continuation-in-part of application No. 12/840,018, filed on Jul. 20, 2010.

(60) Provisional application No. 61/890,180, filed on Oct. 12, 2013.

Publication Classification

(51) **Int. Cl.**

F04C 18/16	(2006.01)
F01C 21/02	(2006.01)
F04C 28/08	(2006.01)

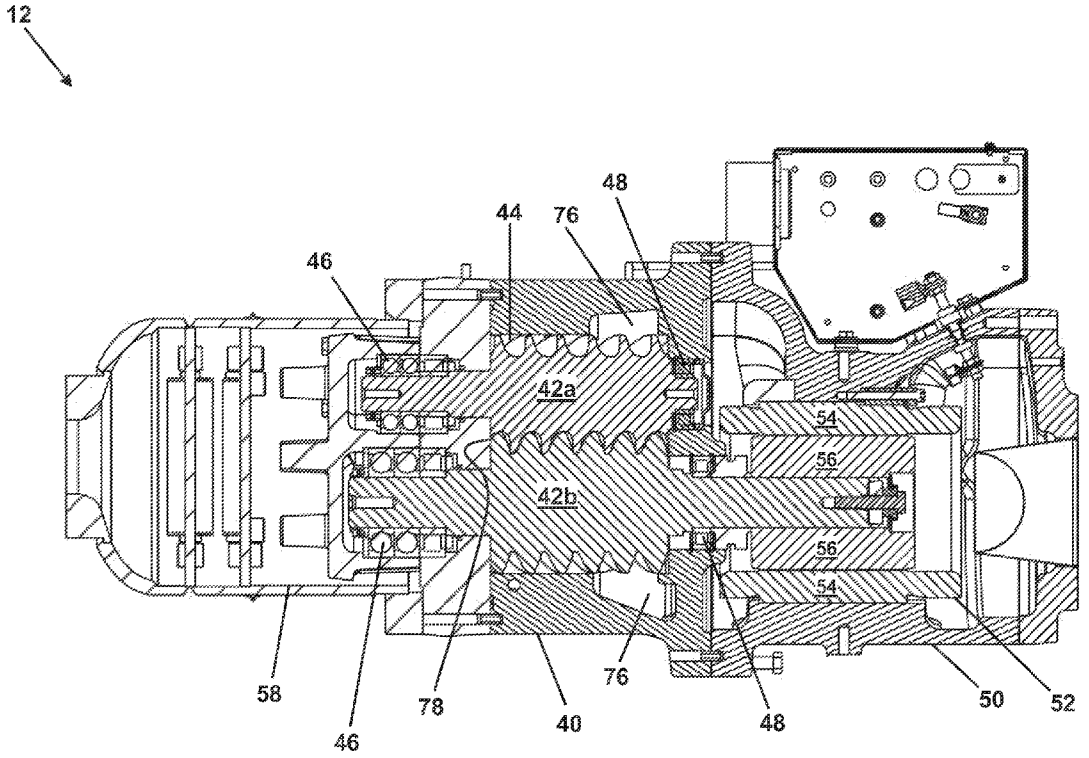
(52) **U.S. Cl.**

CPC ***F04C 18/16*** (2013.01); ***F04C 28/08*** (2013.01); ***F01C 21/02*** (2013.01)

USPC **418/201.1**

(57) **ABSTRACT**

An improved bearing housing of a rotary screw compressor is described. The bearing housing is generally shorter than a convention bearing housing. The bearing housing can be configured to enclose and support radial bearings of the screw compressor. The bearing housing can be configured not to enclose axial bearings of the screw compressor in an axial direction.



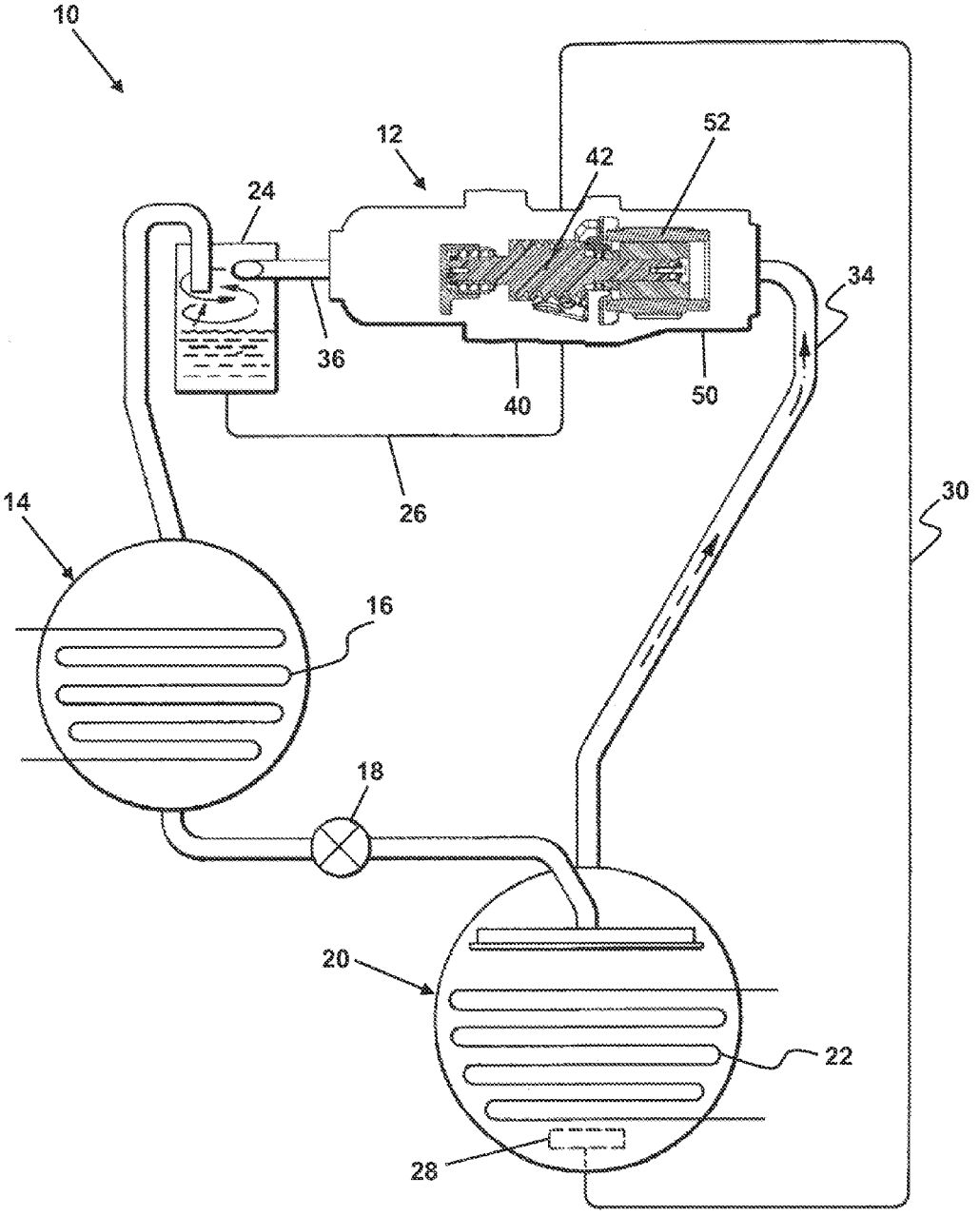


FIG. 1

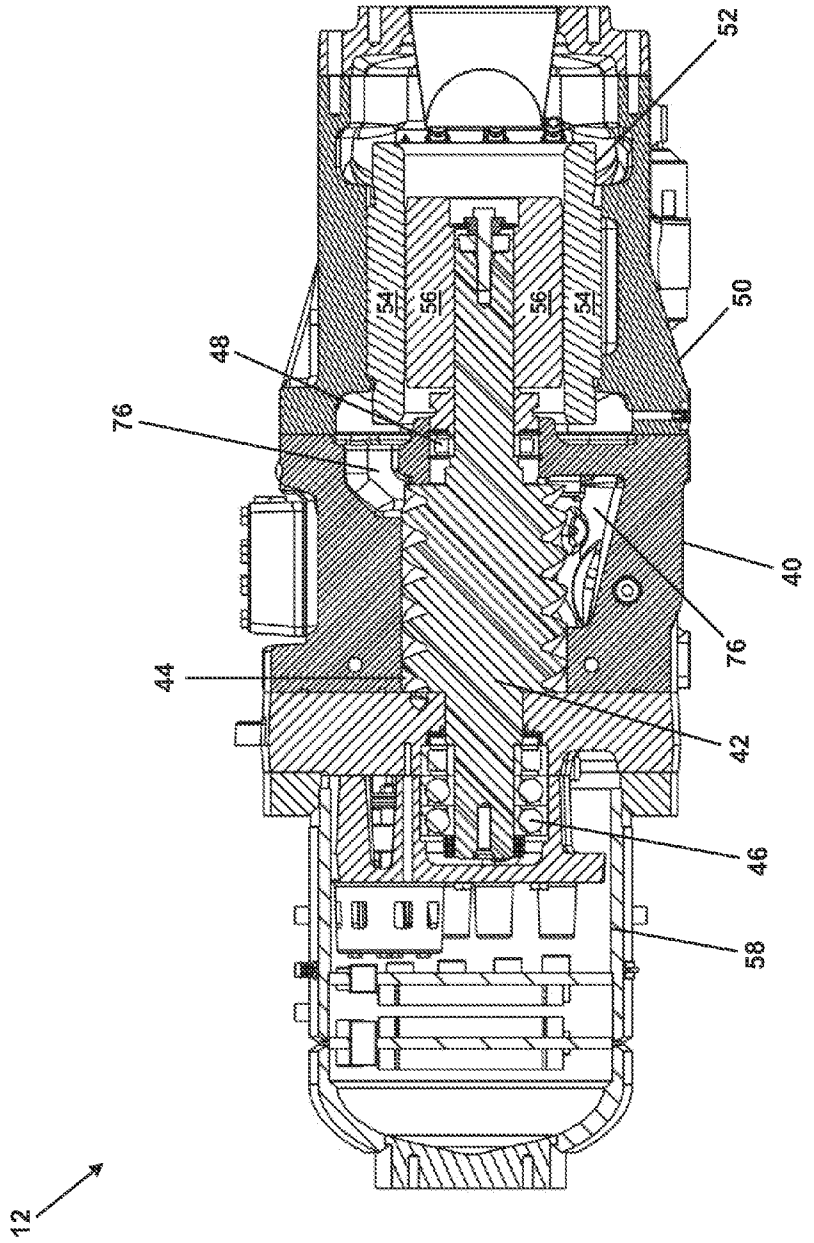


FIG. 2

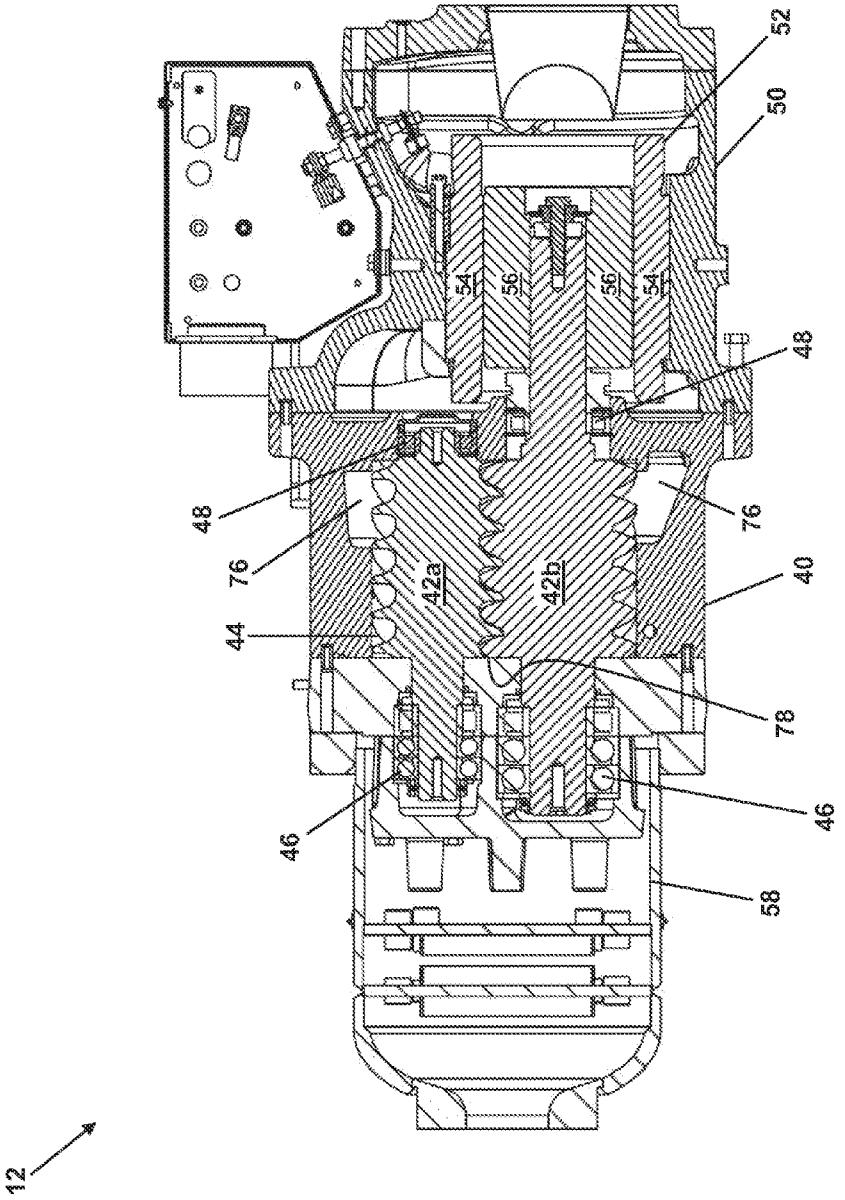


FIG. 3

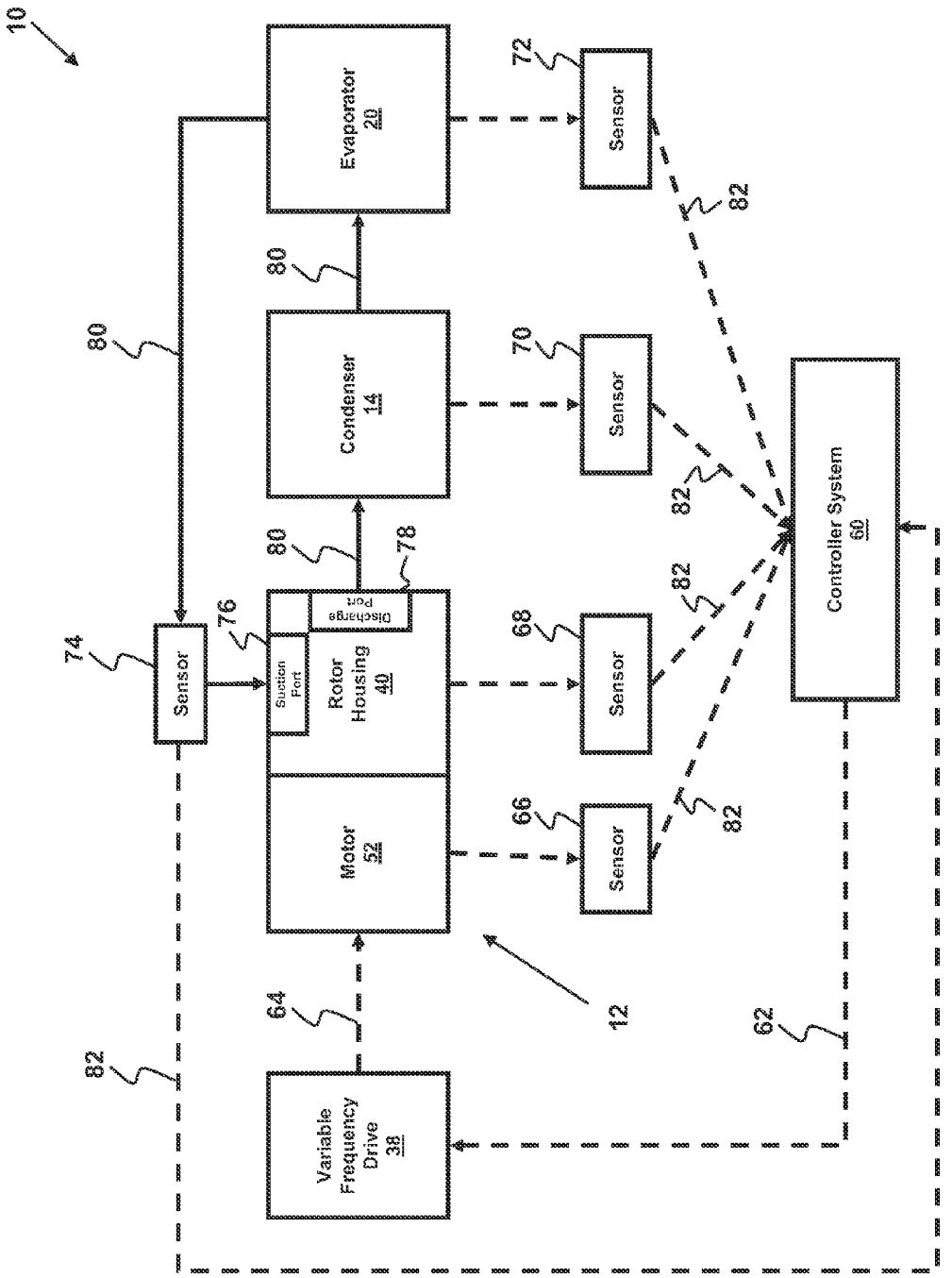

10
Evaporator 20
Condenser 14
Rotor Housing 40
Suction Port
Discharge Port
Motor 52
Variable Frequency Drive 38
Sensor
Controller System 60
12
62
64
66
68
70
72
74
76
78
80
82

FIG. 4

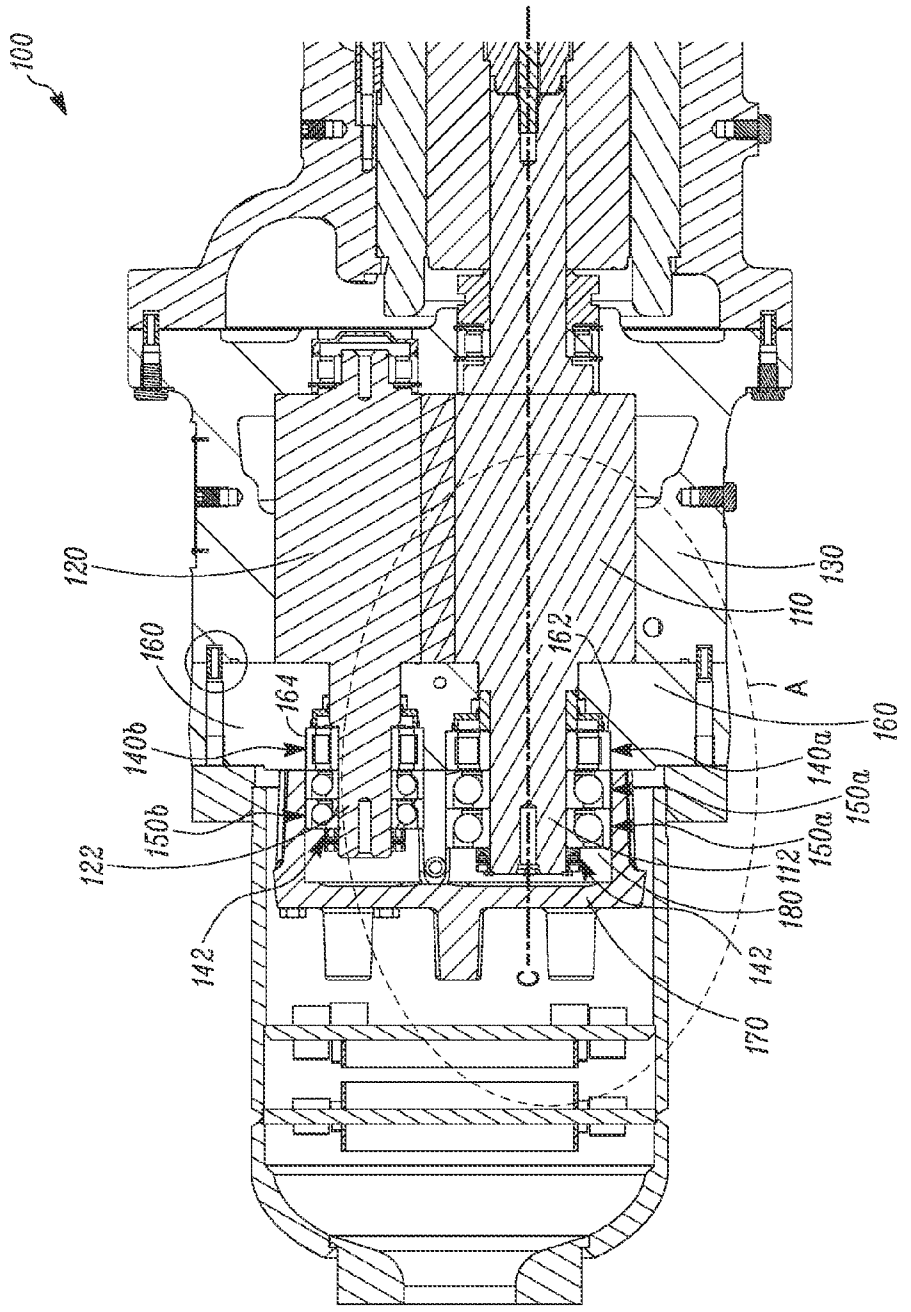

100
120
130
110
160
162
164
140b
140a
150b
150a
122
112
180
142
170
A
C

FIG.5

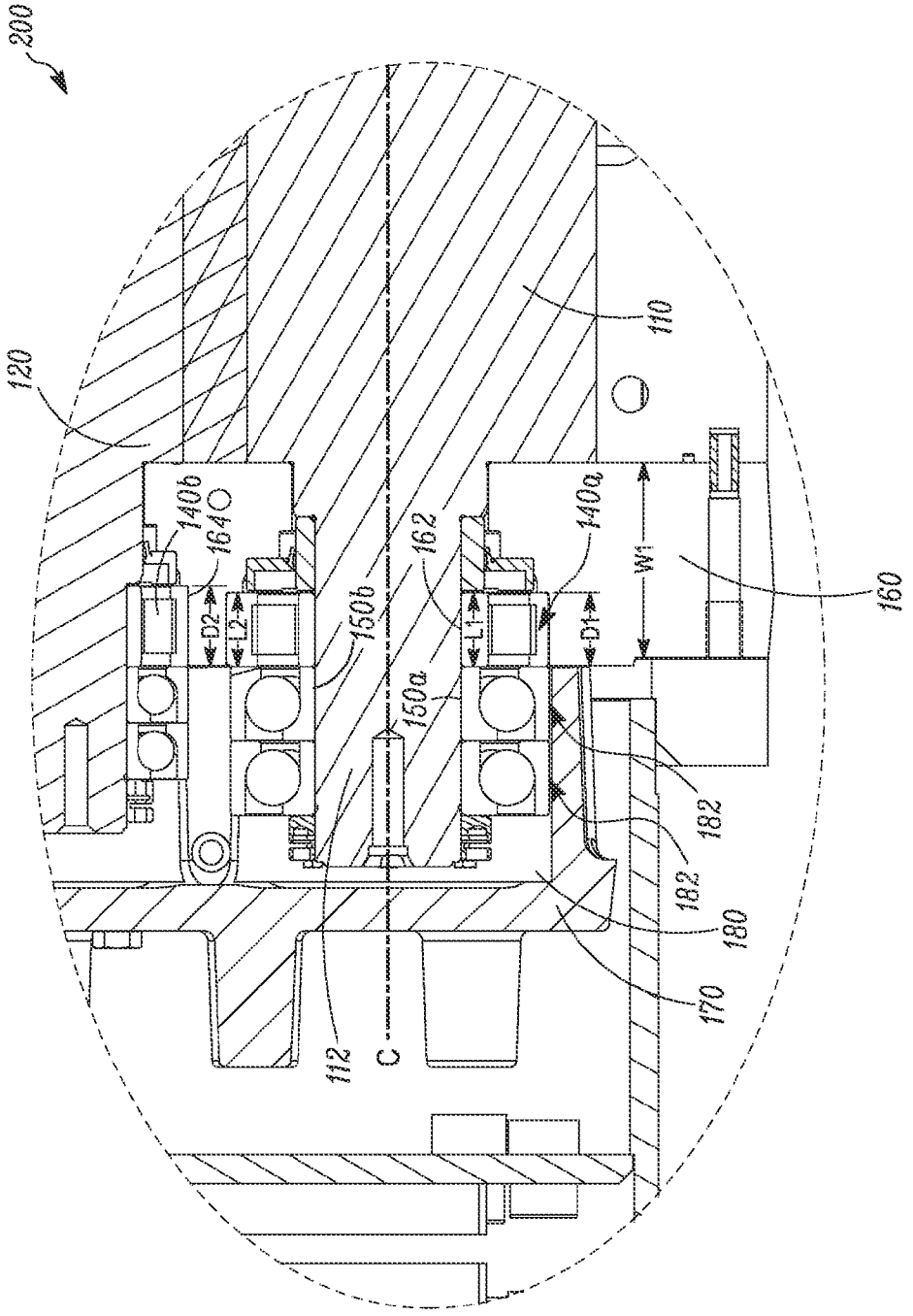

200
120
110
140b
164
162
140a
W1
160
D2
L2
150b
L1
D1
150a
182
182
180
170
112
C

FIG. 6

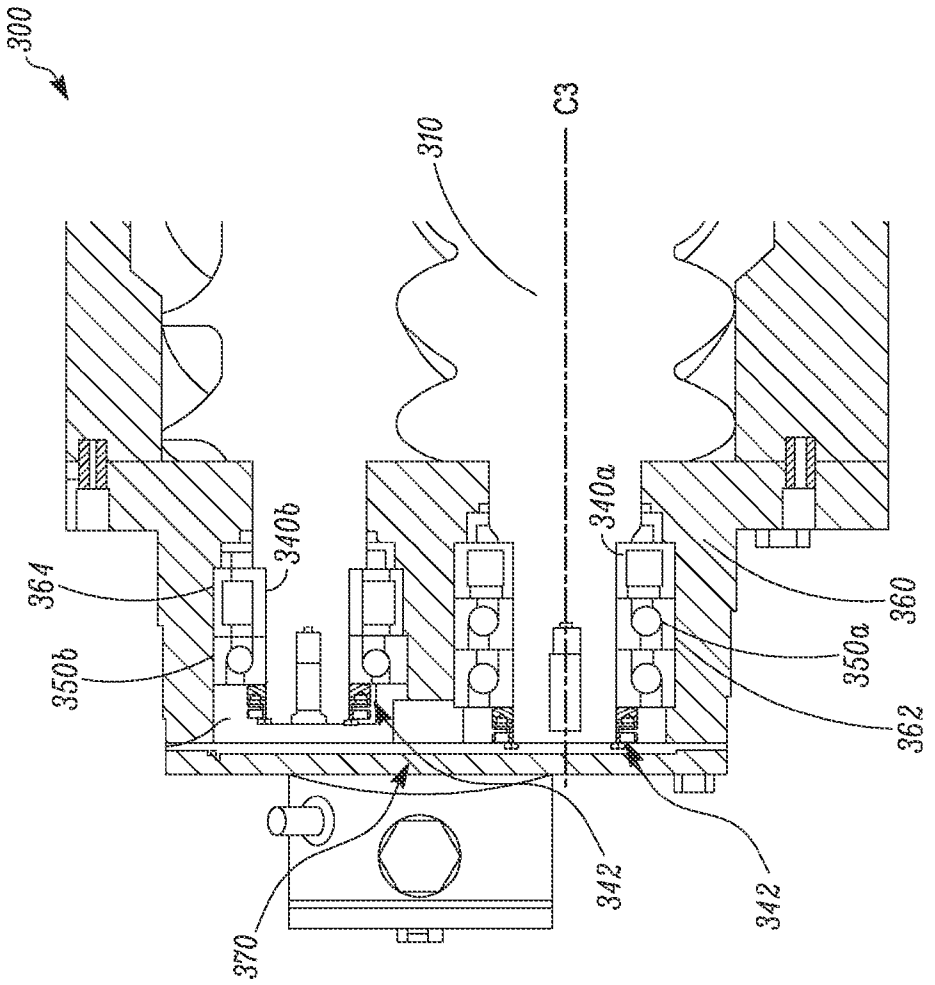


FIG. 7 (PRIOR ART)

BEARING HOUSING AND ASSEMBLY OF A SCREW COMPRESSOR

FIELD

[0001] The disclosure herein relates to a rotary type compressor, such as a rotary screw compressor, which can be used in, for example, a heating, ventilation, and air-conditioning ("HVAC") system. More specifically, the disclosure relates to a bearing housing of a rotary screw compressor to support and enclose discharge axial bearings. The bearing housing herein can improve location of the rotors which may result in improved compressor performance and reliability, and machining capability of the housing may also be improved.

BACKGROUND

[0002] A screw compressor is a type of positive displacement compressor that can be used to compress various working fluids, such as for example refrigerant vapor. Such screw compressors may be used in refrigeration units, such as for example, water chillers as part of a HVAC system. The screw compressor typically includes one or more rotors that rotate relative to bearings such as for example radial and axial bearings at the discharge end. A bearing housing and cover are often part of the assembly of the screw compressor to enclose and support the bearings, e.g. radial and axial bearings. During operation, the bearing cover can have a discharge outlet or port so that a compressed working fluid (e.g. refrigerant vapor) can be discharged from an axial end of the rotors and out of the bearing housing and cover.

SUMMARY

[0003] According to an embodiment of the present invention, a variable capacity screw compressor comprises a rotor housing, a motor, and a variable speed drive. The rotor housing comprises a suction port, a working chamber, a discharge port, and at least two screw rotors that comprise a female screw rotor and a male screw rotor being positioned within the working chamber for cooperatively compressing a fluid. The suction port, the at least two screw rotors and the discharge port are configured in relation to a selected rotational speed. The selected rotational speed operates at least one screw rotor at an optimum peripheral velocity that is independent of a peripheral velocity of the at least one screw rotor at a synchronous motor rotational speed for a rated screw compressor capacity. A motor is operable to drive the at least one screw rotor at a rotational speed at a full-load capacity that is substantially greater than the synchronous motor rotational speed at the rated screw compressor capacity. A variable speed drive receives a command signal from a controller and generates a control signal that drives the motor at the rotational speed.

[0004] In another embodiment, a method for sizing at least two screw compressors is provided. The target capacity for each screw compressor is selected. Each screw compressor has a different rated capacity and further comprises a suction port, a working chamber, a discharge port, and at least two screw rotors being positioned within the working chamber for cooperatively compressing a fluid. The rotational speed is selected to operate at least one screw rotor in each screw compressor at an approximately constant optimum peripheral velocity that is independent of the rated capacity of each screw compressor. The suction port, the at least two screw rotors and the discharge port are configured together with the rotational speed for each screw compressor.

[0005] In another embodiment, a refrigeration chiller, having at least one refrigeration circuit, comprises a variable capacity screw compressor, condenser, expansion valve and evaporator. The variable capacity compressor comprises a rotor housing, a motor housing and a variable speed drive. The rotor housing further comprises a suction port, a working chamber, a discharge port, and at least two screw rotors that comprise a female screw rotor and a male screw rotor being positioned within the working chamber for cooperatively compressing a fluid. The suction port, the at least two screw rotors and the discharge port are configured in relation to a selected rotational speed. The selected rotational speed provides at least one screw rotor to operate at an optimum peripheral velocity that is independent of a peripheral velocity of the at least one screw rotor at a synchronous motor rotational speed for a rated screw compressor capacity. The motor housing further comprises a motor, the motor is operable to drive the at least one screw rotor at a rotational speed at a full-load capacity that is substantially greater than the synchronous motor rotational speed at the rated screw compressor capacity. The variable speed drive is configured to receive a command signal from a controller and to generate a control signal that drives the motor at the rotational speed. A condenser is coupled to the discharge port of the variable capacity screw compressor. The condenser is configured to cool and condense fluid received from the discharge port. An expansion valve is coupled to the condenser. The expansion valve is configured to evaporate at least a portion of fluid received from the condenser by lowering pressure of fluid received from the condenser. An evaporator is coupled to the expansion valve. The evaporator is configured to evaporate fluid received from the expansion valve and to provide fluid to the suction port of the variable capacity screw compressor.

[0006] An improved bearing housing of a rotary screw compressor is described. A bearing housing is generally configured to suitably enclose and support discharge radial bearings which are located at a discharge side of the compressor, for example toward the axial end of the rotors.

[0007] In previously known designs, the discharge bearing housing of a screw compressor is constructed to be a relatively long part, which encloses and/or supports the discharge radial bearings, the axial bearings, and the bearing retaining assembly for example the axial bearing retainers.

[0008] In the bearing housing shown and described herein, a shorter bearing housing suitably encloses and/or supports the discharge radial bearings, but does not enclose or support the axial bearings and retaining assembly, e.g. the axial bearing retainers. A bearing cover is provided which encloses the axial bearings and the axial bearing retainers. The shorter bearing housing may be simpler to fabricate and the accuracy of the discharge axial bearing bores may be improved due to shorter reaches and shorter machine tool boring bars, compared to a convention bearing housing. Due to the new bearing housing design, the bearing cover may be fabricated relatively easier using machine tools that may not be as precise as those used to fabricate the bearing housing. That is, the design of the bearing housing herein can improve for example the machining capability of the housing, such as for example by enabling short machine cutter tooling and short reaches for the machining center of the housing. The shorter bearing housing can improve location of the rotors such as during assembly, such as for example by improving the accuracy of the discharge bearing bores, which may result in improved compressor performance and reliability.

[0009] In one embodiment, a bearing assembly may include a bearing cover and a bearing housing. The bearing housing includes a cavity that is configured to enclose and/or support a discharge radial bearing. The cavity has a depth. The discharge radial bearing has a length. The depth of the cavity may be configured to be no more than the length of the discharge radial bearing so that the cavity can be configured to enclose and/or support the discharge radial bearing, but not the axial bearing.

[0010] Other features and aspects of the embodiments will become apparent by consideration of the following detailed description and accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0011] Reference is now made to the drawings in which like reference numbers represent corresponding parts throughout.

[0012] FIG. **1** illustrates an embodiment that incorporates a screw compressor arranged as part of a refrigeration chiller system.

[0013] FIG. **2** illustrates a cross sectional view of a screw compressor according to one embodiment.

[0014] FIG. **3** illustrates an additional cross sectional view of a screw compressor according to one embodiment.

[0015] FIG. **4** illustrates an embodiment of a refrigeration chiller and controller system according to one embodiment.

[0016] FIG. **5** illustrates a partial sectional view of a screw compressor, with which the embodiments as disclosed herein can be practiced.

[0017] FIG. **6** illustrates an enlarged sectional view of an area A of the screw compressor in FIG. **5**.

[0018] FIG. **7** illustrates a conventional bearing housing.

DETAILED DESCRIPTION

[0019] As a preface to the detailed description, as used in this specification and the appended claims, the singular forms "a," "an," and "the" also include plural referents, unless the context clearly dictates otherwise. References in this specification to "one embodiment," "an embodiment," "an example embodiment," etc., indicate that the described embodiment may include a particular feature, structure, or characteristic; however, every embodiment may not necessarily include the particular feature, structure, or characteristic. When a particular feature, structure, or characteristic is described in connection with an embodiment, other embodiments may incorporate or otherwise implement such feature, structure, or characteristic whether or not explicitly described.

[0020] Referring now to FIGS. **1-4**, components of a chiller or chiller system **10** are illustrated. Chiller **10** includes many other conventional features not depicted for simplicity of the drawings.

[0021] Chiller system **10** is directed to refrigeration systems. Chiller **10** is in the range of about 20 to 500 tons or larger, particularly where the refrigeration system includes a multiple stage compressor arrangement. Persons of ordinary skill in this art will readily understand that embodiments and features of this invention are contemplated to include and apply to, not only single stage compressors/chillers, but also to (i) multiple stage compressors/chillers and (ii) single and/or multistage compressor/chillers operated in parallel.

[0022] As shown, chiller **10** comprises a screw compressor system **12** (also sometimes referred to as a screw compressor **12**), a condenser **14**, and an evaporator **20**, all of which are serially connected to form a semi- or fully-hermetic, closed-loop refrigeration system. Chiller **10** may circulate a fluid **80** (such as, for example, a refrigerant) to control the temperature in a space such as a room, home, or building. The fluid **80** may be circulated to absorb and remove heat from the space and may subsequently reject the heat elsewhere.

[0023] Fluid **80** may be a refrigerant. The refrigerant may be selected from an azeotrope, a zeotrope or a mixture or blend thereof in gas, liquid or multiple phases. For example, such refrigerants may be selected from: R-123, R-134a, R-1234yf, R-410A, R-22 or R-32. Because embodiments of the present invention are not restricted to the refrigerant chosen, embodiments of the present invention are also adaptable to a wide variety of refrigerants that are emerging, such as low global warming potential (low-GWP) refrigerants.

[0024] FIG. **1** illustrates the condenser **14**. Condenser **14** is shown as a shell and tube flooded-type. The condenser **14** can be arranged as a single evaporator or multiple evaporators in series or parallel, e.g. connecting a separate or multiple evaporators to each compressor. Condenser **14** may include condenser tubing **16**. Fluid **80** may pass across the condenser tubing **16** through which cool air or cool liquid flows.

[0025] Condenser **14** may be fabricated from carbon steel and/or other suitable material, including copper alloy heat transfer tubing. Condenser tubing **16** can be of various diameters and thicknesses, and comprised typically of copper alloy. In addition, condenser tubing **16** may be replaceable, mechanically expanded into tube sheets and externally finned seamless tubing. Other known types of condenser **14** are contemplated.

[0026] Condenser **14** may be configured to communicate fluid **80** from a discharge passage **36**. Discharge passage **36** may be configured to receive the fluid **80**, or may be coupled to the condenser **14** through an oil separator **24**, as depicted in FIG. **1**. Other configurations are contemplated. The oil separator **24**, when employed, separates oil from the fluid **80** and returns the oil via an oil supply passage **26** to the screw compressor **12** for reuse. The oil may be reused to, for example, cool the fluid **80**, cool screw rotors **42**, seal the interfaces between the screw rotors **42** themselves, seal the interfaces between the screw rotors **42** and the walls of a working chamber **44**, and/or lubricate bearings **46**, **48**.

[0027] Condenser **14** may transform the fluid **80** from a superheated vapor to a saturated liquid. As a result of the cool air or cool liquid passing across the condenser tubing **16**, fluid **80** may reject or otherwise deliver heat from the chiller **10** to another fluid, like air or liquid, in a heat transfer relation, which in turn carries the heat out of the system.

[0028] An expansion valve **18** may employed, as shown in FIG. **1**. Expansion valve **18** may be configured to receive fluid **80** from condenser **14**. Fluid **80** received from condenser **14** typically is in a thermodynamic state known as a saturated liquid. The expansion valve **18** may abruptly reduce the pressure of the fluid **80**. The abrupt pressure reduction may cause adiabatic flash evaporation of at least a portion of the fluid **80**. In particular, the adiabatic flash evaporation may result in a liquid and vapor mixture of the fluid **80** that has a temperature that is colder than the temperature of the space to be cooled.

[0029] Evaporator **20** is shown in FIG. **1** as a shell and tube flooded-type. The evaporator **20** can be arranged as a single evaporator or multiple evaporators in series or parallel, e.g. connecting a separate or multiple evaporators to each compressor. Evaporator **20** may include evaporator tubing **22**. Fluid **80** may pass across the evaporator tubing **22** through which cool air or cool liquid flows.

[0030] Evaporator **20** may be fabricated from carbon steel and/or other suitable material, including copper alloy heat transfer tubing. Evaporator tubing **22** can be of various diameters and thicknesses, and comprised typically of copper alloy. In addition, evaporator tubing **22** may be replaceable, mechanically expanded into tube sheets and externally finned seamless tubing. Other known types of evaporator **20** are contemplated.

[0031] Evaporator **20** is configured, as illustrated in FIG. **1**, to receive fluid **80** communicated from the expansion valve **18**. Fluid **80** received by the evaporator **20** in the refrigeration loop may be relatively colder than it was when discharged from the screw compressor **12**. The oil return apparatus **28**, when employed, separates oil from the fluid **80** and returns the oil via an oil return passage **30** to the screw compressor **12** for reuse. The oil may be reused to, for example, cool the fluid **80**, cool screw rotors **42**, seal the interfaces between the screw rotors **42** themselves, seal the interfaces between the screw rotors **42** and the walls of a working chamber **44**, and/or lubricate the bearings **46**, **48**.

[0032] The evaporator **20** may absorb and remove heat from the space to be cooled, and the condenser **14** may subsequently reject the absorbed heat to air or liquid that carries the heat away from the space to be cooled. In operation, warm air or liquid may be circulated from the space to be cooled across the evaporator tubing **22**. The warm air or liquid passing across the evaporator tubing **22** may cause a liquid portion of the cold fluid **80** to evaporate. At the same time, the warm air or liquid passed across the evaporator tubing **22** may be cooled by the fluid **80**. It should be understood that any configuration of the condenser **14** and/or evaporator **20** may be employed that accomplishes the necessary phase changes of fluid **80**.

[0033] The chilled or heated water is pumped from the evaporator **20** to an air handling unit (not shown). Air from the space that is being temperature conditioned is drawn across coils in the air handling unit that contains, in the case of air conditioning, chilled water. The drawn-in air is cooled. The cool air is then forced through the air conditioned space, which cools the space.

[0034] Additionally, though not shown, an economizer **32** may be incorporated to include an economizer cycle. Economizer **32** or a subcooling cycle (not shown), or both, may be employed in the refrigeration cycle and return the fluid **80** to the screw compressor **12** via suction passage **34** or other passage (not shown) depending on the configuration required the application conditions.

[0035] Referring to FIGS. **2** and **3**, screw compressor **12** typically comprises a rotor housing **40** and an electric motor housing **50**. Screw compressor **12** may be formed, all or in part, of gray cast iron, for example. Other materials may be used to form the screw compressor **12**. Screw compressor **12**, according to embodiments of the present invention, facilitates highly efficient operation at full-load and part-load conditions over a preselected screw capacity range.

[0036] Motor housing **50** houses a motor **52** in an embodiment of the present invention. Electric motor **52** may coupled to a variable frequency drive **38**. The electric motor **52** drives meshed screw rotors **42**. Motor housing **50** may be integral to the rotor housing **40**.

[0037] The rotor housing **40** may have a low pressure end and a high pressure end that each contain a suction port **76** and discharge port **78**, respectively. Suction port **76** and discharge port **78** are in open-flow communication with the working chamber **44**. The suction port **76** and the discharge port **78** may each be an axial, a radial or a mixed (a combination of a radial and an axial) port.

[0038] The suction port **76** may receive the fluid **80** at a suction pressure and a suction temperature. The suction port **76** may receive fluid **80** from suction passage **34** in thermodynamic states known as a saturated vapor or a superheated vapor. The screw compressor **12** may compress the fluid **80** as the screw compressor **12** communicates the fluid **80** from the suction port **76** to the discharge port **78**. Fluid **80** passing through the discharge port **78** discharges into discharge passage **36**.

[0039] Compressing the fluid **80** may also result in the fluid **80** being discharged at a discharge temperature that is higher than the suction temperature. The fluid **80** discharged from the discharge port **78** may be in a thermodynamic state known as a superheated vapor. Accordingly, fluid **80** discharged from the screw compressor **12** may be at a temperature and a pressure at which the fluid **80** may be readily condensed with a cooling air or a cooling liquid.

[0040] Suction port **76** and discharge port **78** are configured to minimize flow losses, when at least one of the rotors **42** is operated at an approximately constant peripheral velocity. The suction port **76** may be located where fluid **80** exits the suction area of screw compressor **12** and is drawn into the working chamber **44**. The suction port **76** may be sized to be as large as possible to minimize, at least, the approach velocity of the fluid **80**. The location of the suction port **76** in the rotor housing **40** also may be configured to minimize turbulence of fluid **80** prior to entry into the rotors **42**.

[0041] Discharge port **78** may be sized larger than theoretically necessary to provide a thermodynamic optimum size and thereby, reduce the velocity at which the fluid **80** exits the working chamber **44**. The discharge port **78** may be generally located where fluid **80** exits the working chamber **44** of screw compressor **12**. The discharge port **78** location in the rotor housing **40** may be configured such that the maximum discharge pressure can be attained in the rotors **42** prior to being delivered into the discharge passage **36**. In addition, screw compressor **12** may incorporate a muffler **58** or other apparatus suitable for noise reduction.

[0042] Referring again to FIG. **3**, rotors **42** are mounted for rotation in a working chamber **44**. The working chamber **44** comprises a volume that is shaped as a pair of parallel, intersecting flat-ended cylinders, and is closely toleranced to the exterior dimensions and geometry of the intermeshed screw rotors **42**. The plurality of meshed screw rotors **42***a*, **42***b* may define one or more compression pockets between the screw rotors **42***a*, **42***b* and the interior chamber walls of the rotor housing **40**. The rotor housing **40** has little separation from the rotors **42**. Milling, machine grinding or molding can be employed to achieve high accuracy and tight tolerances between rotors **42** flutes and lobes and the rotor housing **40**.

[0043] First screw rotor **42***a* and second screw rotor **42***b* are disposed in a counter-rotating, intermeshed relationship and cooperate to compress a fluid. At least one of rotors **42** is cooperatively configured with motor **52** to be operable at a rotational speed for a screw compressor capacity within a preselected screw compressor capacity range. The selected rotational speed at full-load capacity is substantially greater than a synchronous motor rotational speed at a rated capacity (also referred to herein as rated screw compressor capacity) for screw compressor **12**.

[0044] In the embodiment illustrated, rotor **42***a* may be called a female screw rotor and comprise a female lobed/fluted body or working portion (typically a helical or spiral extending land and groove). Rotor **42***b* may be called a male screw rotor and comprise a male lobed/fluted body or working portion (typically a helical or spiral extending land and groove).

[0045] Rotors **42** include shaft portions, which are, in turn, mounted to the housing of screw compressor **12** by, for example, one or more bearings **46**, **48**. The exemplary bearings **46**, **48** will also be configured with tight clearances in relation to at least rotors **42** and rotor housing **40**.

[0046] Compression of the fluid **80** in screw compressor **12** produces axial and radial forces. The configurations of embodiments of the present invention may also mitigate time varying and non-uniform rotor movements and forces against chamber walls, bearings, and end surfaces of the screw compressor **12** caused by the interaction of the screw rotors **42***a*, **42***b*, the axial forces, and the radial forces.

[0047] As mentioned, a lubricating fluid, typically oil, may be delivered from oil supply passage **26** or oil return passage **30** to the screw compressor **12**. The lubricating fluid provides cushioning films for the walls of the working chamber **44**, rotors **42***a*, **42***b*, and bearings **46**, **48** of the screw compressor **12**, but does little to prevent the transmission of the time varying and non-uniform axial and radial forces. The screw compressor **12** may also utilize an expander (not shown), which may also be integral to screw compressor **12**, to recover energy available from the refrigeration cycle as the high pressure liquid expands through the expander to a lower pressure.

[0048] The electric motor **52** in one exemplary embodiment may drive at least one of the rotors **42** in response to command signals **62** received from the controller **60**. The horsepower of preferred motor **52** can vary in the range of about 125 horsepower to about 2500 horsepower. Torque supplied by the electric motor **52** may directly rotate at least one of the screw rotors **42**. Employing motor **52** and variable speed drive **38**, screw compressor **12** of embodiments of the present invention may have a rated screw compressor capacity within the range of about 35-tons to about 150-tons or more and have a full-load speed range within about 4,000 revolutions per minute to about 15,000 revolutions per minute, when the fluid is an R-134a refrigerant.

[0049] While conventional types of motors, like induction motors, can be used with and will provide a benefit when employed with embodiments of the present invention, a preferred motor **52** comprises a direct drive, variable speed, hermetic, permanent magnet motor. Permanent magnet motor **52** can increase system efficiencies over other motor types. The choice of motor **52** may be affected by cost and performance considerations.

[0050] Referring to FIGS. **2** and **3**, the permanent magnet motor **52** comprises a motor stator **54** and a motor rotor **56**. Stator **54** consists of wire coils formed around laminated steel poles, which convert variable speed drive **38** applied currents into a rotating magnetic field. The stator **54** is mounted in a fixed position in the screw compressor **12** and surrounds the motor rotor **56**, enveloping the rotor **56** with the rotating magnetic field. Motor rotor **56** is the rotating component of the motor **52** and may consist of a steel structure with permanent magnets, which provides a magnetic field that interacts with the rotating stator magnetic field to produce rotor torque. In addition, permanent magnet motor **52** may be configured to receive variable frequency control signals and to drive the at least two screw rotors per the received variable frequency control signals.

[0051] The motor rotor **56** may have a plurality of magnets and may comprise magnets buried within the rotor steel structure or be mounted at the rotor steel structure surface. Motor rotor **56** surface mount magnets are secured with a low loss filament, metal retaining sleeve or by other means to the rotor steel support. Further manufacturing, performance, and operating advantages and disadvantages can be realized with the number and placement of permanent magnets in the motor rotor **56**. For example, surface mounted magnets can be used to realize greater motor efficiencies due to the absence of magnetic losses in intervening material, ease of manufacture in the creation of precise magnetic fields, and effective use of rotor fields to produce responsive rotor torque. Likewise, buried magnets can be used to realize a simpler manufactured assembly and to control the starting and operating rotor torque reactions to load variations.

[0052] The performance and size of the permanent magnet motor **52** is due in part to the use of high energy density permanent magnets. Permanent magnets produced using high energy density magnetic materials, typically at least 20 MGOe (Mega Gauss Oersted), produce a strong, more intense magnetic field than conventional materials. With a motor rotor **56** that has a stronger magnetic field, greater torques can be produced, and the resulting motor **52** can produce a greater horsepower output per unit volume than a conventional motor, including induction motors. By way of comparison, the torque per unit volume of permanent magnet motor **52** can be at least about 75 percent higher than the torque per unit volume of induction motors used in refrigeration chillers of comparable refrigeration capacity. The result is a smaller sized motor to meet the required horsepower for a specific compressor assembly.

[0053] The permanent magnet motor **52** of an embodiment of the present invention is compact, efficient, reliable, and relatively quieter than conventional motors. As the physical size of the screw compressor **12** is reduced, motor **52** used can be scaled in size to fully realize the benefits of improved fluid flow paths and compressor element shape and size. Motor **52** is reduced in volume by approximately 30 percent or more, when compared to conventional existing designs for compressor assemblies that employ induction motors and have refrigeration capacities in excess of 35-tons. The resulting size reduction of embodiments of the present invention provides a greater opportunity for efficiency, reliability, and quiet operation through use of less material and smaller dimensions than has been achieved through more conventional practices.

[0054] Any bearings employed with motor **52** may be rolling element bearings (REB) or hydrodynamic journal bearings. Such bearings may be oil lubricated. Oil-free bearing systems may be employed. A special class of bearing which is refrigerant lubricated is a foil bearing and another bearing type uses REB with ceramic balls. Each bearing type has advantages and disadvantages that should be apparent to those of skill in the art. Bearings should be selected to facilitate highly efficient operation of the screw compressor **12** at reduced speeds for capacity modulation and to minimize rotor dynamics and vibration associated with reduced speeds. Any bearing type may be employed that is suitable of sustaining rotational speeds in the range of about 2,000 RPM to about 20,000 RPM.

[0055] The motor rotor **56** and motor stator **54** end turn losses for the permanent magnet motor **52** are very low compared to some conventional motors, including induction motors. The motor **52**, therefore, may be cooled by means of fluid **80** (typically, refrigerant). When fluid **80** is employed for cooling motor **52**, fluid **80** may only need to contact the outside diameter of the stator **54**. Cooling the motor **52** in this way allows for the elimination of the motor cooling feed ring that is typically used in induction motor stators. Alternatively, refrigerant may be metered to the outside surface of the stator **54** and to the end turns of the stator **54** to cool the motor **52**.

[0056] In addition, the torque that is needed from motor **52** comes essentially from the internal pressure distribution in the rotors **42**, which is a function of rotors **42** geometry and the operating conditions. That internal pressure distribution within the rotors **42** provides the load against which the motor **52** has to work. Employing embodiments of this invention without a mechanical unloader results in a theoretical torque that may be essentially constant over a full range of operating conditions, and for a given operating condition, a ratio of theoretical to actual torque on the motor **52** that may be approximately constant, despite decay in the actual torque during operation due to changing losses and leakage, for example. In contrast, for a given operating condition, conventional screw compressors invoking a mechanical unloader will have significant torque fluctuations or variations over time.

[0057] As illustrated in FIG. **4**, a variable speed drive **38** may drive the motor **52** and in turn, screw compressor **12**. The speed of the motor **52** can be controlled by varying, for example, the frequency of the electric power that is supplied to the motor **52**. Use of a permanent magnet motor **52** and variable speed drive **38** moves some conventional motor losses outside of the refrigerant loop. The efficiency of the variable speed drive **38**, line input to motor shaft output, preferably can achieve a minimum of about 95 percent over the system operating range.

[0058] The variable speed drive **38** drives the screw compressor **12** at the optimum, or near optimum, rotational speed at each capacity over the preselected screw compressor capacity range for a screw compressor **12** of a given rated capacity. The variable speed drive **38** may be refrigerant cooled, water cooled or air cooled. As mentioned, similar to cooling of motor **52**, the variable speed drive **38**, or portions thereof, may be by using a refrigerant circulated within the chiller system **10** or by other conventional cooling means. How the motor **52** and/or variable speed drive **38** are cooled should be understood as dependent on the operational and environmental conditions in which the motor **52** and/or variable speed drive **38** reside in operation.

[0059] The variable speed drive **38** typically will comprise an electrical power converter comprising a line rectifier and line electrical current harmonic reducer, power circuits and control circuits (such circuits further comprising all communication and control logic, including electronic power switching circuits). Conditions in which the screw compressor **12** is employed may justify employing more than one variable speed drive **38** for chiller **10**.

[0060] The variable speed drive **38** can be configured to receive command signals **62** from a controller **60** and to generate a control signal **64**. The variable speed drive **38** will respond, for example, to signals **62** received from a microprocessor (also not shown) associated with controller system **60** to increase or decrease the speed of the motor **52** by changing the frequency of the current supplied to motor **52**. Controller **60** may be configured to receive status signals **82** indicative of an operating point of the screw compressor, and to generate command signals that requests the electric motor system to drive the screw compressor per a preselected operating parameter. Status signals **82** may deliver similar or different status information depending, for example, on the intended purpose of the sensor selected. Controller **60** may generate command signals **62** per a preselected operating parameter, like a torque profile for screw compressor **12**. Control signal **64** can drive the high energy density motor **52** at a rotational speed substantially greater than a synchronous motor rotational speed for the rated screw compressor capacity and drive the motor **52**, and in turn at least one screw rotor **42**, at an optimum peripheral velocity independent of the rated screw compressor capacity.

[0061] The motor **52** and the variable speed drive **38** have power electronics for low voltage (less than about 600 volts), 50 Hz and 60 Hz applications. Typically, an AC power source (not shown) will supply multiphase voltage and frequency to the variable speed drive **38**. The AC voltage or line voltage delivered to the variable speed drive **38** will typically have nominal values of 200V, 230V, 380V, 415V, 480V, or 600V at a line frequency of 50 Hz or 60 Hz depending on the AC power source.

[0062] By the use of motor **52** and variable speed drive **38**, the speed of motor **52** can be varied to match varying system requirements. Speed matching results in approximately 30 percent more efficient system operation compared to a compressor without a variable speed drive **38**. By running compressor **12** at lower speeds when the load on the chiller is not high or at its maximum, sufficient refrigeration effect can be provided to cool the reduced heat load in a manner which saves energy, makes the chiller **10** more economical from a cost-to-run standpoint, and facilitates highly efficient chiller **10** operation as compared to chillers which are incapable of such load matching at the rotational speeds possible via embodiments of the present invention. For example, a rated screw compressor capacity of about 100-tons configured according to embodiments of the present invention could be efficiently operable over a preselected screw capacity range of about 75-tons to about 125-tons.

[0063] Screw compressor **12** can be operated at rotational speeds substantially higher than synchronous motor rotational speeds for a given rated capacity of the screw compressor **12**. The specific optimum speed for the rated screw compressor capacity range is a function of screw compressor capacity and head pressure. Embodiments of the present invention dramatically improve the discharge porting of fluid **80** and in turn, allow for screw compressor **12** to be operated at a significantly increased rotational speed over the rotational speed that gives the best performance for conventionally sized rotors and ports. For example, the selected rotational speed for a rated screw compressor capacity of about 100-tons, according to embodiments of the present invention, is about 5800 revolutions per minute, when the fluid is an R-134a refrigerant. In contrast, a conventional screw compressor with a rated capacity of about 100-tons has a synchronous motor rotational speed is about 3400 revolutions per minute, when the fluid is an R-134a refrigerant.

[0064] The allowable range of rotational speed for a particular rated capacity of a screw compressor **12** is selected to achieve an optimum peripheral velocity of at least one of the screw rotors independent of the rated capacity of screw com-

pressor **12** that results in a relatively uniform high efficiency across the screw compressor product family (e.g. 60-tons, 80-tons, 100-tons and 150-tons.) The optimum peripheral velocity is a constant product of the rotational speed and the radius of at least one of the rotors **42**, typically, the male rotor **42***b*. The approximately constant optimum peripheral velocity is, for example, in the range between about 131 feet per second (about 40 meters per second) to about 164 feet per second (about 50 meters per second). In one embodiment, the approximately constant optimum peripheral velocity is between about 42 meters per second (about 137 feet per second) to about 45 meters per second (about 147 feet per second) in high pressure applications, when R-134a refrigerant is the fluid **80**. Persons of skill in the art would understand that, for a low pressure application or for a different primary fluid **80**, or both, the optimum peripheral velocity may be different.

[0065] The rotational speed of the motor **52** may be selected in combination with configuring rotors **42**, suction port **76** and discharge port **78** for each target capacity to achieve an approximately constant optimum peripheral velocity of at least one of the screw rotors **42** regardless of the rated capacity of the screw compressor **12**. That is, specific combinations of screw rotors **42**, inlet port **76**, discharge port **78** and the operational rotational speed are selected such that each specific combination enables each screw compressor **12** to run at approximately the same optimum peripheral velocity for each different rated capacity and, in turn, to produce relatively the same high efficiency between or among each different rated capacity of screw compressor **12**.

[0066] Embodiments of the present invention include a method of sizing of at least two screw compressors **12** with different rated capacities that achieve approximately constant efficiency across the screw compressor product family (e.g. 60-tons, 80-tons, 100-tons and 150-tons.). By employing embodiments of this invention, the isoentropic efficiency versus capacity (in tons) of screw compressor **12** is significantly increased, on the order of 15 percent, over a conventional screw compressor. In addition, because screw compressor **12** is operated at relatively higher speed, the screw compressor **12** can slowed down on the order of 20-30 percent of the speed for the operating capacity and still have an approximately constant peak efficiency or efficiency plateau as compared to the efficiency at the rated screw compressor capacity.

[0067] The target capacity for each screw compressor **12**, each having a different rated capacity, is selected. The rotational speed is also selected based on the target capacity of each screw compressor **12** to operate at least one screw rotor **42** in each screw compressor **12** at an approximately constant optimum peripheral velocity that is independent of the rated capacity of each screw compressor **12**. The suction port **76**, the at least two screw rotors **42** and the discharge port **78** are configured together with the rotational speed selected for each screw compressor **12**.

[0068] Specifically, driving screw compressor **12** at an optimum peripheral velocity allows for each rotor **42** to have a geometry and a profile that may remain the same for a wide range of preselected screw compressor capacities for the rated screw compressor capacity. Each of the rotors **42**, though, may have a different geometry and a profile for each different rated screw compressor capacity that will enable at least one screw rotor to be operated at a selected rotational speed that produces an approximately constant optimum peripheral velocity between or among each rated capacity of each screw compressor **12**. The volumetric ratio of the screw compressor **12** is selected as a function of the loading conditions in which the screw compressor **12** will be used. By way of example, in embodiments of the present invention, more than two volumetric ratios, potentially four, five or more, are contemplated over a range of rated screw compressor capacities. The volumetric ratio may also be such that the system compression ratio and the internal compression ratio closely match. The rotor **42** profile may be a balance of the length of the sealing line, flow cross sectional area and blow-hole area size.

[0069] The geometry and profile are generally defined, in part, by the number of lobes in each rotor, the wrap angle, the length of the rotors and the diameter of the rotors, for example. Screw rotor **42** has a profile taken in a plane transverse to the parallel axes of the male rotor **42***b* and the female rotor **42***a*. The profile of rotors **42** can be symmetric or asymmetric, and circular, elliptical, parabolic, hyperbolic, for example. Rack generation of rotors **42** profile may be employed. Selecting a profile of rotors **42** is a balance of the internal leakage path of fluid **80** during operation of screw compressor **12** and the porting configuration of suction port **76** and discharge port **78**, such that screw compressor **12** has an approximately constant optimum peripheral velocity.

[0070] More specifically, for example, at an about 44 m/s optimum peripheral velocity for at least one rotor **42** of a 100-ton screw compressor, the resulting male rotor **42***b* has a wrap angle of about 347 degrees and the female rotor **42***a* has a wrap angle that is 6/7ths of the male rotor **42***b*. The wrap angle of the female rotor **42***a* varies with the ratio of number of lobes. The female rotor **42***a* has a radius of about 2.5 inches (6.35 centimeters) and **7** lobes and the male rotor **42***b* has a radius of about 3 inches (7.62 centimeters) and **6** lobes. The length of rotors **42** is significantly smaller, on the order of about 20-30 percent smaller, than a conventionally sized screw compressor at the rated screw compressor capacity. A person of skill in the art will appreciate that analytical techniques can be employed for other combinations of rotor **42** profiles for a given rated screw compressor capacity within the scope of the present invention.

[0071] Employing a geometry/profile of rotors **42** for a screw compressor **12** having a preselected screw compressor range and operable at an approximately constant optimum peripheral velocity, allows for operation of the screw compressor **12** at 25 or more percent less than the rated screw compressor capacity without significant adverse rotor dynamic effects. Screw compressor **12** has an improved rotor profile that maximizes internal flow area, internal friction due to relative motion of the rotor **42** surfaces is minimized, and leakage paths are reduced. This reduced leakage and higher flow tend to increase the screw compressor **12** efficiency and reduce power wasted, which increases overall efficiency.

[0072] Referring now to FIG. **4**, further details regarding an embodiment of the chiller **10** are presented. In particular, chiller **10** may include a controller or controller system **60**. Controller **60** may be arranged to communicate with the variable frequency drive **38**, screw compressor **12**, condenser **14** and evaporator **20**. Chiller **10** may further include one or more sensors. Sensors **66**, **68**, **70**, **72** and **74**, for example, may be employed to sense and/or communicate torque, suction pressure and/or temperature, discharge pressure and/or temperature, and/or other measurable parameter. Other sensors could be employed depending on the application in which screw compressor **12** is used. Signals **82** may be communicated via wiring, fiber optics, wireless and/or a combination

of wiring, fiber optics and wireless. The sensors **66**, **68**, **70**, **72** and **74** communicate status signals **82** to controller **60** with data that are indicative of the operation of various components of the chiller **10**.

[0073] The controller **60** may include processors, microcontrollers, analog circuitry, digital circuitry, firmware, and/or software (not shown) that cooperate to ultimately control operation of the screw compressor **12**. The memory may comprise non-volatile memory devices such as flash memory devices, read only memory (ROM) devices, electrically erasable/programmable ROM devices, and/or battery backed random access memory (RAM) devices to store an array of performance related characteristics for the screw compressor **12**. The memory may further include instructions which the controller **60** may execute in order to control the operation of the screw compressor **12**.

[0074] The controller **60** may receive status signals from one or more sensors **66**, **68**, **70**, **72** and **74** that provide information regarding operation of the screw compressor **12**. Based upon the status signals, the controller **60** may determine an operating mode and/or operating point of the screw compressor **12** and may generate, based upon the determined operating mode and/or operating point, one or more command signals **62** to adjust the operation of the screw compressor **12**. The controller **60** may then generate command signals **62** that request the motor **52** to operate according to a preselected operating parameter(s) (e.g. a torque profile). For example, the controller **60** may enable operation at an optimal torque and speed of screw compressor **12** to minimize losses, mechanical wear and losses. Further disclosure of a controller system **60** suitable for use with embodiments of the present invention may be found in co-pending application U.S. patent Ser. No. 12/544,582, assigned to the assignee of the instant application, which is hereby incorporated by reference.

[0075] It should be apparent that variations on the control system **60** described above will be apparent to those skilled in the art. The control system **60** may be implemented with electronic digital, analog, or a combination of digital/analog control elements and low-voltage wiring. Other conventional pneumatic tubing, transmitters, controllers, and relays are contemplated.

[0076] In addition, it also will be readily apparent to one of ordinary skill in the art that the compressor system disclosed can be readily implemented in other contexts at varying scales. Use of various motor types, drive mechanisms, and configurations with embodiments of this invention should be readily apparent to those of ordinary skill in the art.

[0077] Employing embodiments of the present invention, as compared to conventional approaches, increase full-load efficiency, yield higher part-load efficiency and have a practically constant efficiency over a given capacity range, controlled independently of power supply frequency or voltage changes. Also, an advantage of embodiments of the present invention is that screw compressors **12** of different rated capacity can each have a variable capacity and still have the approximately same the level of efficiency and without mechanical unloading.

[0078] Additional advantages include a reduction in the physical size of the screw compressor and chiller system arrangement, improved scalability of the screw compressors throughout the operating range and a reduction in total sound levels. Employing embodiments of screw compressor **12** can also effectively reduce costs for the manufacturer, because it allows for one screw compressor at a rated screw compressor capacity (e.g. 100-tons) to serve as an efficient screw compressor at a range of preselected screw compressor capacity range (e.g. 80 tons and 125 tons) without the need for multiple other screw compressors to be manufactured at each additional target capacity within the preselected screw compressor rated capacity range. Practically, embodiments of the present invention also allow for lower physical part count and inventory for a product family with no loss in capacity or performance due to power supply because, for a given rated capacity of screw compressor (e.g. 100-tons), the screw compressor **12** at 50 Hertz and 60 Hertz are nearly identical.

[0079] Embodiments of a bearing housing of a screw compressor are further described. A screw compressor of a HVAC system may include one or more rotors. The one or more rotors of the screw compressor can be typically supported by bearings, such as for example axial and/or radial bearings. In some screw compressors, the bearings supporting the rotors can be enclosed and/or supported by a bearing housing.

[0080] FIG. **5** illustrates a partial side sectional view of a screw compressor **100** that includes a first rotor **110** and a second rotor **120**. The relative motions of the first rotor **110** and the second rotor **120** can be used to compress a working fluid, such as for example refrigerant vapor.

[0081] The first and second rotors **110** and **120** are positioned inside a rotor housing **130**. A bearing assembly including a bearing cover **170** and a bearing housing **160** that is positioned at an axial end of the rotor housing **130** along an axial direction defined by an axis C of the first rotor **110**. The bearing assembly generally covers the rotor housing **130** at the axial end. The compressed working fluid is typically discharged through the bearing assembly.

[0082] The first rotor **110** and the second rotor **120** include an axially extended first shaft **112** and a second shaft **122** respectively along the axial direction defined by the axis C of the first rotor **110**. The first shaft **112** and the second shaft **122** can be supported by one or more bearings, such as for example, discharge radial bearings **140***a* and **140***b* and/or axial bearings **150***a* and **150***b* (sometimes referred to as "thrust bearings") respectively. The discharge radial bearings **140***a* and **140***b* and the axial bearings **150***a* and **150***b* can be attached to the first and second shafts **112** and **122**. The axial bearing **150***a* and **150***b*, as well as the discharge radial bearings **140***a*, **140***b* can be retained in the axial direction by axial bearing retainers **142**.

[0083] The bearings **140***a*, **140***b*, **150***a* and **150***b* can help support the first rotor **110** and the second rotor **120** during operation, and facilitate the rotation of the first and second rotors **110** and **120** around the axis C. For example, the discharge radial bearings **140***a* and **140***b* and the axial bearing **150***a* and **150***b* can help withstand forces that may be produced during operation.

[0084] The bearing housing **160** is positioned at the axial end of the rotor housing **130**. The bearing housing **160** can be configured to enclose and/or support the discharge radial bearings **140***a* and **140***b*. The discharge radial bearings **140***a* and **140***b* are typically positioned closer to the rotor housing **130** than the axial bearings **150***a* and **150***b* in the axial direction.

[0085] The bearing housing **160** includes a first cavity **162** and a second cavity **164** configured to enclose and/or support the discharge radial bearings **140***a* and **140***b* respectively. The bearing housing **160** can be a slab-like structure, which may help provide a relatively uniform thermol expansion. The discharge radial bearings **140***a* and **140***b* can be supported by

walls of the first cavity **162** and the second cavity **164** respectively, and may form interference fit with the walls of the cavities **162** and **164** respectively so that the bearing housing **160** can support the discharge radial bearings **140***a* and **140***b* during operation, such as for example, help the discharge radial bearings **140***a* and **140***b* withstand forces that may be produced during operation.

[0086] The screw compressor **100** also includes the bearing cover **170**, which can be attached to the bearing housing **160** to form an enclosed space **180**. As illustrated in FIG. **5**, the axial bearings **150***a* and **150***b*, portions of the first and second shafts **112** and **122**, and the axial bearing retainers **142** can be enclosed inside the space **180**.

[0087] FIG. **6** illustrates an enlarged view of an area A of FIG. **5**. The slab-like bearing housing **160** includes the first cavity **162** and the second cavity **164** to enclose the first discharge radial bearing **140***a* and the second discharge radial bearing **140***b* respectively.

[0088] The first discharge radial bearing **140***a* has a first length L1 and the second discharge radial bearing **140***b* has a second length L2 in the axial direction. The first cavity **162** has a first depth D1 and the second cavity **164** has a second depth D2 in the axial direction. The first depth D1 can be configured to be about the same as the first length L1. In some embodiments, the first depth D1 can be configured to be no more than the first length L1. The second depth D2 can be configured to be about the same as the second length L2. In some embodiments, the second depth D2 can be configured to be no more than the second length L2. The first depth D1 and the second depth D2 generally are configured so that the first cavity **162** and the second cavity **164** can enclose and/or support the first discharge radial bearing **140***a* and the second discharge radial bearing **140***b* respectively. In some embodiments, the first cavity **162** and the second cavity **164** typically are configured not to enclose and/or support the first and second axial bearings **150***a* and **150***b*.

[0089] In some embodiments, the first discharge radial bearing **140***a* can be fully supported by the first cavity **162**. In some embodiments, the second discharge radial bearing **140***b* can be fully supported by the second cavity **164**. The term "fully supported" is referred to a situation where the support (e.g. interference fit) provided by the walls of the first cavity **162** or the second cavity **164** to the first discharge radial bearing **140***a* or the second discharge radial bearing **140***b* respectively will stop increasing as the first depth D1 or the second depth D2 increases.

[0090] The bearing housing **160** has a depth W1 in the axial direction. The depth W1 can have a minimal depth, which may be configured for example to withstand a compression pressure produced during operation.

[0091] In some embodiments, it can be beneficial to limit or minimize the depth W1, so that a mass of the bearing housing **160** can be limited or minimized. This can help limit or minimize, for example, thermal expansion of the bearing housing **160**, which may help maintain a clearance between the bearing housing **160** and the first rotor **110** or the second rotor **120**.

[0092] The depth W1 can be affected by, for example, the minimal depth as well as other factors such as the depth D1 and/or the depth D2. Generally, the larger the depth D1 and/or the depth D2 is, the larger the depth W1. In some embodiments, the depth D1 and/or the depth D2 may be configured so that the first cavity **112** and/or the second cavity **122** can be configured to fully support the first radial bearing **140***a* and/or the second radial bearing **140***b*. In some embodiments, the depth D1 and/or the depth D2 may be configured so that the bearing housing **160** does not extend to the axial bearings **140***a* and/or **140***b* in the axial direction. In some embodiments, the depth D1 and/or the depth D2 may be configured to be no more than the length L1 and the length L2 respectively. In some embodiments, the depth D1 and/or the depth D2 may be configured so that the first cavity **112** and/or the second cavity **122** may be no more than a depth that is required to fully support the first radial bearing **140***a* and/or the second radial bearing **140***b*. In some embodiments, the difference between the depth D1 and/or the depth D2 and the length L1 and/or L2 may be about 3 mm, so that a maximum of about 3 mm of the first and/or second discharge radial bearings **140***a* and **140***b* is not enclosed and/or unsupported by the first and/or the second cavities **162** and **164**.

[0093] Referring to FIGS. **1** and **2**, the bearing cover **170** generally has a dome shape, which can help define the space **180** with the bearing housing **160**. The first and second axial bearings **150***a* and **150***b* can be enclosed in the space **180**. Referring to FIG. **6**, the first axial bearing **150***a* (and/or the second axial bearing **150***b*, which is not illustrated in FIG. **5**) can have a clearance **182** between the first axial bearing **150***a* (and/or the second axial bearing **150***b*) and the bearing cover **170**. The first axial bearing **150***a* and/or the second axial bearing **150***b* does not generally need a support from the bearing housing **160** and/or the bearing cover **170** because the first axial bearing **150***a* and/or the second axial bearing **150***b* typically are configured to withstand forces in the axial direction.

[0094] It is to be appreciated that because the bearing cover **170** can have the clearance **182** with the first axial bearing **150***a* (and/or the second axial bearing **150***b*), therefore the bearing cover **170** does not have to be precision machined.

[0095] In the illustrated embodiment, the first axial bearing **150***a* and/or the second axial bearing **150***b* are arranged next to the first discharge radial bearing **140***a* and/or the second discharge radial bearing **140***b* in the axial direction. The first axial bearing **150***a* and the second axial bearing **150***b* are positioned further away from the first rotor **110** and/or the second rotor **120** relative to the first and second discharge radial bearings **140***a* and **140***b* respectively in the axial direction. Because the axial bearing **150***a* and/or the second axial bearing **150***b* do not necessarily require a support provided by the bearing housing **160** or the bearings cover **170**, it is not necessary to enclose and/or support the first axial bearing **150***a* and/or the second axial bearing **150***b* in the first cavity **162** or the second cavity **164**. To help minimize or limit, for example, the mass of the bearing housing **160**, the first depth D1 and/or the second depth D2 can be configured so that the first cavity **162** and/or the second cavity **164** do not extend in the axial direction to enclose and/or support any portion of the first and/or second axial bearing **150***a*, **150***b*. The first axial bearing **150***a*, the second axial bearing **150***b*, and the axial bearing retainer **142** can be enclosed by the bearing housing **170**.

[0096] It is noted that the arrangement of the first and second radial bearings **140***a* and **140***b*, as well as the first and second axial bearings **150***a* and **150***b* are exemplary. The arrangement of bearings can be varied. The number of discharge radial bearings and/or axial bearings can also vary.

[0097] FIG. **7** illustrates a known conventional bearing housing **360** of a screw compressor **300**. The bearing housing **360** includes a first cavity **362** and a second cavity **364**. The

first cavity **362** and the second cavity **364** are configured to enclose both a first and a second discharge radial bearings **340***a*, **340***b*, a first and a second axial bearings **350***a*, **350***b*, as well as axial bearing retainers **342**. The bearing housing **360** is covered by a bearing cover **370**.

[0098] It can be seen for example that the bearing housing **360** is relatively longer in an axial direction defined by an axis C3 of a first rotor **310** relative to the bearing housing **160** of FIG. **6**. That is, the bearing housing **160** in FIG. **6** is relatively shorter in that the bearing housing **160** encloses the first and second radial bearings **140***a*, **140***b*, but does not enclose the first and second axial bearings **150***a*, **150***b* or the axial bearing retainers **142**, whereas the bearing housing **360** in FIG. **7** extends to enclose both the first and second discharge radial and axial bearings **140***a*, **140***b*, **150***a* and **150***b*, as well as the axial bearing retainers **342**.

[0099] Screw compressor performance and reliability may be linked to the precise location of the discharge radial bearings (e.g. the first and second radial bearings **140***a*, **140***b*), which can be configured to support and locate the rotors (e.g. the first and second rotors **110** and **120**). By reducing the width W1 of the bearing housing **160** relative to the conventional bearing housing **360**, the first and second cavities **162**, **164** can be produced more precisely and simply than the first and second cavities **362**, **364** of the convention bearing housing **360** due to shorter reaches and shorter machine boring bars. This can help position the first and second radial bearings **140***a* and **140***b* more accurately compared to the first and second radial bearings **150***a* and **150***b*, which can help both the performance and reliability of the screw compressor, by for example more accurately positioning the first and second rotors **110**, **120**.

[0100] The bearing cover (e.g. the bearing housing **170**) is typically not a critical part, and can be machined relatively cheaply and generally does not require precision machining tools. In the embodiment as disclosed herein, the bearing assembly includes the relatively long bearing cover **170** and the relative short bearing housing **160** compared to the relatively short bearing cover **370** and the relatively long bearing housing **360**. The overall cost of making the bearing assembly is reduced compared to a convention design. It is also simpler to make.

[0101] It is to be understood that it is possible to make the bearing housing thicker than it is necessary to accommodate the discharge radial bearing to it enclosed at lead a portion of the discharge radial bearings

[0102] With regard to the foregoing description, it is to be understood that changes may be made in detail, without departing from the scope of the present invention. It is intended that the specification and depicted embodiments are to be considered exemplary only, with a true scope and spirit of the invention being indicated by the broad meaning of the claims.

What is claimed is:

1. A bearing housing of a screw compressor, comprising:
a cavity defined by the bearing housing, wherein a depth of the cavity is configured to be no larger than a length of a radial bearing of a screw compressor.

2. The bearing housing of claim **1**, wherein the bearing housing is configured to support a radial bearing of the screw compressor.

3. The bearing housing of claim **1**, wherein the bearing housing is configured to be positioned to cover an axial end of the screw compressor.

4. A screw compressor, comprising:
a rotor including a shaft;
one or more radial bearings attached to the shaft;
one or more axial bearings attached to the shaft; and
a bearing housing, wherein the bearing housing defines a cavity that is configured to support the one or more radial bearings but not the one or more axial bearings.

5. The screw compressor of claim **4**, wherein the cavity has a depth and the one or more radial bearings have a length, and the depth of the cavity is not larger than the length of the one or more radial bearings.

6. The screw compressor of claim **4**, further comprising:
a bearing cover, the bearing cover configured to be attached to the bearing housing; wherein the bearing cover is configured to have a clearance with the one or more axial bearings.

7. The screw compressor of claim **4**, further comprising:
a bearing cover, the bearing cover defined a space; wherein the space defined by the bearing cover is configured to enclose the one or more axial bearings.

8. The screw compressor of claim **4**, wherein the one or more radial bearings are positioned closer to the rotor than the one or more axial bearings.

9. A variable capacity screw compressor comprising:
a rotor housing comprising a suction port,
a working chamber,
a discharge port, and
at least two screw rotors that include a female screw rotor and a male screw rotor being positioned within the working chamber for cooperatively compressing a fluid;
wherein the suction port, the at least two screw rotors and the discharge port being configured in relation to a selected rotational speed;
the selected rotational speed operates at least one screw rotor at an optimum peripheral velocity that is independent of a peripheral velocity of the at least one screw rotor at a synchronous motor rotational speed for a rated screw compressor capacity;
a motor operable to drive the at least one screw rotor at a rotational speed at a full-load capacity that is substantially greater than the synchronous motor rotational speed at the rated screw compressor capacity; and
a variable speed drive to receive a command signal from a controller and to generate a control signal that drives the motor at the rotational speed.

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