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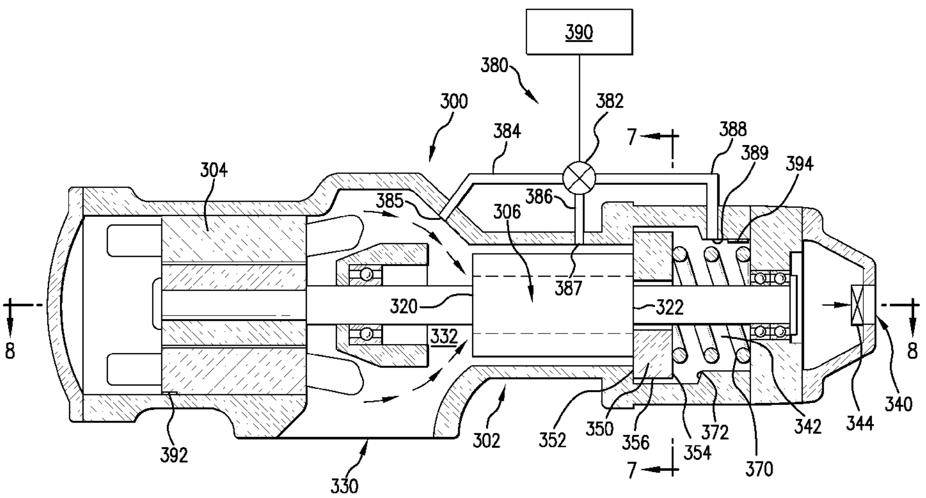
(54) Title: SCREW COMPRESSOR CAPACITY CONTROL



(57) Abstract: A screw compressor has a housing (22; 302) having first (53; 330) and second (58; 340) ports along a flowpath. A first rotor (26; 306) has a lobed body. A second rotor (28; 308, 310) has a lobed body enmeshed with the first rotor body. The rotors and housing cooperate to define a compression path between suction (60; 332) and discharge (62; 342) locations along the flowpath. Means (100, 110, 120; 200, 210, 220; 370, 380, 390) provide relative longitudinal movement between a blocking portion (57; 352) of the housing and at least one of the first rotor and second rotor between: a first condition wherein a pocket of the first and second rotors is closed by the blocking portion; and a second condition wherein the blocking portion does not close the pocket. To provide capacity control, a control system (110; 390) is configured to provide duty cycle control of the movement.

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SCREW COMPRESSOR CAPACITY CONTROL

CROSS-REFERENCE TO RELATED APPLICATION

[0001] Benefit is claimed of US patent application 60/820511, filed July 27, 2006.

BACKGROUND

[0002] The disclosure relates to compressors. More particularly, the disclosure relates to screw-type refrigerant compressors.

[0003] Screw type compressors are commonly used in air conditioning and refrigeration applications. In such a compressor, intermeshed male and female lobed rotors or screws are rotated about their axes to pump the working fluid (refrigerant) from a low pressure inlet end to a high pressure outlet end. During rotation, sequential lobes of the male rotor serve as pistons driving refrigerant downstream and compressing it within the space between an adjacent pair of female rotor lobes and the housing. Likewise sequential lobes of the female rotor produce compression of refrigerant within a space between an adjacent pair of male rotor lobes and the housing. The interlobe spaces of the male and female rotors in which compression occurs form compression pockets (alternatively described as male and female portions of a common compression pocket joined at a mesh zone). In one implementation, the male rotor is coaxial with an electric driving motor and is supported by bearings on inlet and outlet sides of its lobed working portion. There may be multiple female rotors engaged to a given male rotor or vice versa.

[0004] When one of the interlobe spaces is exposed to an inlet port, the refrigerant enters the space essentially at suction pressure. As the rotors continue to rotate, at some point during the rotation the space is no longer in communication with the inlet port and the flow of refrigerant to the space is cut off. After the inlet port is closed, the refrigerant is compressed as the rotors continue to rotate. At some point during the rotation, each space intersects the associated outlet port and the closed compression process terminates. The inlet port and the outlet port may each be radial, axial, or a hybrid combination of an axial port and a radial port.

[0005] It is often desirable to temporarily reduce the refrigerant mass flow through the compressor by delaying the closing off of the inlet port (with or without a reduction in the compressor volume index) when full capacity operation is not required. Such unloading is often provided by a slide valve having a valve element with one or more portions whose positions (as the valve is translated) control the respective suction side closing and discharge side opening of the compression pockets. The primary effect of an unloading shift of the slide valve is to reduce the initial trapped suction volume (and hence compressor capacity); a reduction in volume index is a typical side effect. Exemplary slide valves are disclosed in U.S. Patent Application Publication No. 20040109782 A1 and U.S. Patent Nos. 4,249,866 and 6,302,668.

SUMMARY

[0006] One aspect of the disclosure involves a screw compressor having a housing having first and second ports along a flowpath. A first rotor has a lobed body and an axis and is mounted to the housing for rotation about the axis. A second rotor has a lobed body enmeshed with the first rotor body. The second rotor has an axis and is mounted to the housing for rotation about that axis. The rotors and housing cooperate to define a compression path between suction and discharge locations along the flowpath. Means provide relative longitudinal movement between a blocking portion of the housing and at least one of the first rotor and second rotor between a first condition and a second condition. In the first condition, a pocket of the first and second rotors is closed by the blocking portion. In the second condition, the blocking portion does not close the pocket. To provide capacity control (to achieve a desired loading condition), a control system is configured to provide duty cycle control of the movement.

[0007] In various implementations, at least a movable rotor of the first and second rotors may be mounted for translation between first and second positions along its axis. An actuator may be coupled to at least the movable rotor to shift the movable rotor. Alternatively, the means may provide longitudinal movement of the blocking portion relative to a remainder of the housing between a first position associated with the first condition and a second position associated with the second condition.

[0008] The details of one or more embodiments are set forth in the accompanying drawings and the description below. Other features, objects, and advantages will be apparent from the description and drawings, and from the claims.

BRIEF DESCRIPTION OF THE DRAWINGS

[0009] FIG. 1 is a longitudinal sectional view of a baseline compressor.

[0010] FIG. 2 is a partial, partially schematic, view of a reengineered compressor in a loaded condition.

[0011] FIG. 3 is a view of the compressor of FIG. 2 in an unloaded condition.

[0012] FIG. 4 is a partial, partially schematic, view of a second reengineered compressor in a loaded condition.

[0013] FIG. 5 is a view of the compressor of FIG. 4 in an unloaded condition.

[0014] FIG. 6 is a longitudinal sectional view of a second reengineered compressor in a loaded condition.

[0015] FIG. 7 is a transverse sectional view of the compressor of FIG. 6 taken along line 7-7.

[0016] FIG. 8 is a longitudinal sectional view of the compressor of FIG. 6 taken along line 8-8.

[0017] FIG. 9 is a longitudinal sectional view of the compressor of FIG. 6 in an unloaded condition.

[0018] Like reference numbers and designations in the various drawings indicate like elements.

DETAILED DESCRIPTION

[0019] FIG. 1 shows a baseline compressor 20 having a housing assembly 22 containing a motor 24 driving rotors 26 and 28 having respective central longitudinal axes 500 and 502. For purposes of illustration, the basic structure of the compressor is taken from one existing compressor. However, other existing or yet developed compressor configurations are possible.

[0020] In the exemplary embodiment, the rotor 26 has a male lobed body or working portion 30 extending between a first end 31 and a second end 32. The working portion 30 is enmeshed with a female lobed body or working portion 34 of the female rotor 28. The working portion 34 has a first end 35 and a second end 36. Each rotor includes shaft portions (e.g., stubs 39, 40, 41, and 42 unitarily formed with the associated working portion) extending from the first and second ends of the associated working portion. Each of these shaft stubs is mounted to the housing by one or more bearing assemblies 44 for rotation about the associated rotor axis.

[0021] In the exemplary embodiment, the motor is an electric motor having a rotor 45 and a stator 46. One of the shaft stubs of one of the rotors 26 and 28 may be coupled to the motor's rotor so as to permit the motor to drive that rotor about its axis. When so driven in an operative first direction about the axis, the rotor drives the other rotor in an opposite second direction. The exemplary housing assembly 22 includes a rotor housing 48 having an upstream/inlet end face 49 approximately midway along the motor length and a downstream/discharge end face 50 essentially coplanar with the rotor body ends 32 and 36. Many other configurations are possible.

[0022] The exemplary housing assembly 22 further comprises a motor/inlet housing 52 having a compressor inlet/suction port 53 at an upstream end and having a downstream face 54 mounted to the rotor housing downstream face (e.g., by bolts through both housing pieces). The assembly 22 further includes an outlet/discharge housing 56 having an upstream face 57 mounted to the rotor housing downstream face and having an outlet/discharge port 58. The exemplary rotor housing, motor/inlet housing, and outlet housing 56 may each be formed as castings subject to further finish machining.

[0023] Surfaces of the housing assembly 22 combine with the enmeshed rotor bodies 30 and 34 to define inlet and outlet ports to compression pockets compressing and driving a refrigerant flow 504 from a suction (inlet) plenum 60 to a discharge (outlet) plenum 62. A series of pairs of male compression pockets 66 and female compression pockets 68 are formed by the housing assembly 22, male rotor body 30 and female rotor body 34. Each compression pocket is bounded by external surfaces of enmeshed rotors, by portions of cylindrical surfaces of male and female rotor bore surfaces in the rotor case, and portions of face 57. The pockets sequentially form, close, compress, and then open to a discharge port in the face 57 along a mesh of the associated rotor pair.

[0024] In the prior art, various mechanisms are used for screw compressor unloading. Poppet and slide valves are used for mechanical unloading whereas variable speed drives are used for unloading via modulation of shaft speed. Slide valves offer improved part load efficiency over poppet valves by providing continuous modulation (vs. step changes in capacity). Variable speed drives provide further improvement over slide valves by extending the range of continuous modulation. The cost of these unloading systems increase along with the improved performance (poppets being lowest cost, then slide valves, then variable speed drives being the highest cost). The exemplary baseline compressor has a slide valve system 70 having a slide valve element 72 driven by a fluidic (e.g. refrigerant) actuator 74.

[0025] FIGS. 2 and 3 show an actuator 100 coupled to the second rotor 28 to provide relative longitudinal movement between the second rotor and a blocking portion of the housing (e.g., the upstream face 57). The exemplary relative movement comprises shifting the second rotor between first and second positions. In the first position/condition (FIG. 2), the normal sealing clearance is provided between the body end/face 36 and discharge housing upstream face 57 so that the face blocks/closes the compression pocket(s). In the second position/condition (FIG. 3), the second rotor 28 is shifted relatively away from the discharge housing to open up a non-sealing clearance gap of thickness T between the body end/face 36 and discharge housing upstream face 57. This unloads opens the compression pocket(s) so that the compressor (e.g., fully unloads).

[0026] Intermediate capacities may be achieved by bistatic modulating between the two positions (e.g., changing the duty cycle under a pulse width modulation type control). The exemplary controller 110 is a microcontroller or computer configured by one or both of hardware and software to provide the duty cycle control to achieve a target capacity. The controller could be specific to the compressor or of a broader system. The controller may determine the target/desired capacity (e.g., as a fraction of full capacity) responsive to sensed parameters (e.g., temperatures at various locations in a refrigeration/cooling system) and/or programmed or user entered parameters (e.g., thermostat settings).

[0027] A basic example is a fixed frequency system wherein the duty cycle is controlled. With an exemplary frequency of 0.05Hz, the cycle period/ (time) is twenty seconds. The duty cycle may be determined as the fraction of the cycle period which the rotor body end is in the engaged second position (or alternatively the disengaged first position).

[0028] More complex modulations may be provided. For example, the modulation frequency may be controlled dynamically ("on the fly") for various performance results. For example, a low frequency may be advantageous to minimize wear and energy consumption of the actuator 100. However, a higher frequency may provide smoother overall refrigerant flow and may reduce variations in motor loading and associated motor wear. To control motor wear, a motor temperature may be directly measured or indirectly measured via a discharge temperature. In such a situation, the control system may be configured to operate at an initial frequency and, thereafter, increase the frequency if motor temperature or other motor loading indication exceeds a desired value. For example, the frequency might be incrementally increased up to a maximum value. For example, starting at an initial value of 0.05Hz, the frequency could be incrementally increased up to an upper limit (e.g., a value of 0.4Hz). Feedback control may reduce the frequency back toward or all the way to the initial low value.

[0029] Also, frequency could be similarly increased if sensed temperature variations (e.g., in the conditioned environment such as a refrigerated compartment or climate controlled room) exceed a desired threshold (ΔT). As with motor load, feedback can decrease the frequency responsive to subsequent decreases in temperature fluctuations.

[0030] Thus, the controller may be configured to modulate the rotor position to provide the target capacity (subject to acceptable deviation) while balancing attributes of low modulation frequency (e.g., actuator wear and energy consumption) against attributes of higher frequency (e.g., motor wear and energy consumption and tolerance of fluctuations).

[0031] In various implementations, a spring 120 may bias the second rotor 28 from the unloaded condition to the loaded condition. Alternatively, the bias (and associated normal/default position) may be reversed. The exemplary spring 120 is a metal tension coil spring located at the discharge end/side.

[0032] Similarly, FIGS. 4 and 5 show first and second positions/conditions of a compressor wherein an actuator 200 and metal compression coil spring 220 are located at the suction end/side. The exemplary spring 220 biases the rotor 28 toward the loaded first position (FIG. 4) from the unloaded second position (FIG. 5). The actuator 200 may pull against the spring bias to shift from the first position/condition to the second position/condition. Modulated operation may be similar to that of the actuator 100 discussed above. Yet alternative push-pull actuators may eliminate a spring bias or supplement the force of a spring bias in the corresponding direction.

[0033] In various implementations, the actuator may be fluidic (e.g. operating using fluid pressure such as from the compressor's lubricant oil recovery system or refrigerant gas from sources at the low and high pressure (suction and discharge) sides of the refrigeration system). Alternative actuators may be electromechanical or electromagnetic. The actuator and spring may cooperate with the rotor via one or more of the bearing systems supporting the rotor.

[0034] In alternative implementations, the actuator may be positioned to shift both rotors (e.g., of a two-rotor compressor). In a three-rotor compressor, the actuator may be positioned to shift the central rotor, the other two rotors, or all three. Depending upon implementation, the actuator may be positioned at either end of the associated rotor(s).

[0035] FIGS. 6-9 show an alternate reengineered compressor 300. The compressor 300 is reengineered from a slightly different baseline compressor than the compressor of FIG. 1. Rather than a shifting of a rotor while the housing remains stationary, the exemplary compressor 300 shifts a blocking portion of the housing while the axial/longitudinal positions of the rotors remain unchanged.

[0036] The compressor 300 has a housing assembly 302 containing a motor 304 driving a male lobed rotor 306 and female lobed rotors 308 and 310 (FIG. 8) having respective central longitudinal axes 510, 512, and 514. The male rotor working portion has a first (upstream/suction) end 320 and a second (downstream/discharge) end 322. Each of the female rotor working portions has a first end 324 and a second end 326. Other details may be similar to that of the compressor 20. The exemplary housing assembly 302 has an inlet port 330 to a suction plenum 332. The housing assembly includes an outlet port 340 from a discharge plenum 342. A check valve 344 may be proximate the outlet port.

[0037] In the exemplary compressor 300, the modification from a baseline condition differs from the FIG. 2-5 modification of the compressor 20. Whereas the FIG. 2-5 modification of compressor 20 adds means for longitudinally shifting one or more rotors, the compressor 300 reflects a reengineering wherein the discharge housing is modified to include a shiftable plate 350. The plate 350 normally seals with the downstream ends of the rotor working portions to define the associated compression pockets. The plate 350 has an upstream face 352 and a downstream face 354. A periphery 356 joins the upstream face 352 and the downstream face 354. The plate 350 and its upstream face 352 serve as the housing blocking portion normally blocking/closing the compression pockets. The plate has a plurality of through apertures. FIG. 7 shows the plate 350 having through apertures 358, 359, and 360 accommodating the downstream/discharge end shaft stubs of the rotors. FIG. 7 further shows the plate as having apertures defining a first discharge port 362 and a second discharge port 364. The first discharge port is positioned to discharge from the compression pocket between the male rotor 306 and the first female rotor 308. The discharge port 364 is positioned to discharge refrigerant from the compression pocket of the male rotor 306 and second female rotor 310.

[0038] The plate 350 may be disengaged from sealing the compression pockets by a longitudinal translation away from the rotors (e.g., to a second (unloading) condition of FIG. 9). A spring 370 within the discharge housing may bias the plate 350 to the first/sealed/loaded condition from the second/unsealed/unloaded condition. Movement beyond the second condition may be restricted such as by a shoulder 372 of the housing. In the reengineering to the configuration of the compressor 300, the discharge housing may be extended along with the discharge end shaft stubs.

[0039] Capacity may be controlled by a modulated shifting of the plate 350 (e.g., between the first (FIG. 6) and second (FIG. 9) conditions (positions)). An exemplary modulation is fluid-controlled. FIG. 6 shows a fluid-actuated shifting mechanism 380 for shifting the valve. The mechanism may be driven by a controller 390 (e.g., similar to controller 110). FIG. 6 further shows a motor temperature sensor 392 and a discharge temperature sensor 394 which may be used by the controller 390 to provide the feedback control over modulation frequency discussed above. The shifting mechanism 380 includes a three-way valve 382. The three-way valve is coupled by a first line (conduit) 384 to a suction condition/location (e.g., to a port 385 at the suction plenum 332). A second line 386 is coupled to a high pressure location (e.g., to ports 387) positioned to intersect the compression pockets right before the compression pockets normally open to the discharge plenum 342. A third line 388 communicates with the discharge plenum 342 (e.g., via a port 389 downstream of the plate downstream face 354).

[0040] To unload the compressor, the valve 382 may be actuated to place the lines 384 and 388 in communication with each other. This communication drops the pressure along the downstream face 354 toward the suction pressure. Meanwhile, the upstream face 352 is still exposed to higher pressure compressed refrigerant in the compression pockets. The pressure differential across the plate 350 will shift the plate 350 from the first condition (FIG. 6) toward the second (FIG. 9) condition (e.g., and compress the spring 370).

[0041] To reload the compressor, the valve 382 is actuated to establish communication between the lines 386 and 388. This more closely balances the pressure forces across the plate 350. This force balance, combined with the bias force of the spring 370, will shift the plate 50 back to the first condition maintain sealing of the plate 350 to the rotors and maintain compression pocket integrity. The spring 370 may also preload the plate 350 and prevent vibration of the plate 350 from partially unloading the compressor when a fully loaded condition is desired. Furthermore, additional damping means may be provided (e.g., a viscous or hydraulic damper (now shown)).

[0042] Various implementations may have one or more of several advantages. For example, there may be an advantageous balance of cost and performance. Continuous control similar to relatively expensive systems (e.g., slide valve or variable speed systems) could be provided at cost similar to relatively inexpensive systems (e.g., poppet valve systems). For example, in a reengineering situation, the reengineered compressor configuration could be less expensive to manufacture than the baseline compressor. Such a reengineering may involve eliminating an unloading valve (e.g., a slide valve) and its associated actuation hardware. Such a reengineering may eliminate variable speed motor control (e.g., by eliminating a variable frequency drive (VFD) also known as a variable speed drive (VSD)). However, although some systems may thus lack an unloading valve and/or lack variable speed motor control, the present features may also be implemented in compressors having one or both of unloading valves and variable speed motor control.

[0043] One or more embodiments have been described. Nevertheless, it will be understood that various modifications may be made. For example, in a reengineering or remanufacturing situation, details of the existing compressor configuration may particularly influence or dictate details of the implementation. Accordingly, other embodiments are within the scope of the following claims.

CLAIMS

What is claimed is:

1. A screw compressor comprising:

a housing (22; 302) having first (53; 330) and second (58; 340) ports along a flowpath;

a first rotor (26; 306) having a lobed body (30) and an axis (500) and mounted to the housing for rotation about said first rotor axis; and

a second rotor (28; 308, 310) having:

a lobed body (34) enmeshed with the first rotor body (30); and

an axis (502), the second rotor mounted to the housing for rotation about said second rotor axis and cooperating with the first rotor (26; 306) and housing (22; 302) to define a compression path between suction (60; 332) and discharge (62; 342) locations along the flowpath,

characterized by:

means (100, 110, 120; 200, 210, 220; 370, 380, 390) for providing relative longitudinal movement between a blocking portion (57; 352) of the housing and at least one of the first rotor and second rotor between:

a first condition wherein a pocket of the first and second rotors is closed by the blocking portion; and

a second condition wherein the blocking portion does not close the pocket; and

a control system (110; 390) configured by one or both of hardware and software to provide duty cycle control of the movement.

2. The compressor of claim 1 wherein:

the means (380, 390) provides longitudinal movement of the blocking portion (352) relative to a remainder of the housing between a first position associated with the first condition and a second position associated with the second condition.

3. The compressor of claim 2 wherein the means comprises:

a spring (370) biasing the blocking portion from the second position toward the first position.

4. The compressor of claim 1 wherein:

the means (100, 110, 120; 200, 210, 220) provides longitudinal movement of a single one of the first rotor and second rotor relative to the blocking portion (57).

5. The compressor of claim 1 wherein:

the means provides longitudinal movement of at least a movable rotor (28) of the first and second rotors between first and second positions; and

the means comprises an actuator (100; 200) coupled to at least the movable rotor of the first and second rotors to shift the movable rotor.

6. The compressor of claim 5 wherein:

a spring (120; 220) biases the movable rotor from the second position toward the first position.

7. The compressor of claim 1 wherein:

the compressor lacks an unloading valve.

8. The compressor of claim 1 wherein:

the compressor lacks variable speed motor control.

9. The compressor of claim 1 wherein:

the control system is configured to vary a frequency of the duty cycle control responsive to a condition of a motor of the compressor.

10. The compressor of claim 1 wherein:

the control system is configured to vary a frequency of the duty cycle control responsive to a condition of a motor of the compressor and a fluctuation in a sensed temperature in a conditioned environment.

11. A method for operating the compressor of claim 1, the method comprising:

determining a desired loading state; and

duty cycle modulating of the means to provide the desired loading state.

12. The method of claim 11 further comprising:

altering a frequency of the duty cycle modulating responsive to a loading condition of a motor of the compressor.

13. The method of claim 11 further comprising:

altering a frequency of the duty cycle modulating responsive to a sensed temperature fluctuation of a conditioned environment.

14. The method of claim 11 further comprising:

varying a frequency of the duty cycle modulating responsive to a sensed temperature of a motor of the compressor.

15. A screw compressor comprising:

a housing (302) having first (330) and second (340) ports along a flowpath;

a first rotor (306) having a lobed body and an axis (510) and mounted to the housing for rotation about said first rotor axis; and

a second rotor (308; 310) having:

a lobed body enmeshed with the first rotor body; and

an axis (512; 514), the second rotor mounted to the housing for rotation about said second rotor axis and cooperating with the first rotor (306) and housing (302) to define a compression path between suction (332) and discharge (342) locations along the flowpath,

wherein:

a blocking portion (352) of the housing is mounted for longitudinal movement of the blocking portion (352) relative to a remainder of the housing between a first position engaging the lobed bodies to define at least one compression pocket and a second position retracted from the first position.

16. The compressor of claim 15 further comprising:

an actuator (380) coupled to blocking portion to shift the blocking portion between the first and second positions.

17. The compressor of claim 15 further comprising:

means (380, 390) for shifting the blocking portion between the first and second positions.

18. The compressor of claim 15 wherein:

a spring (370) biases the blocking portion from the second position toward the first position.

19. The compressor of claim 15 wherein:

the first position is relatively toward a section end of the housing; and

a spring (370) biases the movable rotor from the second position toward the first position.

20. The compressor of claim 15 wherein:

the actuator is fluid-operated.

21. The compressor of claim 20 wherein:

the fluid is refrigerant.

22. The compressor of claim 15 further comprising:

a controller (390) coupled to the actuator and configured to provide capacity control of the compressor by modulating the blocking portion between the first and second positions.

23. The compressor of claim 15 wherein:

the compressor lacks an unloading valve.

24. The compressor of claim 15 wherein:

the compressor lacks variable speed control.

25. A method for operating the compressor of claim 15 comprising:

determining a desired loading state; and

duty cycle modulating an actuator (380) of the movement between the first and second positions to provide the desired loading state.

26. The method of claim 25 wherein:

the duty cycle modulating is performed while operating a motor of the compressor at an essentially fixed speed.

27. The method of claim 25 wherein:

the duty cycle modulating is performed via a controller and the actuator fluidically operates to provide the desired loading state.

28. The method of claim 25 wherein:

the modulating of the actuator (380) operates against a bias spring (370).

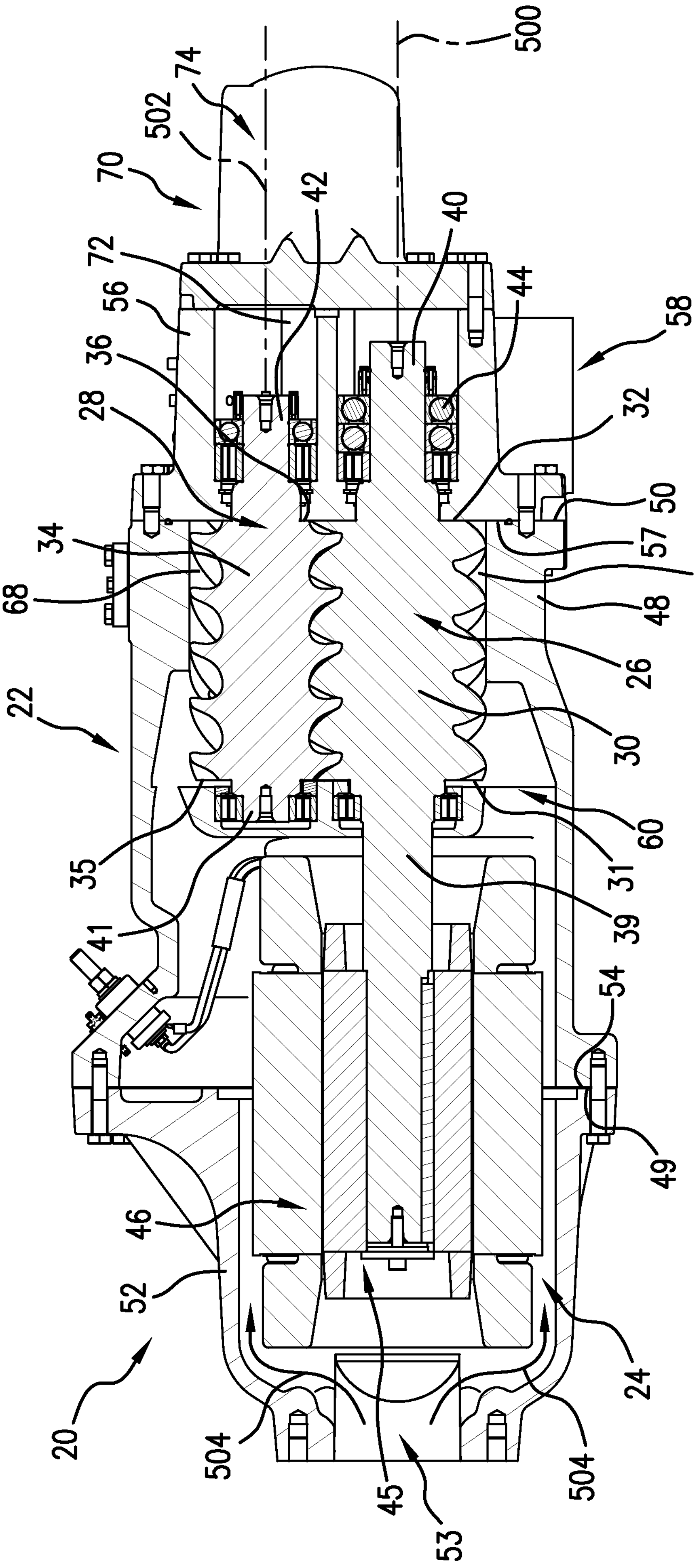
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FIG.1

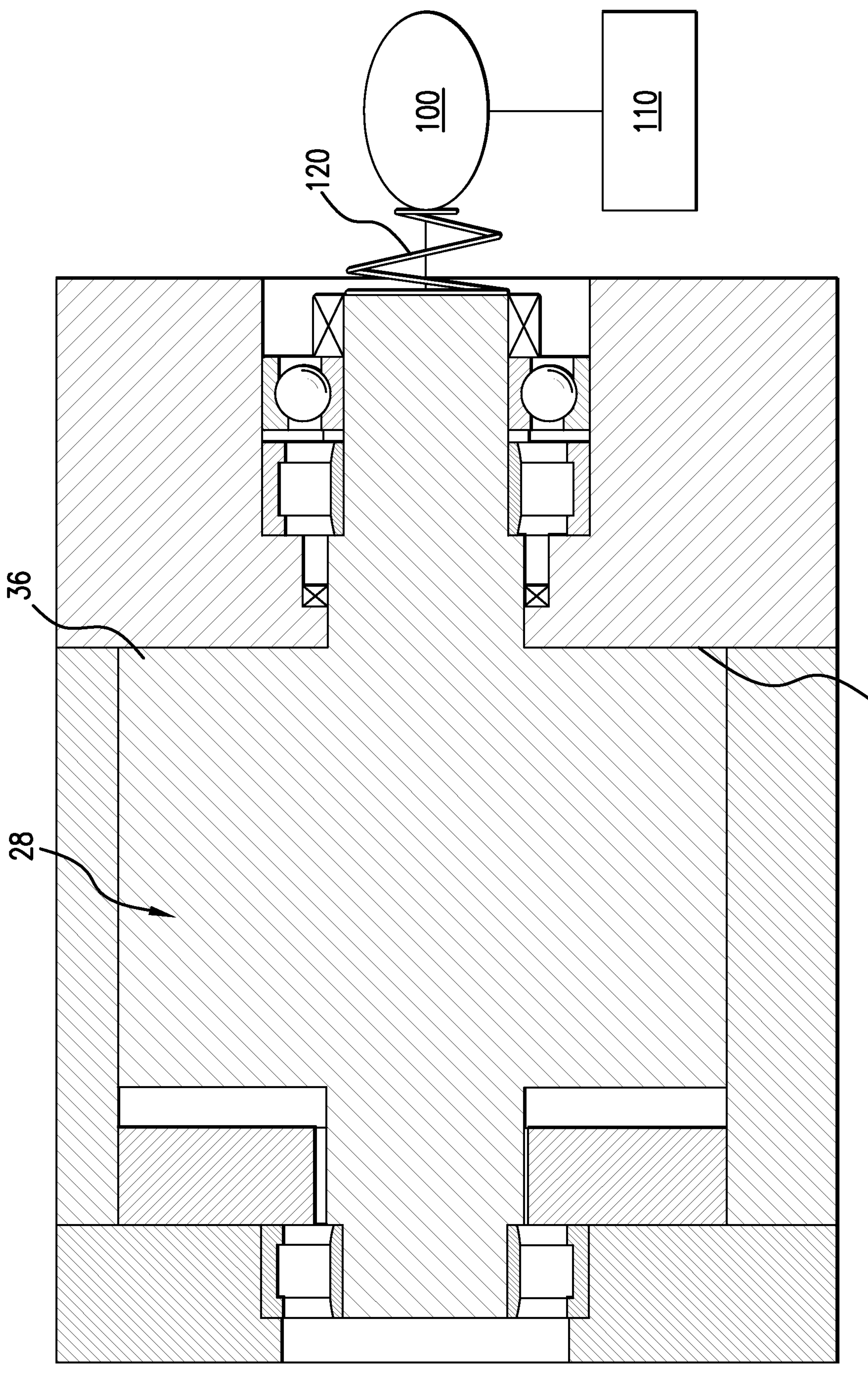

FIG.2

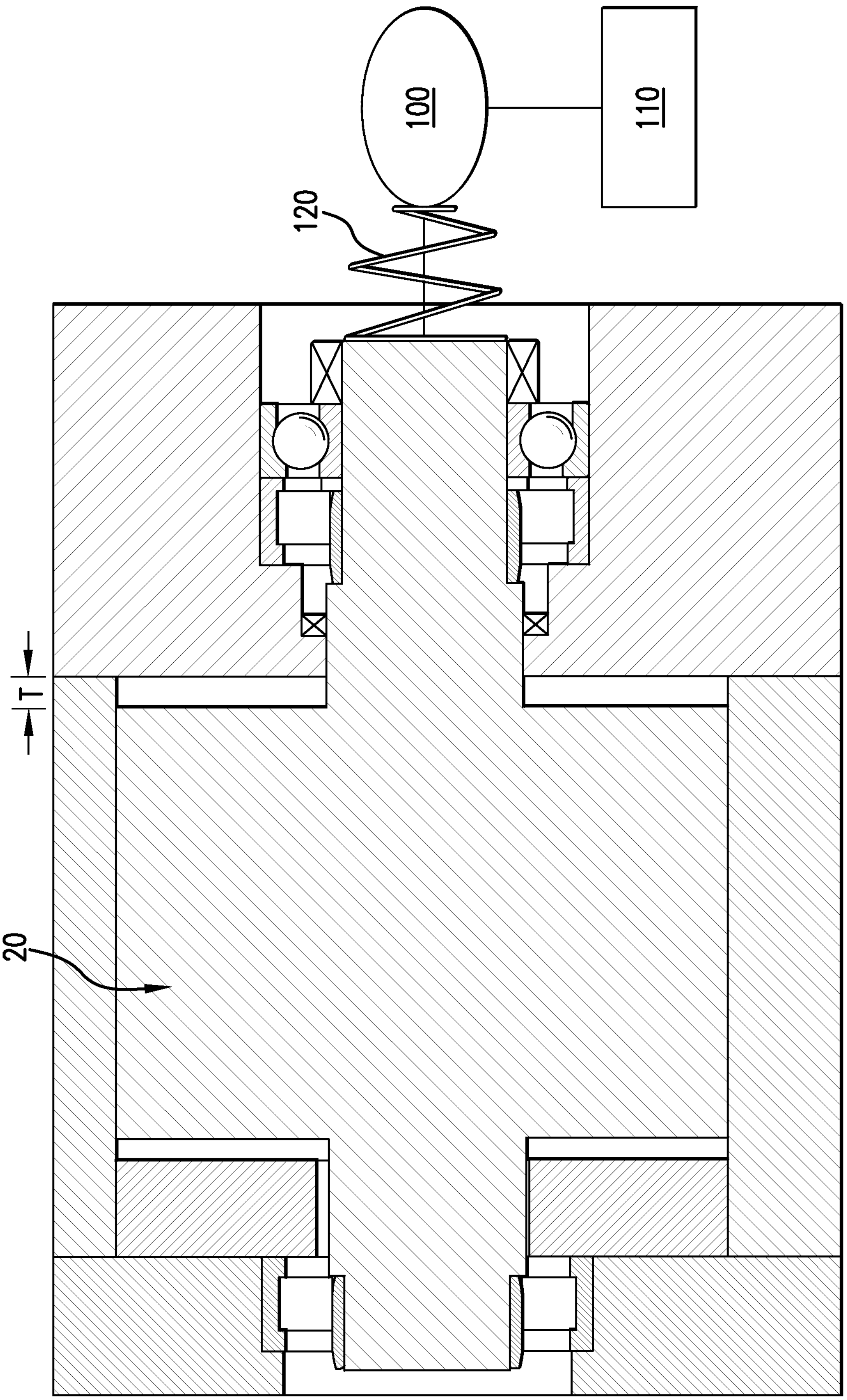

FIG.3

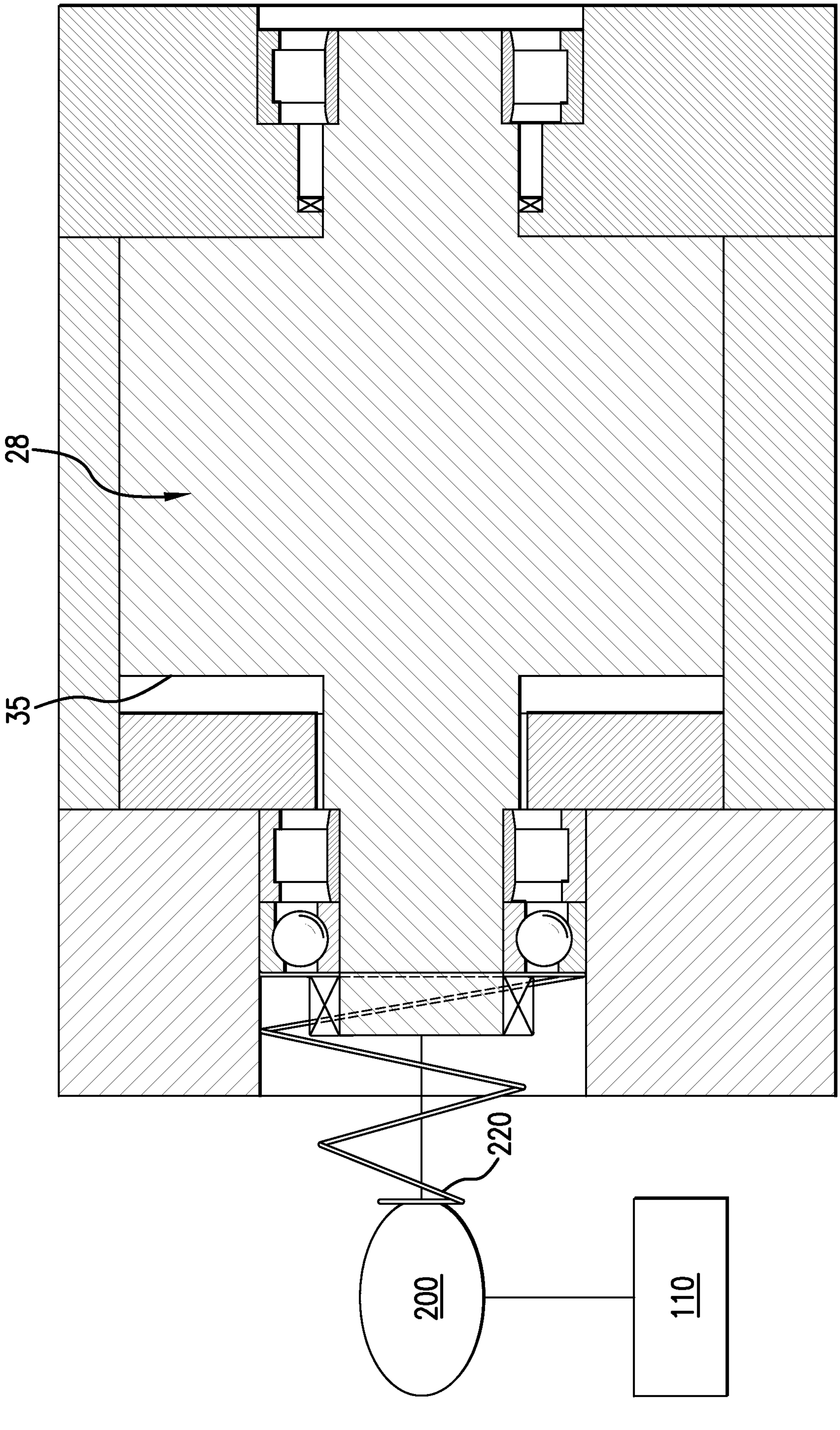

FIG.4

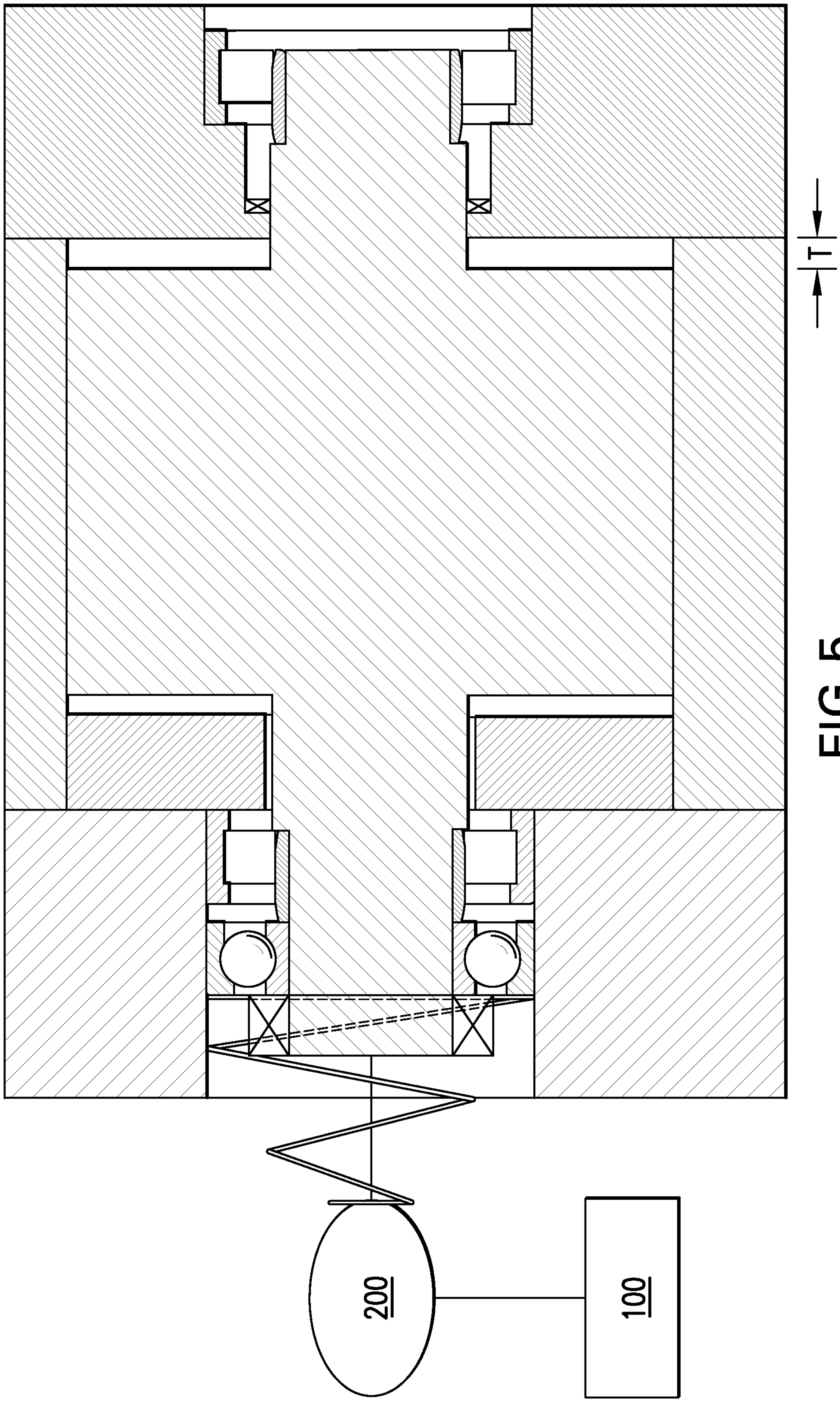

FIG.5

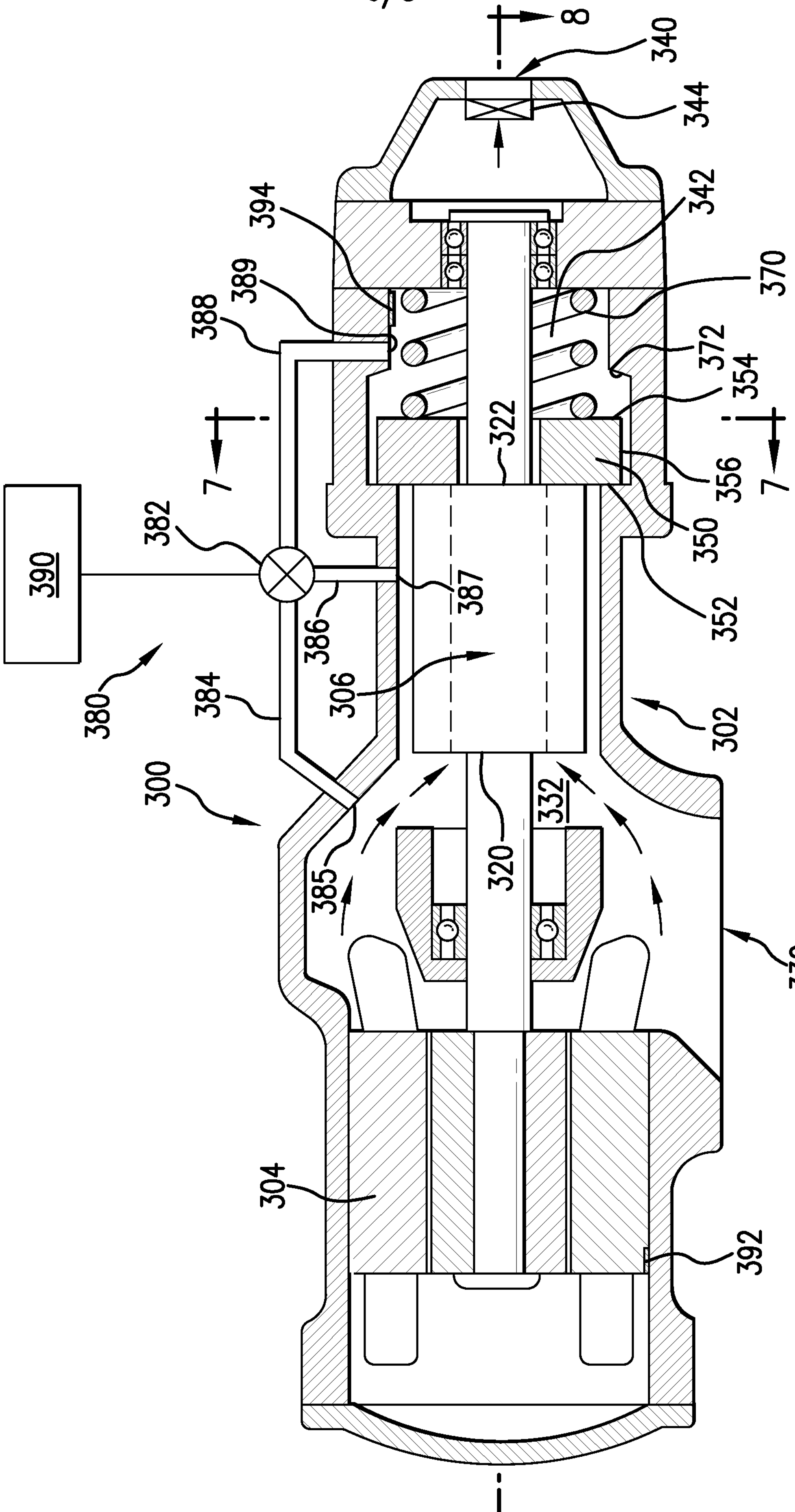
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FIG.6

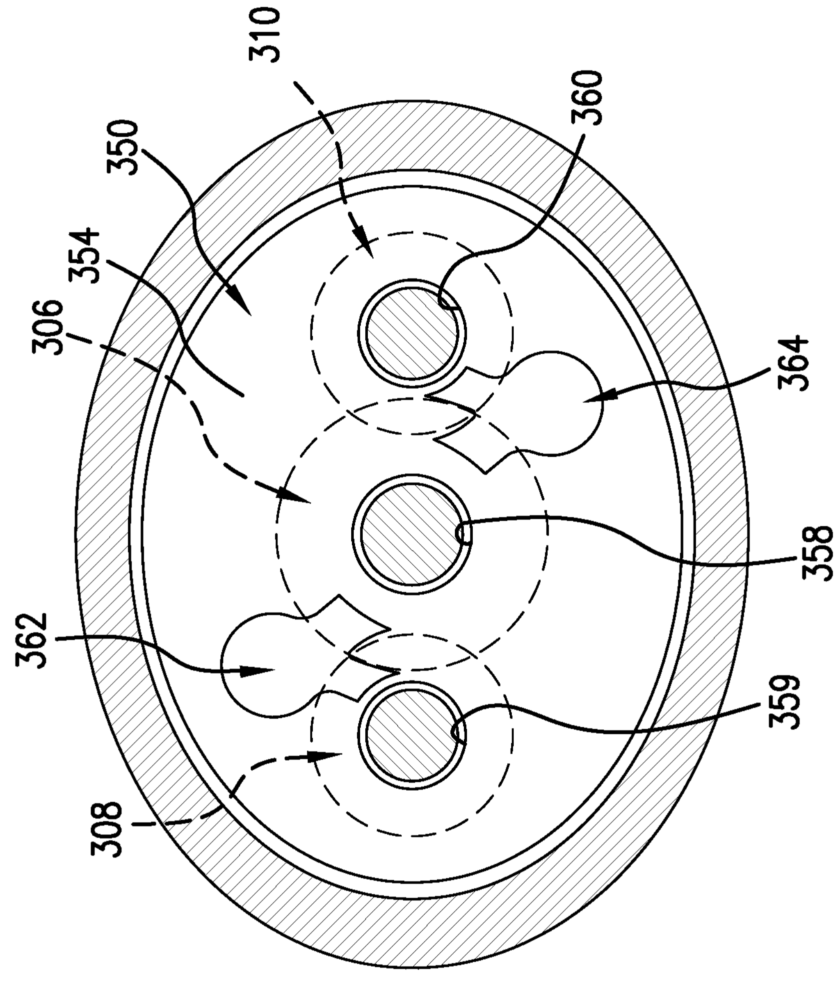

FIG. 7

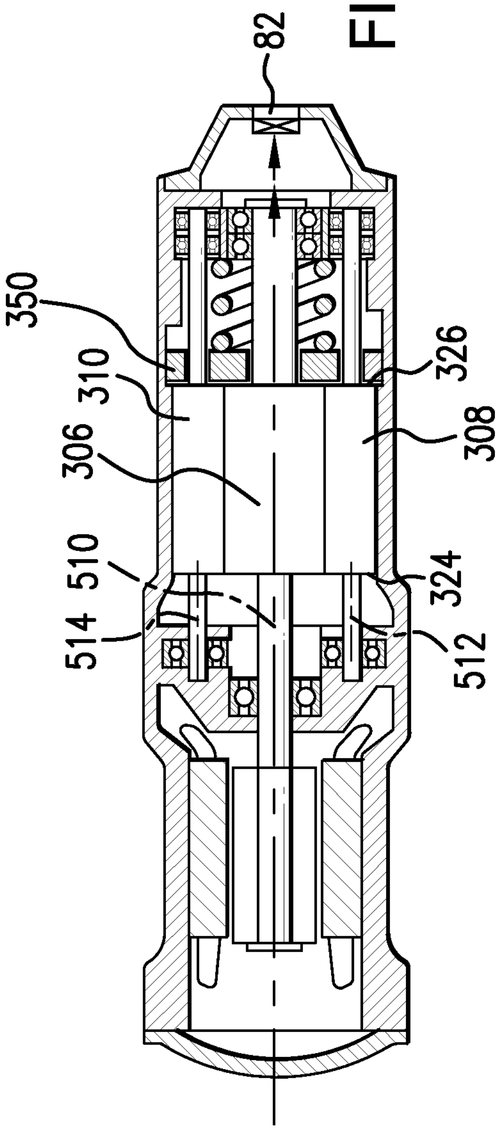

FIG.8

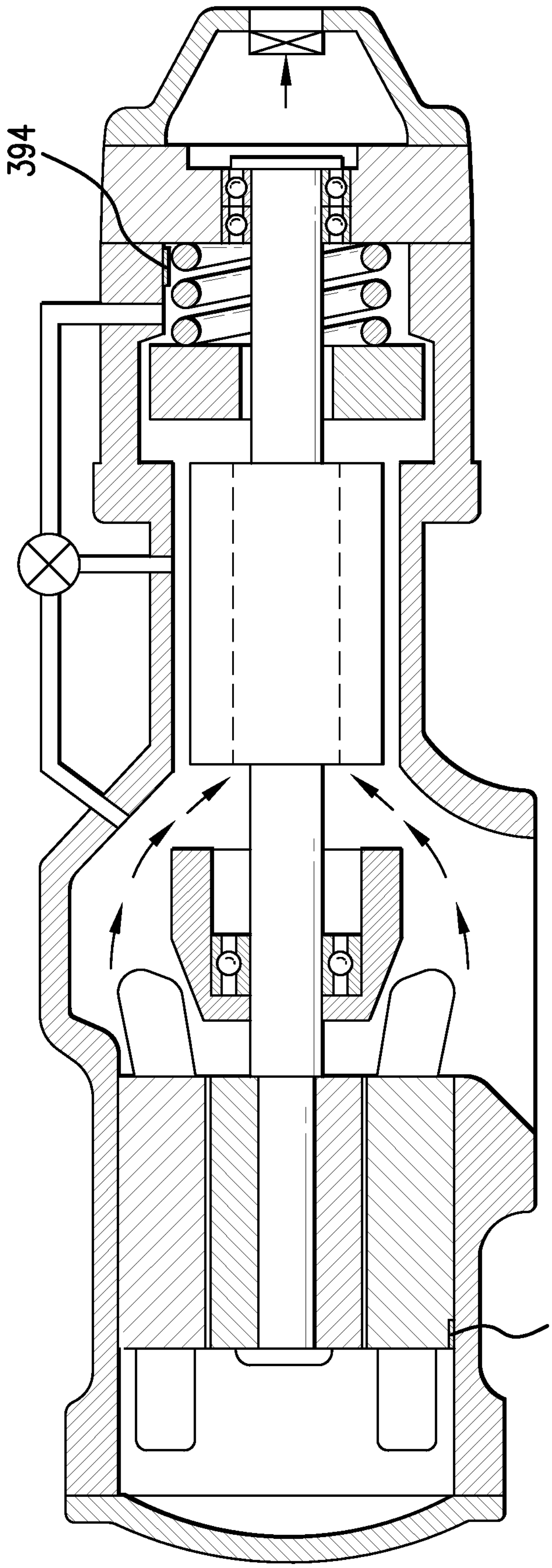

FIG.9

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US2007/074548

A. CLASSIFICATION OF SUBJECT MATTER

F04C 18/16(2006.01)i, F04C 28/00(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 8 : F04C 18/16, F04C*, F04B*, F25B*

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
Korean utility models and applications for utility models since 1975
Japanese utility models and applications for utility models since 1975

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
eKIPASS (KIPO internal) & keywords: screw, compressor, control*, capacity

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A	US 6,003,324 A (SHAW) December 21, 1999 See Abstract, figure 3, column 6 line 54 to column 7 line 16, and claim 1	1, 4, 5, 11, 12 2, 3, 6 - 10, 13 - 28
X A	JP 61- 093294 A (DAIKIN IND LTD) May 20, 1986 See Abstract, figures and claims	1, 4, 5 2, 3, 6 - 28
A	US 5,211,026 A (LINNERT) May 18, 1993 See Abstract and claims	1 - 28

☐ Further documents are listed in the continuation of Box C.   ☒ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
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"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents,such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
30 NOVEMBER 2007 (30.11.2007)	**30 NOVEMBER 2007 (30.11.2007)**

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No.

PCT/US2007/074548

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 6,003,324 A	21. 12. 1999	None	
JP 61- 093294 A	20. 05. 1986	None	-
US 5,211,026 A	18. 05. 1993	CA2074444AA	20 .02. 1993
		JP05215086 A	24. 08. 1993
		KR1019940701102 A	22. 04. 1994