



US 20070286751A1

(19) **United States**(12) **Patent Application Publication**
Manole et al.(10) **Pub. No.: US 2007/0286751 A1**(43) **Pub. Date: Dec. 13, 2007**(54) **CAPACITY CONTROL OF A COMPRESSOR****Publication Classification**(75) Inventors: **Dan M. Manole**, Tecumseh, MI (US);
Phillip A. Tomell, Adrian, MI (US)(51) **Int. Cl.**
F04B 35/04 (2006.01)(52) **U.S. Cl.** **417/417**

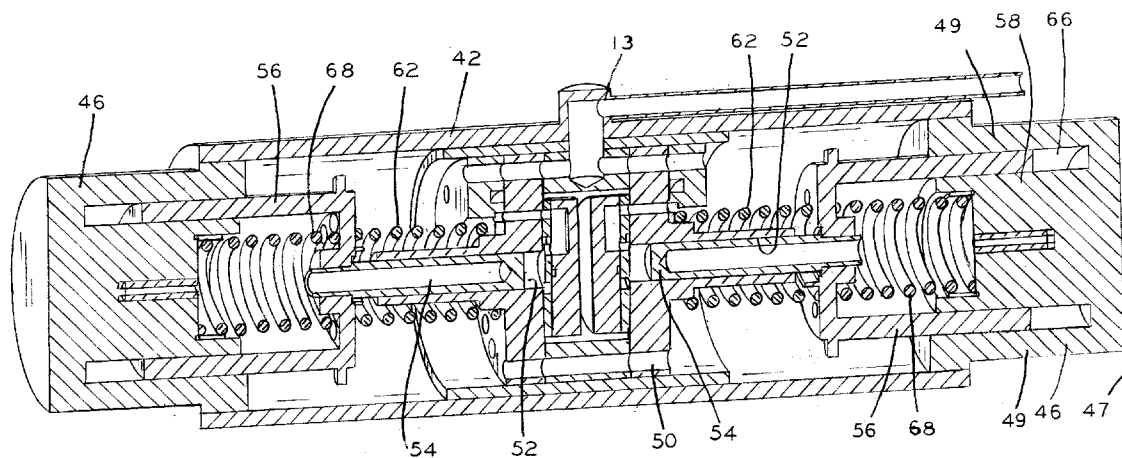
Correspondence Address:

BAKER & DANIELS LLP
111 E. WAYNE STREET
SUITE 800
FORT WAYNE, IN 46802 (US)(73) Assignee: **TECUMSEH PRODUCTS COMPANY**, Tecumseh, MI (US)(21) Appl. No.: **11/759,295**(22) Filed: **Jun. 7, 2007****Related U.S. Application Data**

(60) Provisional application No. 60/812,890, filed on Jun. 12, 2006.

(57) **ABSTRACT**

A linear compressor that is operated at a frequency greater than the natural frequency of the spring-mass system of the compressor. Operating the compressor at such a frequency can increase the output of the compressor. In one embodiment, the linear compressor includes a cylinder block having a cylinder bore, a piston positioned within the cylinder bore, first and second springs for positioning the piston where the piston and the first and second springs comprise the spring-mass system, and an armature operably engaged with the piston to drive the piston at a frequency greater than the natural frequency of the spring-mass system. The linear compressor can also include a controller which monitors the instantaneous natural frequency of the spring-mass system and modulates the frequency of the current passing through the armature such that it exceeds the natural frequency of the spring-mass system.



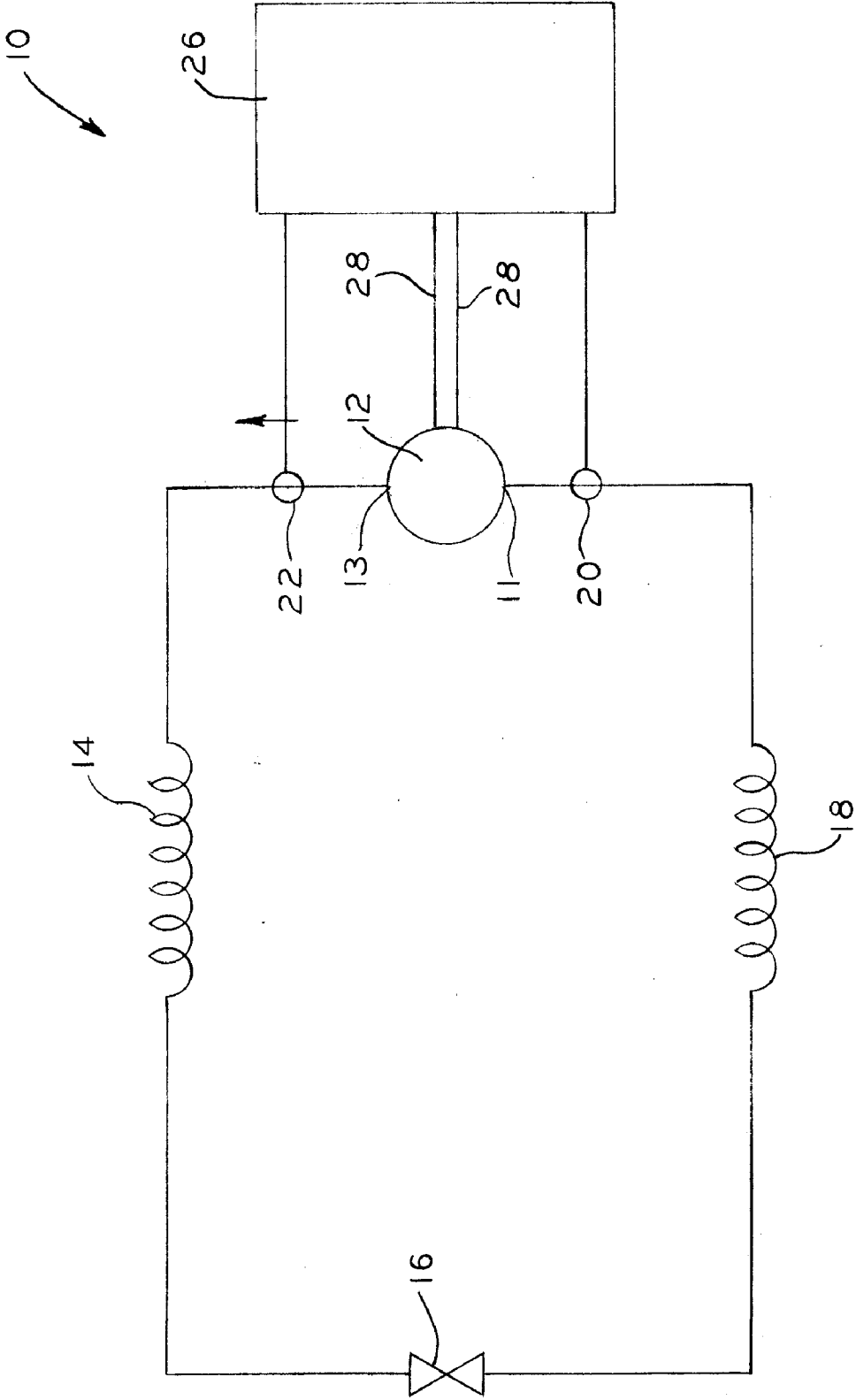


FIG. 1

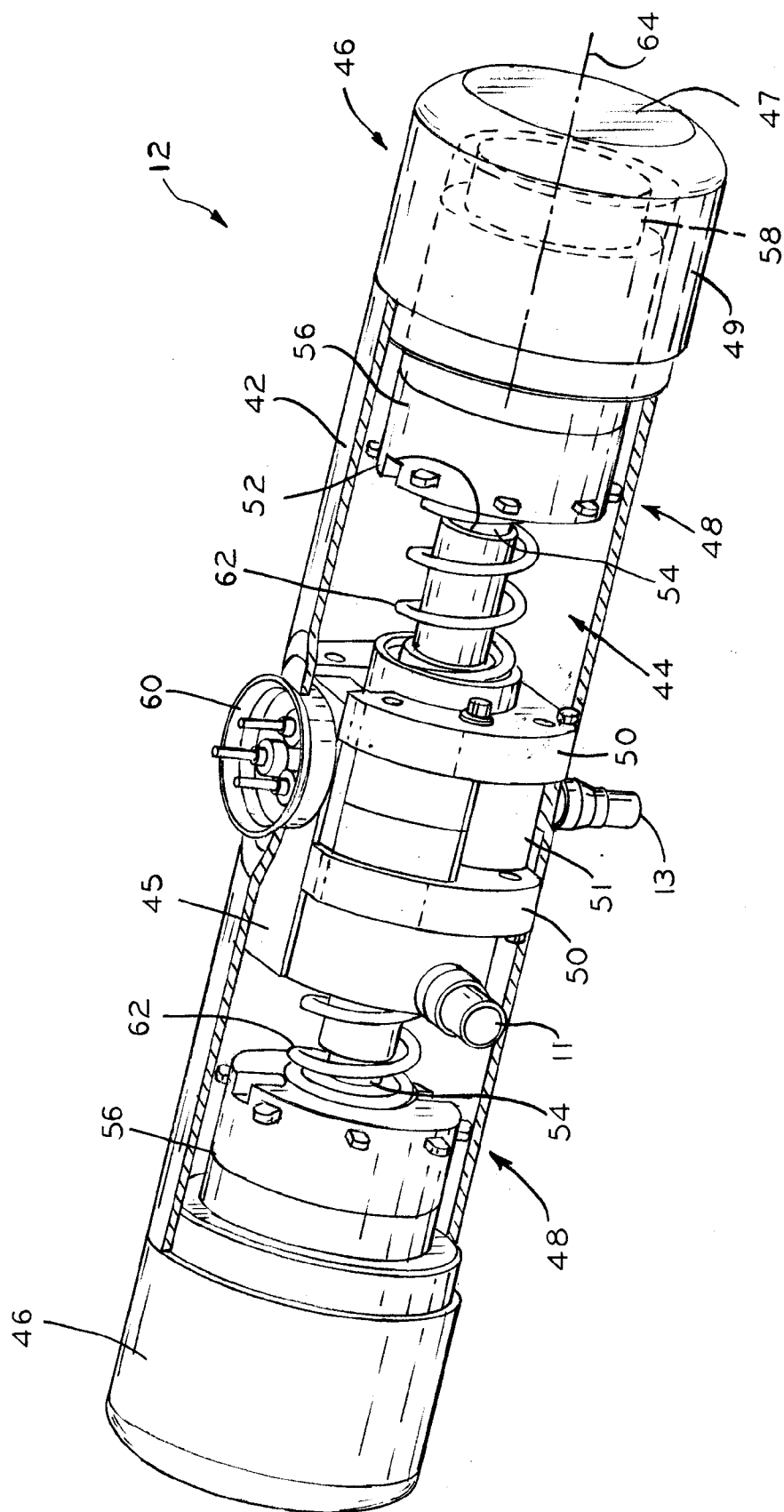


FIG. 2

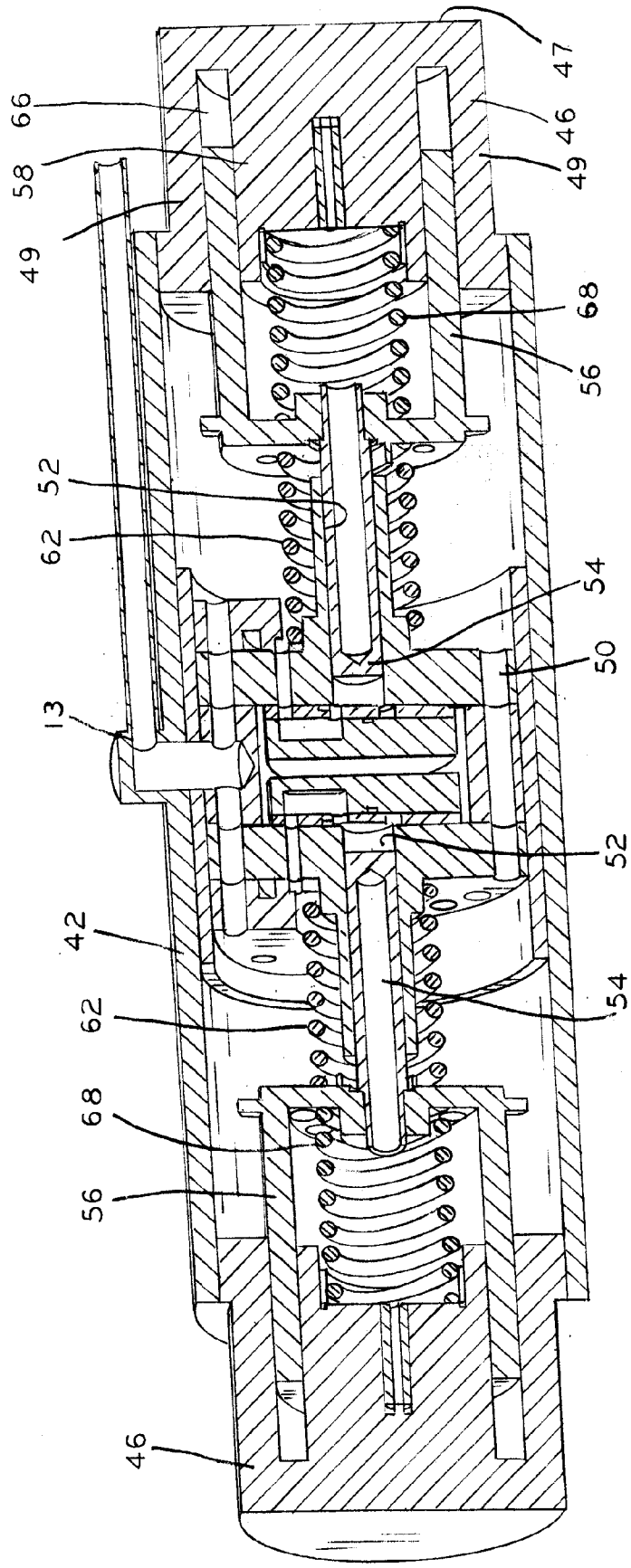
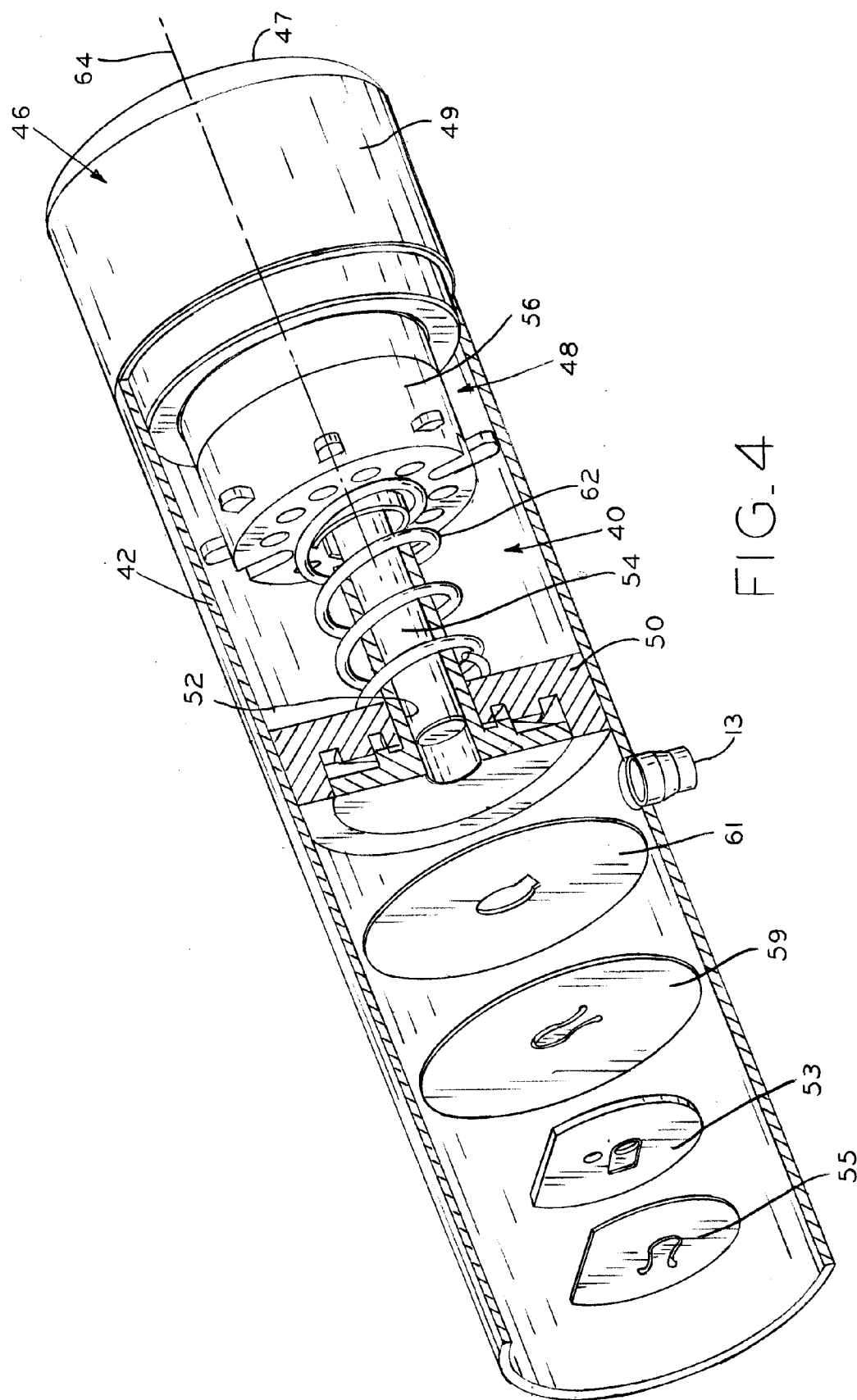


FIG. 3



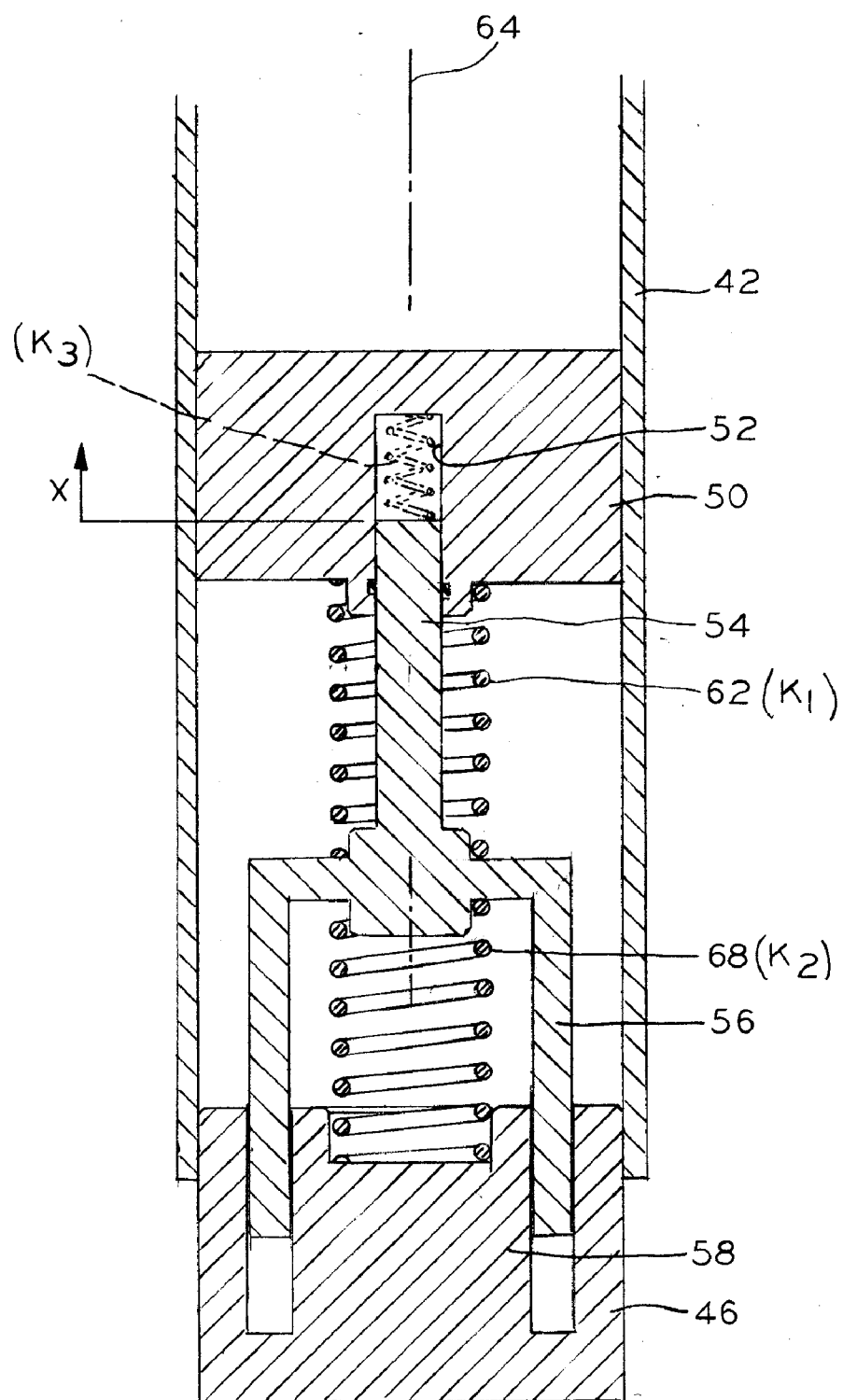


FIG. 5

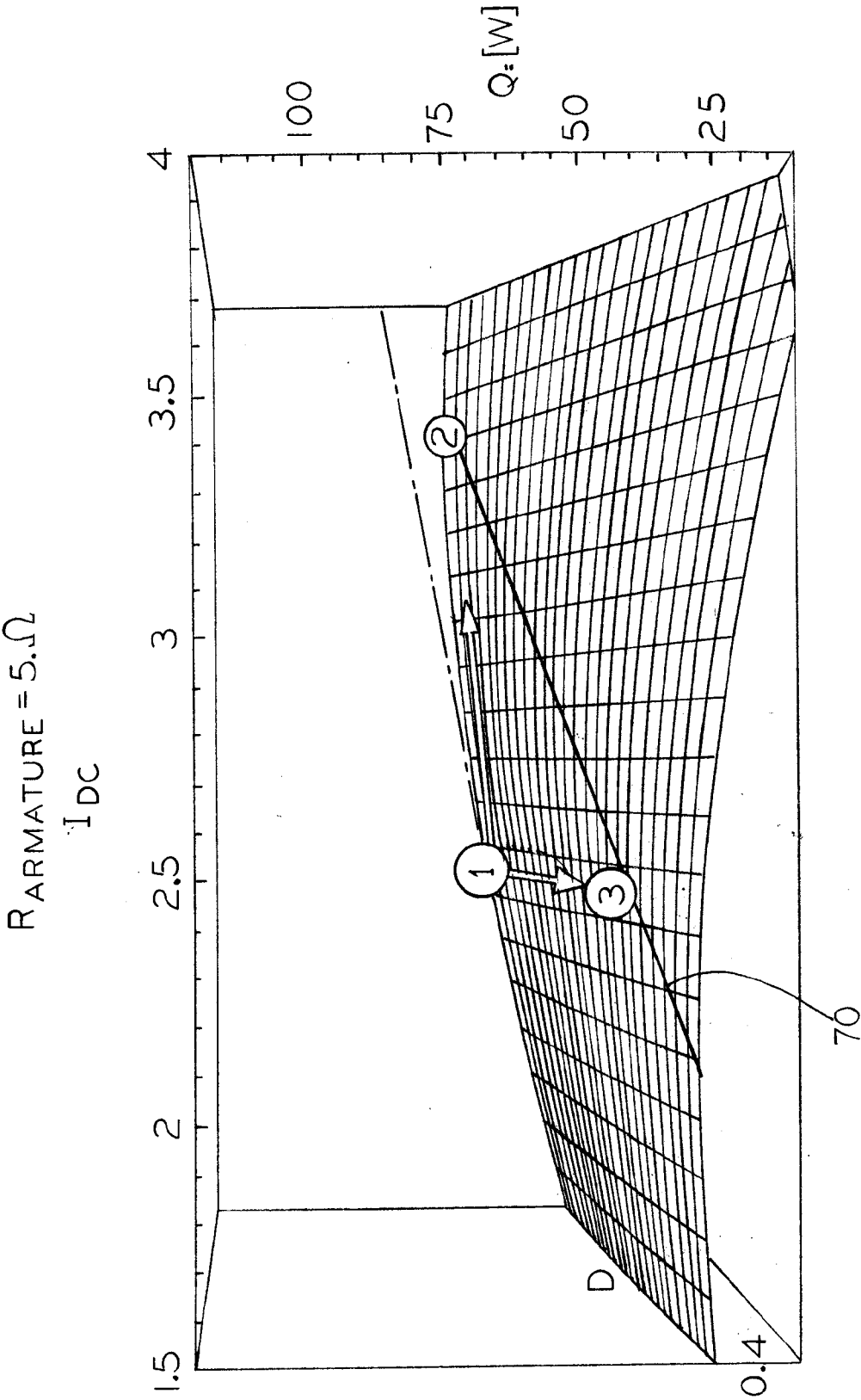


FIG. 6

CAPACITY CONTROL OF A COMPRESSOR

BACKGROUND OF THE INVENTION

[0001] 1. Field of the Invention

[0002] The present invention relates to compressors, and more particularly, to the capacity control of linear compressors.

[0003] 2. Description of the Related Art

[0004] Compressors can include a piston which is reciprocated within a cylinder bore to compress refrigerant, for example, in the cylinder bore. The compressor can further include a spring, or springs, which bias the piston into position. In some linear compressors, the piston is positioned intermediate two springs which hold the piston in a substantially stationary position until the piston is moved by an electromagnetic armature or motor, for example. The piston and springs comprise a spring-mass system having a natural, or resonant, frequency, as known in the art. If the piston is driven, via the armature or motor, at the natural frequency of the spring-mass system, the spring-mass system will resonate. Driving the piston of the compressor at, or very close to, the natural frequency of the system allows the compressor to operate more efficiently. In effect, when the spring-mass system is driven at, or close to, its natural frequency, the driving force has less inertial forces in the system to overcome.

[0005] In view of the above, previous compressors were typically operated at the natural frequency of their spring-mass systems. To increase or decrease the capacity of these compressors, the displacement, or stroke, of the piston was adjusted to change the output of the compressor. For example, if a greater capacity was needed, the stroke of the piston was increased to draw in, compress, and discharge a larger quantity of refrigerant per stroke. To increase the stroke of the piston, the magnitude of the current flowing through the armature was increased, thereby causing a greater displacement between the piston and the armature. However, modulating the capacity of the compressor in this way has some limitations. For example, increasing the magnitude of the current flowing through the armature can increase the resistance losses in the armature windings, thereby reducing the efficiency of the compressor. Further, large displacements of the piston draws large quantities of refrigerant into the cylinder bore which may bog down or overpower the compressor.

[0006] Previously, as discussed above, it was desirable to operate linear compressors at the natural frequency of their spring-mass system. However, owing to changes in the parameters of the refrigerant in the cylinder bore, the natural frequency of the spring-mass system can change throughout the operation of the compressor. More specifically, when the refrigerant is compressed by the piston in the cylinder bore, the refrigerant gas acts as an elastic spring force against the piston. The magnitude of this elastic force depends on, among other things, the fluid being compressed and its density, pressure, and temperature. As known in the art, the magnitude of the spring force from the refrigerant gas affects the natural frequency of the spring-mass system, and, when the parameters of the refrigerant change, the natural frequency of the spring-mass system typically changes as well. In order to determine the natural frequency of the spring-

mass system at any instant during the operation of the compressor, a parameter, or parameters, of the refrigerant and/or refrigeration system can be monitored. For example, it was known to monitor temperature of the refrigerant and/or the voltage drop across the armature driving the piston of the compressor. In view of the information obtained from monitoring these parameters, the frequency of the driving force acting on the piston was altered to match the instantaneous natural frequency of the system.

[0007] In effect, some previous compressors actively monitored the natural frequency of the spring-mass system and corrected the frequency of the driving force to match the natural frequency of the system. However, when these compressors were required to produce a greater output of compressed refrigerant, their output was limited to that generated at the natural frequency of the compressor. As a result, as discussed above, these compressors were sometimes unable to keep up with the demands of the refrigeration system. To accommodate a potentially greater demand, a compressor having a larger capacity was typically used. However, these larger-capacity compressors are typically more expensive and may become less efficient when lower demands of the compressor are required. What is needed is an improvement over the foregoing.

SUMMARY OF THE INVENTION

[0008] The present invention includes a linear compressor that is operated at a frequency greater than the natural frequency of the spring-mass system of the compressor. Operating the compressor at such a frequency can increase the output of the compressor. In one embodiment, the linear compressor includes a cylinder block having a cylinder bore, a piston positioned within the cylinder bore, first and second springs for positioning the piston where the piston and the first and second springs comprise a spring-mass system having a natural frequency, and an armature operably engaged with the piston to drive the piston at a frequency greater than the natural frequency of the spring-mass system.

[0009] In another embodiment, the linear compressor includes a controller which monitors the instantaneous natural frequency of the spring-mass system and modulates the frequency of the current passing through the armature. As discussed above, the natural frequency of the spring-mass system can change as a result of fluctuations in the temperature and/or pressure of the refrigerant in the cylinder bore. In this embodiment, a parameter of the refrigerant in the refrigerant circuit, such as the pressure and/or temperature of the refrigerant, for example, or the electrical power transmitted to the armature, such as the voltage and/or current, for example, is monitored by the controller. In view of the information obtained from monitoring these parameters, the controller can determine the instantaneous natural frequency of the spring-mass system and evaluate whether the frequency of the current being transmitted to the armature is greater than the instantaneous natural frequency of the system. If necessary, the controller can increase the frequency of the current such that it exceeds the natural frequency of the spring-mass system, or, even if the driving frequency is already greater than the natural frequency, it can increase the driving frequency to increase the output of the compressor to meet the demands of the refrigeration system.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010] The above-mentioned and other features and objects of this invention will become more apparent and the invention itself will be better understood by reference to the following description of an embodiment of the invention taken in conjunction with the accompanying drawings, wherein:

[0011] FIG. 1 is a schematic of a typical refrigeration circuit including a compressor and a controller for operating the compressor;

[0012] FIG. 2 is a partial cut-away view of a linear compressor in accordance with an embodiment of the present invention;

[0013] FIG. 3 is a cross-sectional perspective view of a first alternative embodiment linear compressor;

[0014] FIG. 4 is an exploded cross-sectional view of a second alternative embodiment linear compressor;

[0015] FIG. 5 is a schematic representing the spring-mass system of the compressor of FIG. 2; and

[0016] FIG. 6 is a graph charting the cooling capacity of a linear compressor with respect to the current flowing through the armature of the compressor.

[0017] Corresponding reference characters indicate corresponding parts throughout the several views. Although the exemplifications set out herein illustrate embodiments of the invention, the embodiments disclosed below are not intended to be exhaustive or to be construed as limiting the scope of the invention to the precise form disclosed.

DETAILED DESCRIPTION

[0018] Referring to FIG. 1, typical refrigeration system 10 includes, in serial order, compressor 12, condenser 14, expansion device 16, and evaporator 18 connected in series by fluid conduits. As is well known in the art, compressor 12 draws a refrigerant or working fluid through compressor inlet 11, compresses the refrigerant, and expels the compressed refrigerant through compressor outlet 13. The refrigerant expelled from compressor 12 is communicated into condenser 14 where thermal energy of the refrigerant is dissipated. Subsequently, the cooled, compressed refrigerant is communicated to expansion device 16 where it is decompressed. The cooled, low-pressure refrigerant is then communicated to evaporator 18 where the refrigerant in evaporator 18 draws heat from an environment surrounding the evaporator. Subsequently, the refrigerant exits evaporator 18 and is communicated to compressor 12 and the cycle described above is repeated.

[0019] Referring to FIG. 2, compressor 12, in the present embodiment, is a dual-cylinder linear compressor having two axially-driven compressor mechanisms 48 mounted therein. Compressor 12 further includes housing 42 having interior cavity 44 and end caps 46 on opposite ends thereof which also define cavity 44. Generally, in operation, refrigerant is drawn into compressor 12 through suction inlet 11 and suction manifold 45, compressed by compressor mechanisms 48, and is then discharged into discharge muffler chamber 51 through discharge valves 55. Referring to FIG. 4, which illustrates an alternative embodiment of a linear compressor, each compressor mechanism 48 can include

gasket 61, suction valve 59, valve plate 53, and discharge valve 55 for controlling the flow of suction refrigerant into, and the flow of discharge refrigerant out of, the compression cylinder of compressor mechanism 48. Thereafter, the compressed refrigerant is discharged from compressor 12 through discharge outlet 13.

[0020] Each compressor mechanism 48 includes a cylinder block 50 having cylinder bore 52 therein, a piston 54 positioned within cylinder bore 52, an armature 56 mounted to one end of piston 54, and a permanent magnet 58 positioned within end cap 46. In operation, piston 54 is reciprocatingly driven within cylinder bore 52 by the interaction of armature 56 and permanent magnet 58. More particularly, armature 56 is energized by an electrical source which conducts electricity to armature 56 through terminal cluster 60 and spring 62 positioned intermediate cylinder block 50 and armature 56. Armature 56 includes a series of copper windings, or coils, which are, in this embodiment, arranged in a cylindrical configuration. The cylindrical configuration of armature 56 is sized and configured to fit in gap 66 defined between permanent magnet 58 and end cap 46 so that relative movement of armature 56 therebetween is possible. Owing to a magnetic field created by permanent magnet 58, armature 56, when energized, is motivated to move axially along axis 64.

[0021] Permanent magnet 58, as is known in the art, contains two poles of opposite polarity which create the above-mentioned magnetic field. The magnetic field of permanent magnet 58 radiates through bottom 47 of end cap 46, through side walls 49 of end cap 46, and then to the other pole of permanent magnet 58 through annular air gap 66 between end cap 46 and permanent magnet 58. Stated in another way, the magnetic field extends through gap 66 in a radial direction, i.e., a direction substantially perpendicular to axis 64. As the coils of armature 56 are positioned in gap 66, the magnetic field crosses the coils and interacts with the current flowing through the coils to generate Lorentz forces that will move armature 56 in a direction perpendicular to the electrical current and the magnetic field, i.e., along axis 64. By alternating the current polarity, the direction of the axial force acting on armature 56 can be changed to reciprocate armature 56, and piston 54 attached thereto, along axis 64.

[0022] In the present embodiment, the armature is mounted on the piston and the stationary permanent magnet is mounted in the housing. However, in other embodiments, the permanent magnet may be mounted on the reciprocating piston and the armature may be stationary within the compressor.

[0023] As discussed above, compressor mechanism 48 includes spring 62 positioned between armature 56 and cylinder block 50. Compressor mechanism 48 further includes second spring 68 positioned between armature 56 and permanent magnet 58 positioned in end cap 46. Springs 62 and 68 act to hold armature 56, and piston 54 mounted thereto, in a substantially stationary position until the coils of armature 56 are energized. Also, spring 68 completes the electrical circuit between terminal cluster 60, spring 62 and armature 56, as described above. Once energized, one of springs 62 and 68, depending on the polarity of the current, is compressed by the Lorentz forces acting on armature 56 placing piston 54 in one of a top-dead-center (TDC) position

or a bottom-dead-center (BDC) position. The TDC and BDC positions define the limits of the stroke of piston 54 within cylinder bore 52, however, the distance between the TDC and BDC positions is dependent upon the root mean square average value (RMS), or magnitude, of the current passing through the armature. For example, the TDC and BDC positions are further apart from each other when the RMS of the current passing through the armature is increased, and, as a result, the TDC and BDC positions define a longer stroke of the piston and a potentially larger output of refrigerant.

[0024] Referring to FIG. 5, piston 54 and springs 62 and 68 approximate a spring-mass system. Generally, a spring-mass system represents a harmonic system that satisfies the second order differential equation: $x = A \sin \omega_0 t + B \cos \omega_0 t$, where x represents the displacement of piston 54 and ω_0 represents the circular natural frequency, where $\omega_0 = (k/m)^{0.5}$ and is typically measured in radians per second. The constant k represents the spring constant of the spring-mass system, including, in the present embodiment, the spring constants of springs 62 and 68, and the constant m represents, in the present embodiment, the combined mass of piston 54, armature 56, and $1/3$ of the mass of springs 62 and 68. In the foregoing equation, A and B are determined by an initial driving input into the system. The natural frequency of this spring mass system is determined by the following equation: $f = \omega_0 / (2\pi) = ((k/m)^{0.5}) / (2\pi)$. When the spring-mass system is driven by a force having a frequency matching, or nearly matching, the natural frequency of the spring-mass system, the system will resonate. In resonance, the piston of the spring-mass system will have less inertia in the system to overcome and, as a result, less power is required to operate the compressor. Accordingly, compressor manufacturers previously designed their linear compressors to operate at the natural frequency of the linear compressor's spring-mass system in order to utilize this phenomenon.

[0025] To increase the output of these previous compressors, the RMS of the current flowing through the armature is increased while the frequency of the current is held at the natural frequency. Increasing the RMS of the current causes the piston and armature assembly to displace through a greater distance, thereby increasing the stroke and output of the compressor. However, the stroke of the piston is ultimately limited by the length of the cylinder bore and, thus, some adjustments to the compressor capacity may not be possible. Further, by increasing the stroke of the piston, a greater quantity of refrigerant enters into the cylinder bore per stroke which may be difficult for the compressor to compress, thereby bogging down the compressor. In addition, increasing the RMS of the current flowing through the armature can increase the resistance losses in the armature windings, thereby reducing the efficiency of the compressor, as illustrated in the following example. Referring to FIG. 6, the operating condition of a compressor is represented by point 1. Notably, an increase in the RMS of the current, I_{DC} , increases the cooling capacity, Q , of the compressor as long as the operating condition of the compressor, represented by point 1, is to the left of line 70. However, for operating conditions located to the right of line 70, such as point 2, an increase in the RMS of the current will actually reduce the cooling capacity of the compressor owing to losses in the armature. In addition, the cooling capacity of the compressor can be controlled by adjusting the duty cycle, D , of the current through the armature. The duty cycle is the ratio of

the pulse duration of the current to the pulse period, i.e., the ratio of the duration of when the windings are energized divided by the time between the beginning of one energization and the next. Referring to FIG. 6, the duty cycle of the armature current can be increased to the point where the operating condition of the compressor is to the right of line 70, such as point 3, where any additional increase in the duty cycle actually decreases the cooling capacity of the compressor. To prevent such occurrences in these previous compressors, larger-capacity compressors may be necessary. Larger-capacity compressors are typically more expensive and less efficient at lower capacities.

[0026] While a linear compressor may generally approximate the harmonic spring-mass system described above, this approximation is somewhat simplified. For example, the above equations do not account for damping, or losses, in the system. Most spring-mass systems are at least somewhat damped, i.e., they have losses in the system which dissipate energy and cause the motion of the piston to gradually decay. Further, although the above equations account for an initial input, they do not account for a continuous driving force. An equation mathematically representing a spring-mass system which accounts for system damping and a continuous driving force is $d^2x/dt^2 + (b/m)dx/dt + (k/m)x = A_0 \cos(\omega t)$, where b represents the damping coefficient of the system and ω represents the circular frequency of the driving force applied to the mass.

[0027] As indicated above, the spring constant k for the spring-mass system is mostly defined by the spring constants of springs 62 and 68, i.e., k_1 and k_2 , respectively. As known in the art, the spring constants of linear springs, for example, are substantially constant, although they may change slightly throughout their use. Referring to FIG. 5, in addition to the spring forces applied to piston 54 by springs 62 and 68, the refrigerant within cylinder bore 52 can apply an additional spring force acting on piston 54. More particularly, as the refrigerant in cylinder bore 52 is compressed by piston 54, the pressure of the refrigerant in cylinder bore 52 increases and the pressurized refrigerant exerts a force against piston 54. The refrigerant substantially acts like an elastic spring where the spring stiffness of the compressed refrigerant can be represented by a spring constant, i.e., k_3 . In view of this, in the present embodiment, the spring stiffness, k , of the spring-mass system equals $k_1 + k_2 + k_3$. However, owing to changes in pressure and temperature of the refrigerant, for example, the spring stiffness of the refrigerant, k_3 , may change throughout the operation of the compressor. For example, a change in either the temperature or pressure of the refrigerant may increase the stiffness, k_3 , of the refrigerant whereas a decrease in the temperature or pressure may decrease the stiffness. As the spring stiffness, k , of the spring-mass system affects the natural frequency of the system, when k_3 changes, the natural frequency of the spring-mass system changes.

[0028] Unlike previous compressors, compressors embodying the present invention are designed such that the frequency, ω , of the driving force acting on the spring-mass system can be increased above the natural frequency of the spring-mass system. In the illustrated embodiment, the frequency of the current passing through armature 56 determines the frequency of the driving force acting on the spring-mass system. In this embodiment, the frequency of the current substantially equals the frequency of the driving

force. Accordingly, to increase the frequency of the driving force acting on the spring-mass system, for example, the frequency of the current passing through armature 56 is increased. Increasing the frequency of the driving force acting on the spring-mass system can increase the strokes per minute of piston 54 within cylinder bore 52, thereby potentially increasing the output of the compressor. In one embodiment, the frequency of the current is increased without increasing the magnitude of the current, i.e., without increasing the stroke length of piston 54. In this embodiment, the potential disadvantages described above with respect to previous compressors can be avoided. However, in other embodiments, in addition to increasing or decreasing the output of the compressor through frequency modulation, the output of the compressor can also be increased or decreased by modulating the magnitude of the current and, thus, the stroke length of the piston.

[0029] As discussed above, the natural frequency of a spring-mass system can change throughout the operation of a compressor owing to changes in the temperature and/or pressure of the refrigerant being compressed in the cylinder bore of the compressor, for example. In one embodiment, the range of potential natural frequencies can be determined before the compressor is placed into service and the minimum frequency of the driving force can be set above the maximum potential natural frequency of the spring-mass system. In this embodiment, the natural frequency of the driving force can be established without continuously monitoring the parameters of the refrigerant being compressed. While this embodiment is a contemplated embodiment of the present invention, other embodiments are envisioned where the parameters of the refrigerant being compressed, or other parameters of the refrigeration system, are monitored and the frequency of the driving force is adjusted accordingly.

[0030] In one embodiment, the compressor includes a controller, such as controller 26 (FIG. 1), which monitors at least one system parameter and, in view of the information obtained from monitoring this parameter, makes a running correction to the frequency of the current passing through electrical wires 28 and armature 56. In one embodiment, referring to FIG. 1, temperature sensors 20 and 22 are placed in communication with the flow of refrigerant entering into and flowing out of compressor 12. In one embodiment, the controller is programmed with a first table of data that correlates the temperature of the refrigerant at one or both of these locations with a natural frequency of the spring-mass system. This table of data can be established empirically or via equations which calculate the spring constant k_3 of the refrigerant in the cylinder bore and, ultimately, the natural frequency of the system.

[0031] In a further embodiment, the controller can monitor a parameter of the electrical power provided to armature 56 including, for example, the current flowing through the armature or the voltage measured across the armature during operation. In one embodiment, the controller can be programmed with a second table of data which correlates the parameters of the armature current and/or voltage with the instantaneous natural frequency of the system. The controller can be programmed to compare the data in the first table and the data in the second table and make a running correction to the current flowing through the armature. The data contained in the second table can be derived from

equations which associate the operating parameters of the compressor with the natural frequency of the system. In one embodiment, the equation, $I_{\max} = f \cdot W_{\text{cycle}} / (D \cdot U_{\max})$ can be utilized, where W_{cycle} represents the work performed by the compressor per cycle of the compressor, where D represents the duty cycle of the current passing through the armature, where $U_{\max}$ represents the voltage drop across the armature, and where $I_{\max}$ represents the current passing through the armature. In one embodiment, the controller can include a frequency converter for converting the frequency of the current flowing into the armature to another frequency. The frequency converter can include electromechanical and/or solid state components, as known in the art.

[0032] In another embodiment, the instantaneous natural frequency of the spring-mass system can be determined by measuring the vibrations of the compressor. In one embodiment, an accelerometer can be affixed to the compressor housing and/or cylinder block, for example, to measure the vibrations produced by the spring-mass system. As known in the art, a spring-mass system produces different vibrations when the system is driven at its natural frequency as compared to when the system is driven above or below its natural frequency. The accelerometer can be placed in communication with the controller where the controller evaluates whether the spring-mass system is being driven at its natural frequency and makes any necessary adjustments to the frequency and/or magnitude of the current passing through the armature.

[0033] As discussed above, the capacity of compressors utilizing an embodiment of the present invention can be adjusted via adjustments to the current flowing through the armature. If the frequency of the current flowing through the armature is increased, the piston will typically be cycled through more strokes per minute. Likewise, if the frequency of the current is decreased, then the piston will be cycled through less strokes per minute. In view of this, a compressor which has a stroke that is close to its physical boundaries in the cylinder bore can be used and still provide capacity modulation for the refrigeration system. As a result, a smaller, less-expensive compressor can be used.

[0034] Although the advantages of operating the above-described compressors above the natural frequency of their spring-mass systems have been outlined herein, the compressors of the present invention are nonetheless capable of being operated at or below the natural frequency of their spring-mass systems. These circumstances typically arise when the compressor is being cycled on or off and/or when the demands of the refrigeration circuit drop and a lower output of the refrigeration system is required.

[0035] While this invention has been described as having an exemplary design, the present invention may be further modified within the spirit and scope of this disclosure. This application is therefore intended to cover any variations, uses, or adaptations of the invention using its general principles. Further, this application is intended to cover such departures from the present disclosure as come within known or customary practice in the art to which this invention pertains.

What is claimed is:

1. A linear compressor, comprising:
 - a cylinder block having a cylinder bore;
 - a piston, wherein at least a portion of said piston is positioned within said cylinder bore;
 - a first spring for positioning said piston, said piston and said first spring comprising a spring-mass system having a natural frequency;
 - an armature operably engaged with said piston to drive said piston at a driving frequency; and
 - a controller for increasing said driving frequency above said natural frequency.
2. The linear compressor of claim 1, further comprising a sensor for measuring the frequency of electrical current flowing through said armature, said sensor in communication with said controller.
3. The linear compressor of claim 1, further comprising a sensor for measuring one of the temperature and the pressure of a refrigerant compressed by said compressor.

4. A method for operating a linear compressor, comprising the steps of:

providing a cylinder block having a cylinder bore, a piston positioned within said cylinder bore, and at least one spring for positioning said piston, said piston and said at least one spring comprising a spring-mass system having a natural frequency;

applying a driving force to said piston to drive said piston within said cylinder bore at a driving frequency; and

increasing said driving frequency above said natural frequency, whereby said increasing step increases the capacity of said linear compressor.

5. The method of claim 2, further comprising the step of monitoring said natural frequency.

* * * * *