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Hover

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[54] **REFRIGERATION PROCESS, APPARATUS AND METHOD**

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[51] **Int. Cl.** **F25b 41/06**

[58] **Field of Search** 62/83, 115, 215, 62/217, 296, 508, 511, 505, 527; 417/299, 312

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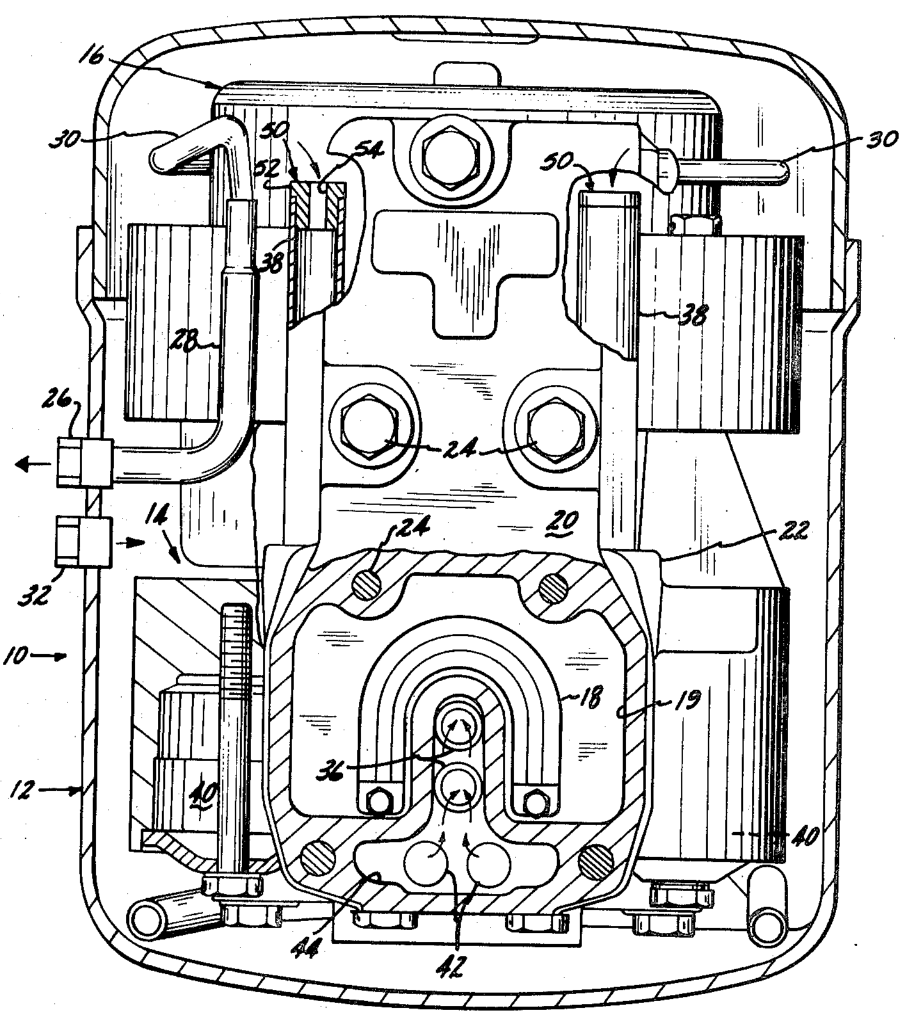
[57] **ABSTRACT**

A method in which a fixed size orifice is selected and inserted between the evaporator outlet and the suction inlet of a compressor in a refrigeration system to restrict gas flow to the compressor and thereby decrease the peak torque required to operate the compressor. The disclosed apparatus is a hermetic compressor unit with such a calibrated orifice installed in each of the compressor intake tubes, thereby allowing the use of an electric drive motor which otherwise would be incapable of developing sufficient peak load torque after starting to operate the compressor in a refrigeration system with an evaporator at ambient temperature in the absence of such orifices.

27 Claims, 5 Drawing Figures



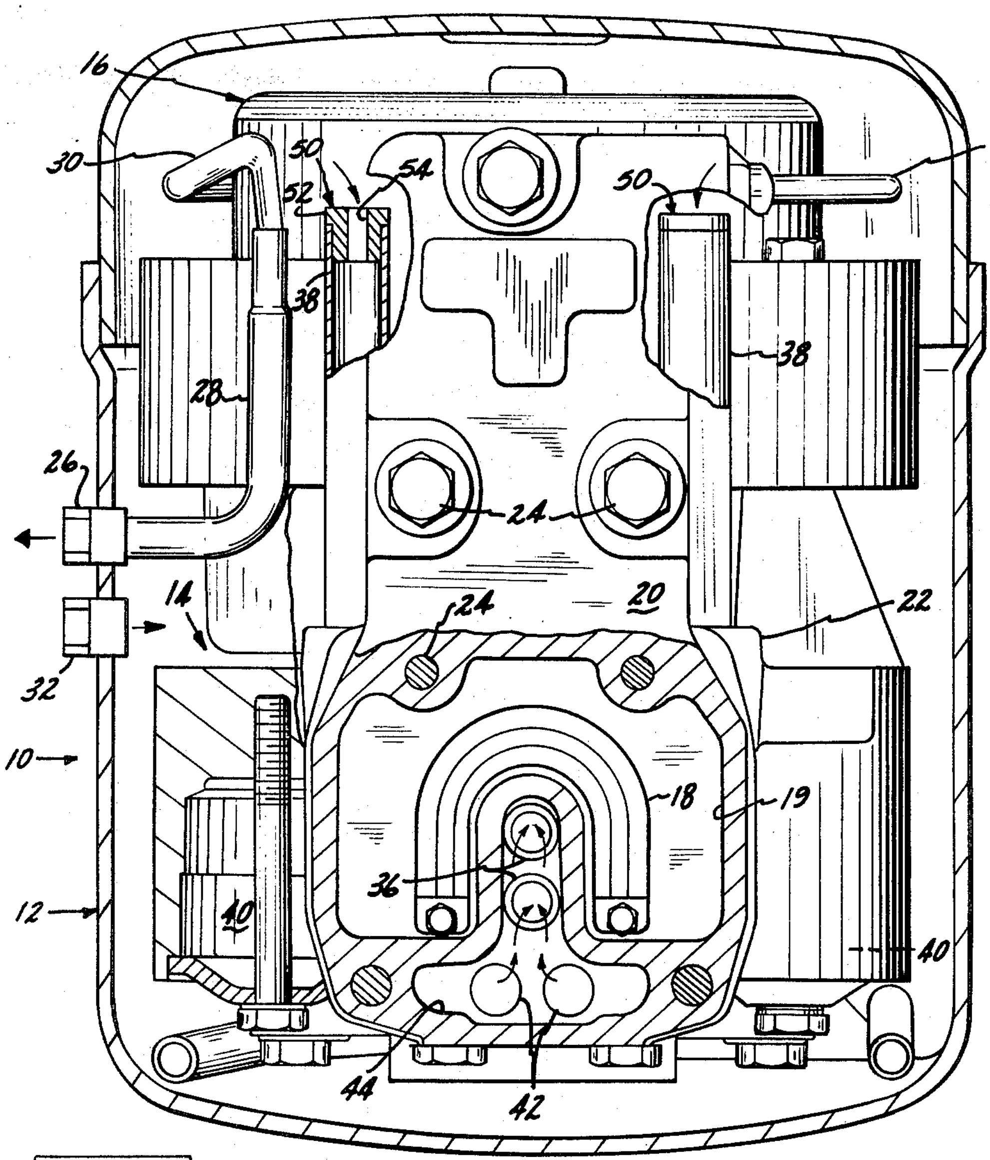


Fig-1

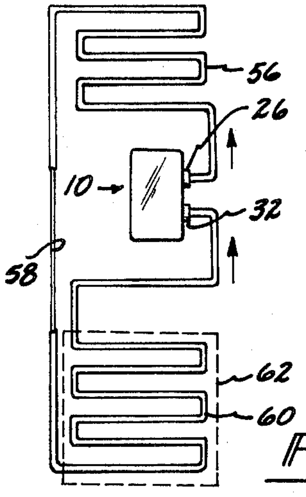


Fig-5

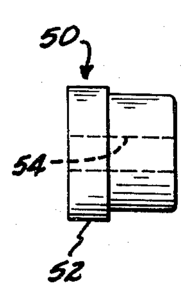


Fig-2

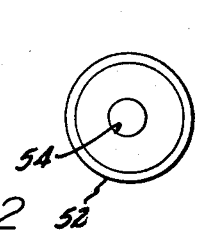


Fig-3

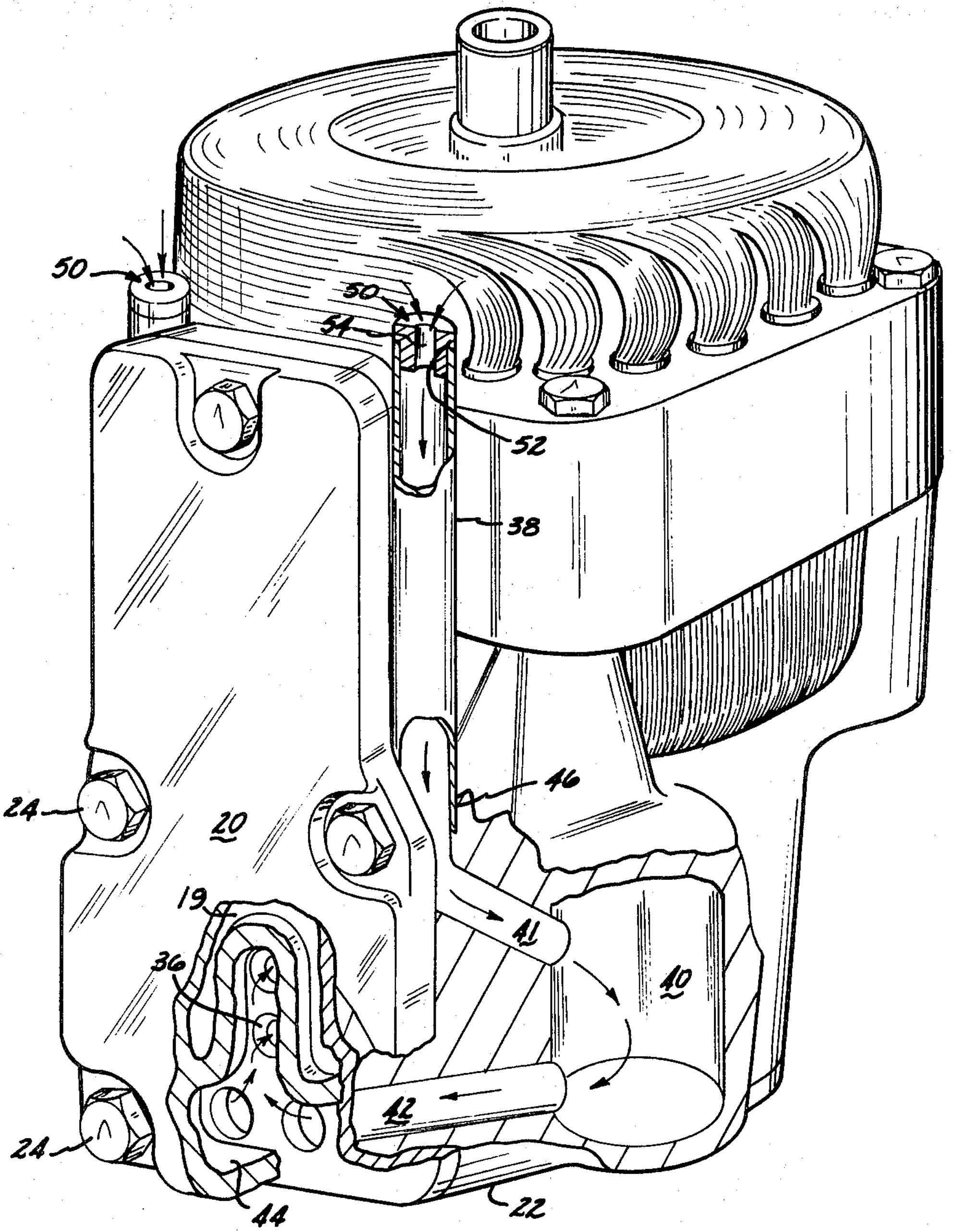


Fig-4

REFRIGERATION PROCESS, APPARATUS AND METHOD

This invention relates to refrigeration systems and more particularly to a vapor compression refrigeration process, apparatus and method of constructing apparatus utilizing the process.

A vapor-compression refrigeration system with a hermetically sealed compressor and a refrigerant such as R–12 is used in most conventional home refrigerators and freezers. Under normal operating conditions in such a conventional refrigerating system, the gaseous refrigerant is discharged from the evaporator and received at the suction inlet of the compressor at a pressure in the neighborhood of 20 pounds per square inch absolute (p.s.i.a.). However, if the evaporator becomes abnormally heated, such as by defrosting of the refrigerator or freezer or because the system is at ambient temperature and is being put into operation for the first time or after a prolonged shutdown, the pressure of the refrigerant at the suction inlet to the compressor may increase approximately three-to-five fold. During a prolonged shutdown of say over 12 hours, the pressure and temperature tends to become equalized throughout the refrigeration system, such equalization being referred to as temperature equilibrium or equalization. During a short shutdown, the pressure tends to become equalized while the temperatures are still not equalized, such equalization being referred to as pressure equalization.

When the compressor is first started in either a pressure equalized or temperature equalized refrigeration system under such abnormally high temperature conditions, there is an abnormally large mass flow rate of refrigerant through the compressor and into the condenser which results in a temporarily abnormally high pressure at the discharge of the compressor. After the refrigeration system has operated for a few minutes, the mass flow rate of refrigerant through the compressor and the discharge pressure return to normal. However, the temporary abnormally large mass flow rate of refrigerant through the compressor and the resulting high discharge pressure require considerably greater torque or work to run the compressor. Hence, such a refrigeration system has hitherto required an electric motor or other prime mover capable of producing the abnormally high torque required to run the compressor under such temporary abnormal load conditions.

In the method of this invention the abnormally high mass flow rate of refrigerant through the compressor during start-up of a pressure and/or temperature equalized refrigeration system is substantially decreased by coupling the discharge of the evaporator to the inlet of the compressor through an inexpensive orifice of a preselected fixed size. The term "orifice" as referred to here and throughout the remainder of this specification is intended to mean one or more nozzle or sharp-edged orifices (including thin plate orifices), with plural orifices being arranged in parallel. The orifice automatically throttles abnormally high mass flow rates of the refrigerant so that the compressor is not unduly loaded during this temporary load condition and yet allows a sufficient mass of refrigerant to flow into the compressor during normal operation of the refrigeration system. The use of the orifice thus allows the compressor to be run by a less powerful electric motor without materially adversely affecting the overall efficiency of the refrigeration system. Such an electric motor is less expensive to manufacture and requires less space in a hermetic unit than would otherwise be required. Apparatus constructed pursuant to the method of this invention includes a calibrated orifice of a given selected size coupled to the inlet of a conventional compressor gas pump so as to enable it to be driven by an electric motor of a selected reduced capacity; i.e., a motor having a maximum torque rating below that otherwise required to initially start-up and drive the compressor in a refrigeration system with an abnormally large mass flow rate condition were the orifice not present, as described in greater detail hereafter.

Object of this invention are to provide a method of constructing a compressor for a mechanical vapor compression refrigeration system which substantially decreases the torque load imposed on a compressor in a pressure and/or temperature equalized refrigeration system, and a motor-driven compressor unit constructed pursuant to the method for use in such a refrigeration system which is compact and of economical and reliable construction.

These and other objects, features and advantages of this invention will be apparent from the following description, claims and accompanying drawings in which:

FIG. 1 is an end view partially in vertical section of a hermetically sealed compressor unit with an electric drive motor for use in a vapor compression refrigeration system embodying this invention.

FIGS. 2 and 3 are side and end views, respectively, of a restricted orifice of the hermetically sealed compressor unit of FIG. 1.

FIG. 4 is an isometric view partially in section of the compressor unit of FIG. 1 with the hermetic shell and various other component parts removed therefrom.

FIG. 5 is a semi-schematic drawing of a vapor compression refrigeration system embodying this invention with an evaporator and condensor respectively connected to the inlet and outlet of the hermetically sealed compressor unit of FIG. 1.

Referring in more detail to the accompanying drawings, FIG. 1 illustrates a hermetically sealed compressor unit **10** embodying apparatus for performing the method of this invention in a vapor compression refrigeration system. Hermetic compressor unit **10** has an outer shell **12** encasing a compressor **14** resiliently suspended therein which is driven by an electric motor **16**. Compressor **14** is of the positive displacement reciprocating piston type with a discharge leaf valve **18** mounted in a discharge chamber **19** of a cylinder head **20** secured to a cylinder block **22** by bolts **24**. Preferably, compressor **14** is a low-side casing type and hence has a discharge outlet coupling **26** fixed to a side wall of shell **12** which is coupled with the interior of discharge chamber **19** through tubular conduits **28** and **30**. A refrigerant such as R–12 (dichlorodifluoromethane) is received in the casing space defined by hermetic shell **12** through an inlet coupling **32**, and the bottom portion of the shell provides a reservoir for lubricating oil.

As shown in FIGS. **1** and **4**, the refrigerant gas to be compressed is supplied from the interior of shell **12** to inlet valve ports **36** of compressor **14** through a pair of upright intake tubes **38** which are mounted at their lower ends in cylinder block **22** and each connected via a muffler chamber **40**, passageway **41**, passageway **42** and suction chamber **44** with ports **36** in cylinder head

20. Intake mufflers **40** and passages **41** and **42** may be cored or machined in cast iron cylinder block **22.** Each intake tube **38** is press fit and/or silver soldered or adhesively secured in a counter bore **46** in the upper end of passageway **41.** The compressor construction described above may be conventional and hence these and other details thereof will be well understood by those skilled in the art.

In accordance with a principal feature of the present invention, a calibrated orifice is selected and installed in the inlet passageway leading to ports **36.** In the example illustrated herein, this orifice is preferably provided in the form of hollow restrictor plugs **50** which are received and fixed one in each of the upper ends of intake tubes **38,** as by press fitting the plugs therein. As shown in FIGS. **2** and **3,** each plug **50** is generally cylindrical and has a shoulder **52** adjacent one end which limits the extent to which the plug can be inserted into the associated tube **38.** Each plug **50** has a fixed flow restriction in the form of a coaxial restricted orifice **54** therethrough which has a minimum cross-sectional area of less than one third and preferably in the range of one fourth to one tenth of the cross-sectional area of the inside diameter of its associated intake tube **38.** Although the orifice **54** is fixed in size, it nevertheless provides a non-linear resistance to flow of gas therethrough; i.e., the pressure drop across orifice **54** varies directly with the mass flow rate of refrigerant gas therethrough in accordance with the known fluid dynamics of restrictive orifices. The sun of the volumes of both intake tubes **38,** both passages **41,** both mufflers **40,** both passages **42** and the common suction chamber **44** is preferably in the range of two-to-five times the volumetric displacement of reciprocating piston compressor **14.**

With respect to the location of the calibrated orifice, it is desirable that there be a chamber of some given volume between the orifice and the suction valve leaf. In the illustrated design, this chamber consists of the usual suction muffler chamber **40** and associated passages **41, 42** and the suction chamber **44** formed in the head of the compressor plus the interior of the suction tube **38** itself downstream from the restriction plug **50.** The optimum point at which plug **50** should be located from the standpoint of the best starting characteristics is at the entrance to the suction tube. However, in some installations from the standpoint of noise reduction, it may be better to install plug **50** at the downstream end of suction tube **38** where it enters the muffler chamber proper. In either location the restricted orifice plug **50** in conjunction with the suction tube **38** actually adds what amounts to additional muffling to the existing muffler **40** and in this manner contributes to noise reduction. Orifice **54** may also be located at the point where the return line from the refrigeration system enters the compressor casing; i.e., in coupling **32,** rather than in the suction tube. However, this latter variation would not give the noise reduction properties of the illustrated design.

As shown in FIG. **5,** compressor unit **10** with restricted orifices **54** installed in the intake tubes thereof is used in a vapor compressor refrigeration system with a condenser **56,** capillary tube **58** and an evaporator **60** adapted for removing heat from a refrigerator cabinet **62.** Discharge outlet coupling **26** is connected to the inlet of condenser **56** and the outlet of the condenser is connected to the inlet of capillary tube **58.** The outlet

of capillary tube **58** is connected to the inlet of evaporator **60** and the outlet of evaporator **60** is connected to inlet coupling **32** so that the compressor unit can circulate a refrigerant through the system.

In most automatic defrosting refrigerators, evaporator **60** is located directly in refrigeration cabinet **62,** and a thermostat senses the temperature of the evaporator and cycles the compressor. In normal operation of such a refrigerator, after the compressor has been shut off by the thermostat, the thermostat prevents the compressor from being restarted until the temperature of the evaporator in the food compartment rises to approximately 40°–45°F. This rise in temperature melts any ice formed on evaporator **60,** thereby defrosting the evaporator. In addition, this entails sufficient down time to allow the refrigeration system pressures to equalize after every compressor pumping cycle.

In the so-called "frost-free" refrigerator, evaporator **60** is located outside cabinet **62** and a fan circulates air over the evaporator and into the cabinet to remove heat therefrom and cool the contents thereof. In a frost-free refrigerator defrosting cycle, hermetic compressor unit **10** and the fan are shut off and an electric coil is used to heat the evaporator to approximately 65°F., thereby melting any ice formed thereon and also usually allowing sufficient time for the refrigeration system pressures to equalize. In some frost-free refrigerators, there are typically four defrost cycles per day controlled by a timing clock. In other frost-free refrigerators, there is a defrosting cycle every time the compressor stops, which usually occurs two to four times per hour.

Also, the entire refrigeration system for all refrigerators and freezers, regardless of their type and whether or not they have a defrost or frost-free cycle, usually reaches room or ambient temperature whenever the motor of the compressor unit is disconnected from its source of power for any substantial period of time. Thus, there are many occasions when the evaporator of the refrigeration system becomes heated substantially above its normal operating temperature and the refrigeration system reaches temperature as well as pressure equilibrium for one reason or another.

In normal substantially steady state operation of the refrigeration system with a refrigerant such as R–12, the compressor unit suction inlet pressure at coupling **32** is substantially equal to the discharge pressure of evaporator **60** and is in the neighborhood of 20 p.s.i.a. Under such conditions, the discharge pressure of the compressor unit at coupling **26** is substantially equal to the inlet pressure to condenser **56** and is in the neighborhood of 200 p.s.i.a. if the refrigeration system is cycled so that the evaporator temperature is maintained in the range of −10°F. to 0°F. However, as previously indicated, whenever the evaporator is heated to an abnormally high temperature, such as by defrosting or by disconnecting the source of power to the compressor motor **16,** the refrigeration system pressure rises abnormally and with the compressor shut down, pressure equalization occurs so that the refrigerant throughout the system tends to equalize at pressures as high as 55 p.s.i.a. when the temperature of the evaporator reaches approximately 45°F., approximately 80 p.s.i.a. at approximately 65°F., and approximately 100 p.s.i.a. at approximately 80°F. As a result of this pressure rise, the density or mass per unit of volume of the gaseous refrigerant within shell **12** of hermetic compressor **10** is in the range of two to five times the density of the gaseous refrigerant in the shell during the substantially steady state operation of the refrigeration system. Hence, in the absence of restrictive orifices **54**, there would be a much greater than normal mass flow rate of refrigerant through compressor **14** and into condenser **56**, causing a rapid rise of the pressure of the refrigerant in the condenser to a value in excess of 250 p.s.i.a. and creating an abnormally high pressure against which the compressor must work, thus substantially increasing Jhe torque demand on the driving motor 15 to 30 seconds after the compressor is started.

However, due to the presence of orifices **54**, when the compressor unit **10** is restarted after the defrosting or other heating of evaporator **60**, the rate of flow of this high density gaseous refrigerant into compressor **14** is automatically restricted, thereby limiting the amount of work done by the compressor and thus, as previously indicated, reducing the maximum torque load demand on the motor driving the compressor during the initial start-up of the refrigeration system after the evaporator has been heated. Usually in about 2 minutes after such a start-up the compressor suction pressure at coupling **32** will have dropped to about 30 p.s.i.a. and in about five minutes after start-up the discharge pressure at coupling **26** of the compressor will have dropped to about 200 p.s.i.a. Once the pressure at the compressor inlet has thus been reduced, orifices **54** due to their non-linear resistance to gas flow present a disproportionately lower resistance to gas flow therethrough and hence present only a slight increase in the pressure drop thereacross compared to that existing through tubes **38** not equipped with restrictor plugs **50**.

The invention is particularly well suited for use with low and medium back pressure refrigeration systems wherein the density of the refrigerant increases sufficiently due to the heating of the evaporator to provide an increase in the mass flow rate through the compressor of sufficient magnitude that the power input or torque required to start and drive the compressor is sharply increased. The back pressure, i.e., the pressure of the refrigerant at the outlet of the evaporator and inlet to the compressor, is dependent on the particular refrigerant used in the system and the temperature of the evaporator during normal substantially steady state operation of the refrigeration system. In low back pressure systems, the pressure of the refrigerant at the evaporator outlet is generally in the range of 10 to 30 p.s.i.a. and in medium back pressure systems, the pressure of the refrigerant at the evaporator outlet is in the range of 20 to 45 p.s.i.a. for R–12 and 30 to 70 p.s.i.a. for R–22. Low back pressure systems using a refrigerant such as R–12 with the evaporator normally operating in the range of −40°F. to 10°F. are commonly used in home refrigerators and freezers. Similarly, medium back pressure systems using a refrigerant such as R–12 or R–22 with the evaporator normally operating in the range of −10°F. to 30°F. are used in package or beverage dispensers and commercial refrigerators such as display cases.

The minimum surface area in radial cross section of orifice **54** through plug **50** is selected to provide an optimum balance between the conflicting parameters of maximum output capacity and operating efficiency of the hermetic motor compressor unit **10** during normal operation thereof versus limiting the maximum torque required to run the compressor under the abnormally large mass flow rate conditions. Under normal running conditions the effect of the orifices as a resistance to the flow of gas becomes much less and hence the capacity of the compressor is not reduced during running to the same extent as it is during abnormal load conditions. Therefore, the small sacrifice in pumping capacity of the compressor under normal running conditions is more than made up by the reduction in torque required to run the compressor under peak load conditions. This reduction in maximum torque imposed by abnormal load conditions in turn permits a smaller size motor to be used, or for the same size motor it provides a higher rating for the compressor because of its ability to run under more severe pressure conditions. In general, the smaller the minimum cross-sectional surface area of the orifices **54**, the greater the reduction in the amount of peak torque or power required to run the compressor and the lesser the overall efficiency and output capacity of the hermetic motor compressor unit. However, the cost benefits achieved from the reduction in the peak torque required to run a compressor in low and medium back pressure refrigeration systems achieved by using a calibrated orifice pursuant to the present invention more than offsets the slight loss in overall efficiency, which may be in the range of one-half to 2 percent. If desired, the loss in capacity resulting from the installation of a calibrated orifice or orifices **54** can be readily compensated for in existing compressor designs by increasing the bore diameter of the cylinder or cylinders in compressor **14** to thereby increase the displacement of the compressor an amount sufficient to offset the reduction in pumping capacity caused by the orifices. This change can be accomplished at very little cost.

Most refrigeration cabinets have an optimum performance obtained by using a compressor of a precise capacity rating, but no one model compressor may have this rating. Hence, compressors are quite often mismatched to some extent with the associated refrigeration system equipment. Moreover, using an oversize or unduly high capacity compressor with a given cabinet does not necessarily produce a corresponding increase in the output or performance of the cabinet in its refrigeration capacity. In accordance with the method of the invention, an appropriate orifice plug or plugs **50** are selected for a given compressor, thereby enabling a standardized compressor design to be readily "matched" to varying types of refrigeration equipment.

Thus far it has been found that a restriction in the form of the orifice **54** provides better characteristics for optimizing of peak torque versus running capacity as compared to a restriction in the form of a long narrow tube. The calibrated orifice could be a plug **50** as shown or it could be a thin plate or a mere necking down of the suction tube **38** to provide the appropriate ratio of area change. Thus, in practicing the method, the compressor manufacturer may provide a standardized compressor design capable of handling a range of system capacities. Then, in order to match this standard compressor to a given system having a given capacity requirement, the maximum load conditions presented by said system to said standard compressor are determined whereupon a calibrated orifice **54** may be installed in the compressor according to parameters disclosed herein to prevent overloading thereof to thereby match the compressor to said selected system.

Calibrated orifices **54** having a minimum cross-sectional area providing a pressure drop across the orifices in the range of 2 to 6 percent, and preferably approximately four percent, of the absolute pressure of the refrigerant in shell **12** during normal substantially steady state operation of the refrigeration system (i.e., after the effects of the heated evaporator and pressure equalization have been dissipated) are believed to function satisfactorily. For the low back pressure refrigeration system of FIG. **5**, a pressure drop in the range of 0.4 to 1.2 p.s.i., and preferably 0.8 p.s.i., is believed to be satisfactory. More particularly, for calculating an orifice such as passage **54** with a circular cross section, the diameter in inches of the minimum surface area in cross section should preferably be substantially equal to

$$\frac{VQ}{K(P)\left[\frac{1}{2+\frac{BP}{AP+1}}\right]}$$

where A is an empirical constant equal to 0.24, B is an empirical constant equal to 0.12, K is an empirical constant equal to 267, V is the specific volume in cubic feet per pound mass of the gaseous refrigerant in shell **12** of the hermetic compressor during normal steady state operation of the refrigeration system, P is the desired pressure drop across the orifice in pounds per square inch (it being understood that the value (P) is raised to the power expressed in the large brackets of the foregoing expression), and Q is the mass flow rate in pounds per hour through the orifice during normal substantially steady state operation of the refrigeration system.

With the use of calibrated orifices **54** in hermetic compressor unit **10**, compressor **14** can be and preferably is driven by an electric motor **16** incapable of producing sufficient maximum torque to initially start and run the compressor after the evaporator has been heated and the refrigeration system pressure equalized in the absence of orifices **54**. For example, it has been found that a hermetic compressor unit constructed in accordance with this invention having a single cylinder positive displacement reciprocating piston-type compressor with a volumetric displacement of 1.067 cubic inches can be successfully started and run in a pressure equalized refrigeration system with a heated evaporator having an R–12 refrigerant by an electric motor capable of developing a maximum torque of approximately 34 ounce feet at 115 volts when calibrated orifices **54** are utilized. Under the same operating conditions, the same electric motor will stall within 30 seconds after the initial start-up of the compressor in the same refrigeration system with the orifices **54** removed. This particular hermetic compressor unit had orifices providing a pressure drop of about 4 percent of the normal substantialy steady state pressure of the refrigerant within shell **12** which resulted in a reduction of more than 25 percent in the maximum torque required to start and run the compressor in a pressure equalized system with a heated evaporator. This substantial reduction in peak torque was achieved with a decrease of less than 4 percent in the overall efficiency of the hermetic compressor unit **10** and a decrease of less than 10 percent of the maximum output of the compressor unit under the standard operating conditions specified in Section 6.2 of Standard 520 of the Air Conditioning and Refrigeration Institute as published in 1968, with the evaporator at −10°F., the gas entering the compressor at 90°F., the compressor ambient temperature at 90°F., the liquid at the expansion valve at 90°F. and the condensing temperature at 130°F. These tests were conducted in accordance with the Methods of Testing for Rating Positive Displacement Refrigerant Compressors effective June 25, 1967, of the American Society of Heating, Refrigeration and Air Conditioning Engineers, Inc. In this compressor unit there were two orifices **54** each having a minimum cross-sectional diameter of 0.125 inch in accordance with the above formula and a length of three-eighths inch. The pressure drop across each orifice was 0.8 pound per square inch when the upstream pressure was 19 p.s.i.a. and the mass flow rate through the compressor during substantially steady state operation was 24 pounds per hour. The cross-sectional area of the opening of each orifice was approximately one-ninth of the cross-sectional area of its associated intake tubes and the combined total volume of the intake tubes **38**, intake passageways **41** and **42**, intake mufflers **40** and suction chamber **44** was approximately seven cubic inches. The refrigeration system was cycled to maintain the temperature of the evaporator in the range of −10°F. to 0°F., except during defrosting when the evaporator was heated to raise the suction pressure to 95 p.s.i.a., and the compressor started and ran with 110 volts to the motor.

Another example of a hermetic compressor unit constructed and used in a vapor compression refrigeration system in accordance with this invention is a hermetic compressor unit having a positive displacement reciprocating piston-type compressor with a volumetric displacement of 1.067 cubic inches, a motor developing a miximum torque of approximately 34 ounce feet at 115 volts and two orifices **54** each having a diameter of **.109** inch and a length of three-eighths inch. The pressure drop across each of the orifices **54** was approximately 1.0 p.s.i. when the upstream pressure was 19 p.s.i.a. The minimum cross-sectional area of each restricted orifice was approximately one-twelfth of its associated intake tube and the combined total volume of the intake tubes **38**, intake passages **41** and **42**, suction mufflers **40** and suction chamber **44** was 7 cubic inches. This hermetic compressor unit was operated in a refrigeration system with an R–12 refrigerant having a normal substantially steady state mass flow rate of 22.8 pounds per hour through the compressor with a normal substantially steady state pressure of 19 p.s.i.a. in shell **12** with the hermetic compressor unit being intermittently operated to maintain the temperature of the evaporator between −10°F. to 0°F. except during defrosting when the evaporator was heated to raise the suction pressure to 104 p.s.i.a., and the compressor started and ran with 110 volts to the motor.

A further example of a hermetic compressor unit constructed and used in a vapor compression refrigeration system in accordance with this invention is a hermetic compressor unit having a positive displacement reciprocating piston-type compressor with a volumetric displacement of 1.067 cubic inches, a motor developing a maximum torque of approximately **34** ounce feet at 115 volts and two restricted orifices **54** each having a diameter of 0.140 inch and a length of three-eighths inch. The pressure drop across each orifice was approximately 0.06 p.s.i. when the upstream pressure was 19 p.s.i.a. The minimum cross-sectional area of each orifice **54** was approximately one-seventh of its associated intake tube **38** and the combined total volume of the intake tubes **38**, intake passages **41** and **42**, suction mufflers **40** and suction chamber **44** was 7 cubic inches. This hermetic compressor unit was operated in a refrigeration system with an R–12 refrigerant having a normal substantially steady state mass flow rate of 24.3 pounds per hour through the compressor with a normal substantially steady state refrigerant pressure of 19 p.s.i.a. in shell **12** with the hermetic compressor unit being intermittently operated to maintain the temperature of the evaporator between −10°F. and 0°F. except during defrosting when the evaporator was heated to raise the suction pressure to 87 p.s.i.a., and the compressor started and ran with 110 volts to the motor.

Both of these latter hermetic unit examples functioned in a refrigeration system in generally the same manner and mode and produced the same results as the first example described above.

By selecting and installing a restricted orifice plug **50** between the discharge of an evaoprator and the inlet of the compressor in accordance with the present invention, a vapor compression refrigeration apparatus and process is provided which substantially decreases the peak power or torque requirements of the compressor in a pressure and/or temperature equalized refrigeration system, particularly with a heated evaporator. The resulting hermetic compressor unit utilizes an electric drive motor having a substantially smaller maximum running torque capacity. The invention thus provides a hermetic compressor unit of more economical construction and assembly compared to prior art hermetic compressor units. Moreover, the use of a calibrated orifice with a fixed minimum cross-sectional area in the form of a simple, reliable and easily installed orifice plug **50** enables this result to be accomplished with an inexpensive structure which is substantially service and maintenance free throughout the useful life of the refrigeration system.

From the foregoing description, it will also now be apparent that the present invention is particularly useful in applications involving low to medium back pressure refrigeration systems, although it is not necessarily limited to such systems. Normally in such systems the initial starting torque demand is less than the peak torque encountered by motor **16** after the compressor **14** has been brought up to running speed, which may be for example 3,400 r.p.m. It only requires a fraction of a second, say one-tenth to one-half second, for the motor to accelerate the compressor from zero velocity to its normal running speed. However, when the system is started after being shut down for a sufficient period of time to cause partial or complete pressure equalization, which normally will occur anywhere from 5 to 15 minutes after shutdown, peak torque demand will be encountered normally from about 20 to 30 seconds to about 1 minute after start-up of the compressor. This peak torque will normally last for a few minutes until the suction or back pressure (normally measured in the compressor housing **12** prior to entry of the refrigerant gas into the suction tubes **38**) has been reduced approximately to normal operating values in the system. Thus, in low to medium back pressure systems, the present invention enables a compressor motor of reduced capacity to drive through this period of peak loading by limiting the maximum gas pumping load which can be presented by the system to the compressor.

As indicated previously, the peak load conditions presented by various types of vapor compression refrigeration systems will, of course, vary due to such factors as the system having a defrost cycle or frost-free mode of operation, as well as the type of condition encountered at start-up of the compressor. During the occurrence of a "soak-out" condition (i.e., wherein the system is shut off for an extended period of time, normally 18 to 24 hours, so that both temperature and pressure equalization will occur throughout the system), an abnormal gas pumping load may be imposed on the compressor. However, pressure equalization can also occur without temperature equilibrium having been established, as in the case of a defrost cycle. At the start of such a cycle, the condenser is already hot, the condenser gas temperature being say in the range of 100°F. to 120°F., and then due to warming of the evaporator the gas temperature therein will be raised from say 0°F. to 40°F. This presents an aggravated load situation upon restarting of the compressor as compared to a soak-out condition wherein system temperature may have equalized at a lower temperature, say anywhere from 60°F. to 110°F., depending upon ambient temperature conditions. The peak load problems, of course, are encountered at the high end of such temperature ranges. Normally there is not enough gas in the system to create enough pressure to cause condensation of the gas in the condenser at such elevated temperatures. Hence, saturated vapor conditions may not prevail and, if so, the pressures do not follow the saturated vapor curves.

Moreover, during a soak-out condition, the lubricating oil present in the oil sump at the bottom of shell or housing **12** will drop in temperature. This factor, coupled with the quiescent state of the oil, will allow the oil to absorb some of the refrigerant vapor. When the vapor goes into solution in the oil, the system will be depleted of some of its refrigerant in vapor form, thereby reducing the gas pumping load encountered in "pulling down" the system after start-up. However, during a shorter shutdown condition, such as occurs in a defrost cycle, the oil remains too hot to absorb much of the refrigerant vapor and hence more refrigerant in vapor form will be present in the system. Moreover, the gas pressure can reach higher levels because the motor will still be hot from its previous operating cycle, and there is more gas in the system because less of the gas can be absorbed by the higher temperature oil. Also, the condenser will be hotter than the ambient temperature and hence the discharge pressure of the compressor will be higher after start-up and the rise in discharge pressure will be accelerated. Nevertheless, if the gas temperature is high enough in the system, there still may not be enough refrigerant present to produce saturated gas conditions. Hence, short shutdown cycles may pose more severe gas loading problems than encountered after a soak-out condition has occurred.

Due to the various factors mentioned previously, the density of the refrigerant gas at the inlet to the compressor may be at least doubled compared to the density of the refrigerant gas during normal operation of the refrigeration system. In many instances there will be a two-to-five fold increase in the density of the gaseous refrigerant when the evaporator is heated from say 0°F. to about 80°F., assuming a saturated vapor condition. However, after the evaporator temperature reaches about 80°F., usually no liquid refrigerant will be left in the system. Hence, saturated vapor conditions will no longer prevail and, therefore, above this temperature the pressure does not increase as fast due to the presence of superheated vapor conditions in the system. Nevertheless, until such superheated vapor conditions are reached, the aforementioned two-to-five fold density change or increase in back pressure can and does occur.

The aforementioned peak gas pumping load problems normally may not be as serious in high back pressure systems but the concepts of the present invention can also be utilized to advantage in such systems. By providing a suitably calibrated orifice means in such a system, gas loading imposed on the gas pump of the compressor can be limited, thereby enabling the amount of refrigerant charge in the system to be increased. In other words, by suitably throttling gas flow into the compressor housing in such a system, the system can tolerate a greater amount of refrigerant charge before the problems of oil pump out or liquid entering the cylinder of the compressor will be encountered. Also, in high back pressure systems, the provision of a calibrated orifice means of the present invention will reduce the maximum pressure and temperature conditions produced in the system when the condenser fan fails or the air flow through the condenser is reduced or restricted due to other adverse conditions, such as dirt or dust accumulation in the condenser, or when the ambient temperature condition of the condenser is abnormally high.

I claim:

1. A method of decreasing the peak torque demand of a hermetically sealed positive displacement compressor in a vapor compression refrigeration system containing a refrigerant and wherein a low or medium back ressure exists at the outlet of an evaporator of the system during normal operation thereof, said method comprising the steps of providing a compressor having a volumetric displacement rating in excess of the requirements of said system, selecting flow restriction orifice means having a fixed minimum cross-sectional area calibrated to limit the mass flow rate of said refrigerant into said compressor after it is just started in the operation of said system under an abnormal condition with said evaporator at an elevated temperature relative to its normal operating temperature, and locating said orifice means in said refrigeration system such that all of the refrigerant entering the suction inlet of the compressor does so exclusively via said orifice means, whereby the peak torque required to run the compressor under said abnormal condition is decreased compared to the torque required to run said compressor under the same conditions without said orifice means.

2. The method of claim **1** in which said orifice means is located between the outlet of the evaporator and the suction inlet of the compressor.

3. The method of claim **2** in which said orifice means is located in a passageway and said minimum area of said flow restriction means is dimensioned to be less than one-third of the minimum cross-sectional area of the passageway immediately downstream of said flow restriction means of said orifice means.

4. The method of claim **2** in which said restricted orifice means is located in a passageway and said minimum area of said flow restriction means is dimensioned to be in the range of one-third to one-tenth of the minimum area of the passageway immediately downstream of said flow restriction means.

5. The method of claim **2** in which the fixed minimum cross-sectional area of said flow restriction means is selected such that the average pressure drop thereacross is less than 6 percent of the normal operating pressure of the refrigeration system at the outlet of the evaporator.

6. The method of claim **2** in which the fixed minimum cross-sectional area of said flow restriction means is selected such that the average pressure drop thereacross is less than 1 pound per square inch absolute during normal substantially steady state operation of the refrigeration system.

7. The method of claim **2** in which said flow restriction means is selected such that the average pressure drop thereacross is in the range of four-tenths to one and two-tenths pounds per square inch during normal substantially steady state operation of the refrigeration system.

8. The method of claim **2** in which said flow restriction means is selected such that the decrease in overall efficiency of the hermetic compressor due to coupling the outlet of the evaporator with the suction inlet of the compressor through the flow restriction means is less than five percent.

9. The method of claim **2** in which the fixed minimum cross-sectional area of said flow restriction means is generally circular and has a diameter *D* in inches substantially equal to

$$\frac{VO}{K(P)\left[\dfrac{1}{2+\dfrac{BP}{AP+1}}\right]}$$

where *K* equals 167, *A* equals 0.24, *B* equals 0.12, *V* equals the specific volume in cubic feet per pound mass of the refrigerant immediately upstream of the orifice, *P* equals the pressure drop across the flow restriction means in pounds per square inch, and *Q* equals the mass flow rate in pounds per hour through the compressor when the refrigeration system is operating under the conditions of Section 6–2 of Standard 520 of the American Refrigeration Institute, with the evaporator temperature at about −10°F., the gas temperature entering the compressor at about 90°F., the compressor ambient temperature at about 90°F., the liquid temperature at the expansion valve at about 90°F., and the condensing temperature at about 130°F.

10. The method of claim **9** in which said flow restriction means is located in an inlet tube coupled to a muffler of the hermetically sealed compressor.

11. The method of claim **2** in which said flow restriction means is located in an inlet tube coupled to a muffler of the hermetically sealed compressor.

12. The method of claim 1 wherein an electric motor is provided to drive the compressor and said motor is selected so as to be incapable of producing sufficient torque to start and run the compressor when said refrigeration system is in a temperature and/or pressure equalized condition with said evaporator at an elevated temperature without said restricted orifice.

13. In a hermetic compressor unit for a refrigeration system having a hermetic casing with inlet means for supplying a refrigerant gas to the space in said casing, a positive displacement compressor with an inlet valve, said compressor being mounted in said casing and adapted to be driven by an electric motor, a muffler chamber having an outlet communicating with said compressor inlet valve and a suction pipe extending generally upright in said casing and communicating at its lower end with an inlet of said muffler chamber and at its upper end with the space in said casing, the improvement comprising restricted orifice means having a fixed minimum cross-sectional area in said suction pipe for restricting the mass flow rate of refrigerant therethrough, said orifice means being located such that all of the refrigerant enters said compressor exclusively via said restricted orifice means and being dimensioned to limit the mass flow rate of refrigerant to a given value through said compressor when said compressor is started under a given condition of abnormally high density of gaseous refrigerant in said casing of at least twice the density of the gaseous refrigerant in said casing during normal operation of the referigeration system, and an electric motor received in said casing and adapted to drive said compressor, said electric motor being incapable of producing the peak torque imposed by the gas pumping load exerted on said compressor under said given condition in the absence of said orifice means and being capable of developing the maximum torque imposed by the gas pumping load exerted on said compressor under said given condition with said orifice means in said suction pipe.

14. The hermetic compressor unit of claim 13 wherein said restricted orifice means is located at the upper end of said suction pipe.

15. The hermetic compressor unit of claim 13 wherein said restricted orifice means is located in the vicinity of the connection of said suction pipe with said muffler chamber.

16. The hermetic compressor unit of claim 13 wherein the combined total volume of all passageways and chambers between said restricted orifice means and said inlet valve of said gas compressor is in the range of two to five times the volumetric displacement of said compressor.

17. The hermetic compressor unit of claim 13 in which the minimum cross-sectional area of said restricted orifice means is less than one-third of the minimum cross-sectional area of the passageway defined by said suction tube downstream of said restricted orifice means.

18. The hermetic compressor unit of claim 17 in which said minimum cross-sectional area of said restricted orifice means is such that the average pressure drop across said restricted orifice means is less than 6 percent of the pressure of the refrigerant gas in said hermetic casing during normal substantially steady state operation of the refrigeration system in which the compressor unit is utilized.

19. The hermetic compressor unit of claim 17 in which said minimum cross-sectional area of said restricted orifice means is such that the average pressure drop across the restricted orifice means is in the range of 0.4 to 1.2 pounds per square inch during normal substantially steady state operation of the refrigeration system in which the compressor unit is utilized.

20. The hermetic compressor unit of claim 17 in which said minimum cross-sectional area of said restricted orifice means is such that the overall efficiency of the hermetic compressor unit during substantially steady state operation in the refrigeration system is decreased less than 5 percent due to said restricted orifice means in said suction pipe.

21. The hermetic compressor unit of claim 17 in which said minimum cross-sectional area of said restricted orifice means is generally circular and has an effective diameter D in inches substantially equal to

$$\frac{VO}{K(P)\left[\frac{1}{2+\frac{BP}{AP+1}}\right]}$$

where K equals 267, A equals 0.24, B. equals 0.12, V equals the specific volume in cubic feet per pound mass of the refrigerant immediately upstream of the orifice, P equals the pressure drop across the orifice in pounds per square inch, and Q equals the mass flow rate through the compressor in pounds per hour where the refrigeration system is operating under the conditions of Section 6–2 of Standard 520 of the American Refrigeration Institute, with evaporator temperature at −10°F., gas temperature entering the compressor at about 90°F., compressor ambient temperature at about 90°F., liquid temperature at the expansion valve at about 90°F., and condensing temperature at about 130°F.

22. A method of decreasing the peak gas pumping torque imposed on a hermetically sealed positive displacement compressor to match the compressor with a vapor compressor refrigeration system having a condenser, expansion valve means and an evaporator serially connected with said compressor and containing a refrigerant, and wherein a low to medium back pressure exists at the outlet of the evaporator of the system during normal operation thereof, said method comprising the steps of providing a compressor having a volumetric displacement rating in excess of that required by said system, selecting restricted orifice means calibrated to reduce the pumping capacity of said compressor to match the requirements of said system with said compressor, and locating said orifice means in said system such that the outlet of the evaporator communicates with the suction inlet of the compressor through said orifice means whereby said compressor is matched with said refrigeration system.

23. The method of claim 22 wherein said orifice means has a fixed minimum cross-sectional area and is located between the outlet of the evaporator and the suction inlet of the compressor so as to continuously couple the same through said orifice means.

24. A vapor compression refrigeration system containing a refrigerant and having a low to minimum back pressure at the outlet of an evaporator of the system during normal operation of said system, said system further having the inlet of a hermetically sealed positive displacement compressor coupled to the outlet of the evaporator exclusively via orifice means having a fixed minimum cross-sectional area calibrated to limit the maximum mass flow rate into the compressor after it is first started and an abnormal back pressure occurs with the evaporator at an elevated temperature compared to its normal operating temperature compared to its normal operating temperature whereby the torque required to operate the compressor is decreased compared to the torque required to operate the compressor under the same conditions without said orifice means.

25. The refrigeration system of claim **24** wherein said hermetically sealed positive displacement compressor comprises an electric motor, a positive displacement gas pump driven by said motor, and a casing encapsulating and hermetically sealing said motor and said gas pump therein, said electric motor being incapable of producing sufficient torque to operate said gas pump with said evaporator at said elevated temperature in the absence of said orifice means and being capable of developing sufficient torque to operate said gas pump with said orifice means.

26. The refrigeration system of claim **24** wherein said elevated temperature of said evaporator is sufciciently greater than said normal operating temperature to at least double the density of the gaseous refrigerant at the outlet of said evaporator compared to the density of the gaseous refrigerant at said outlet of said evaporator at said normal operating temperature.

27. The refrigeration system of claim **25** wherein said elevated temperature of said evaporator is sufficiently greater than said normal operating temperature to at least double the density of the gaseous refrigerant at the outlet of said evaporator compared to the density of the gaseous refrigerant at said outlet of said evaporator at said normal operating temperature.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,763,659 Dated October 9, 1973

Inventor(s) Paul B. Hover

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 5, line 11, "Jhe" should be --the--.

In the formula appearing in columns 7, 12 and 14, that portion thereof reading "VO" should read --VQ--.

Column 12, line 47, "167" should be --267--.

Column 13, line 21, cancel "pipe" and insert --tube means--; line 42, cancel "pipe" and insert --tube means--; line 45, cancel "pipe" and insert --tube means--; line 57, after "tube" insert --means--.

Column 14, line 14, cancel "pipe" and insert --tube means--.

Signed and sealed this 25th day of June 1974.

(SEAL)
Attest:

EDWARD M.FLETCHER,JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents