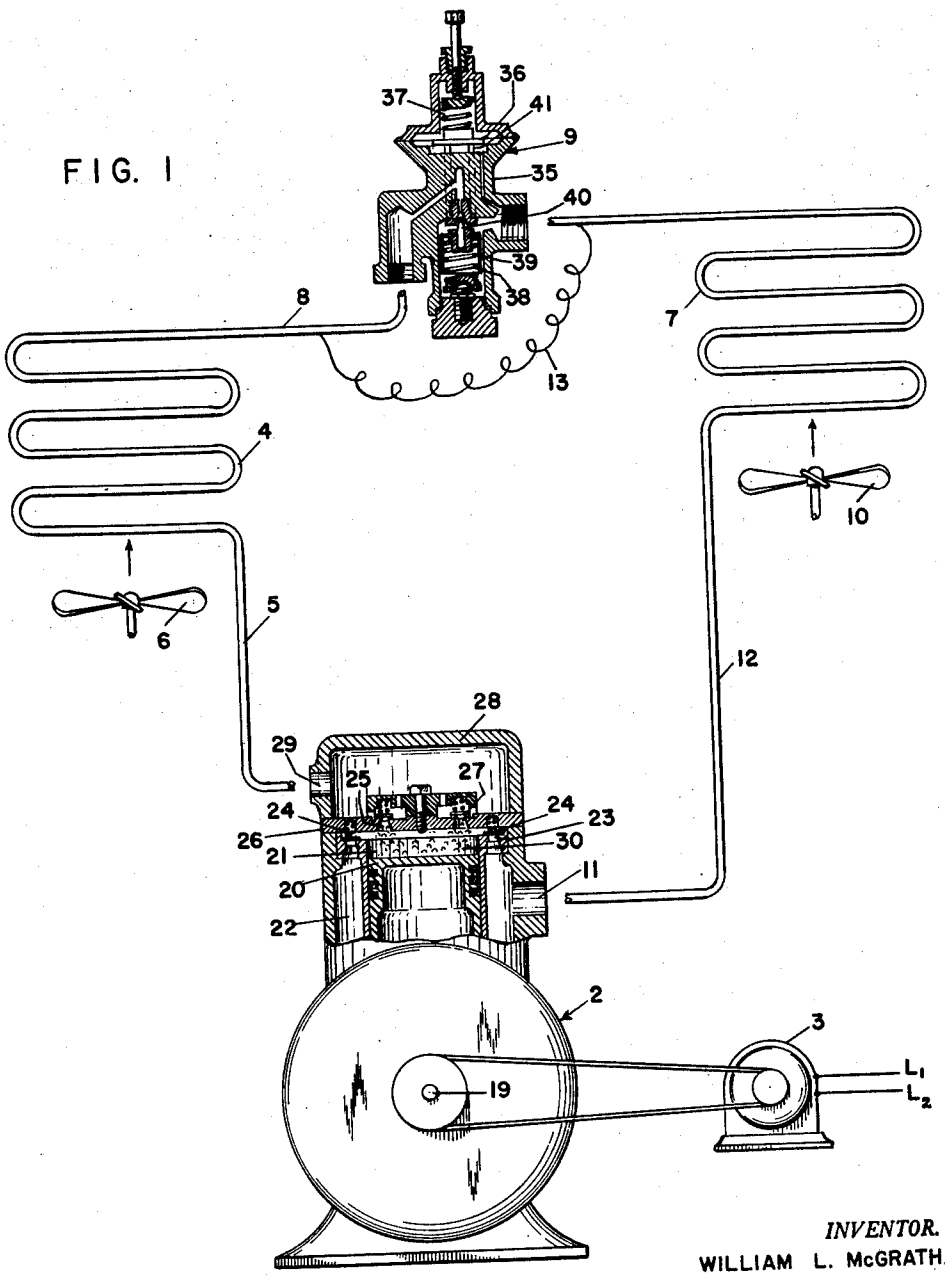
Dec. 15, 1959 W. L. McGRATH 2,916,894

REFRIGERATION SYSTEM

Filed Sept. 17, 1956 5 Sheets-Sheet 1

FIG. 1



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Dec. 15, 1959 W. L. McGRATH 2,916,894

REFRIGERATION SYSTEM

Filed Sept. 17, 1956 5 Sheets-Sheet 2

COMPRESSOR MOTOR POWER.

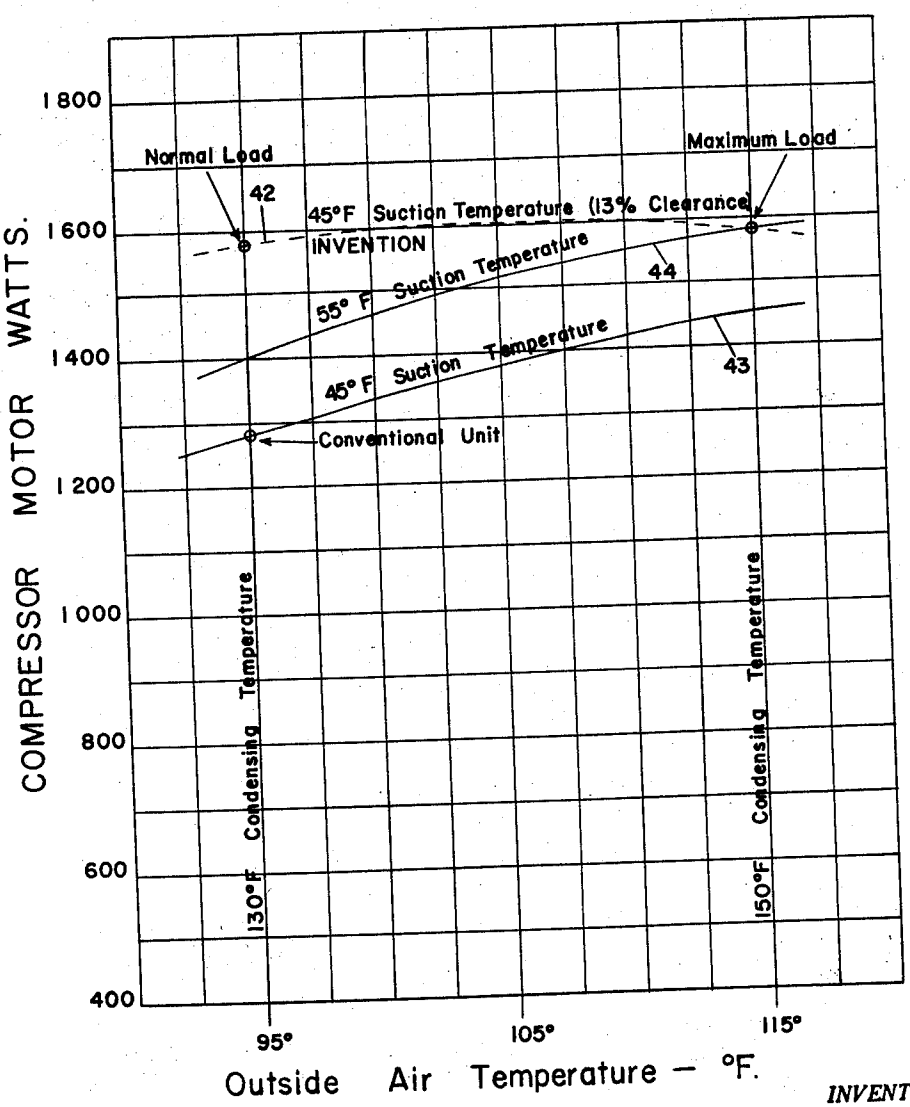


FIG. 2

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REFRIGERATION SYSTEM

Filed Sept. 17, 1956 5 Sheets-Sheet 3

HIGH CAPACITY UNIT.

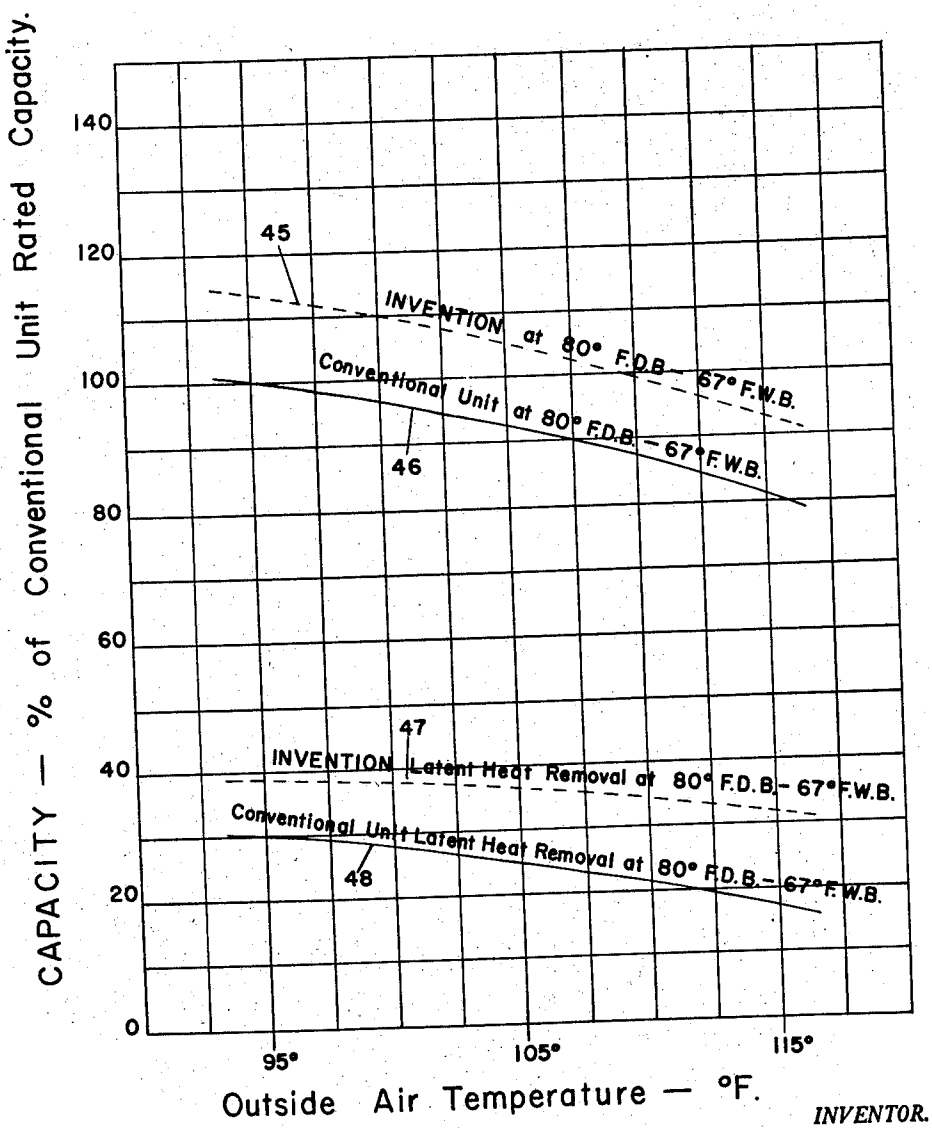


FIG. 3

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Dec. 15, 1959 W. L. McGRATH 2,916,894

REFRIGERATION SYSTEM

Filed Sept. 17, 1956 5 Sheets-Sheet 4

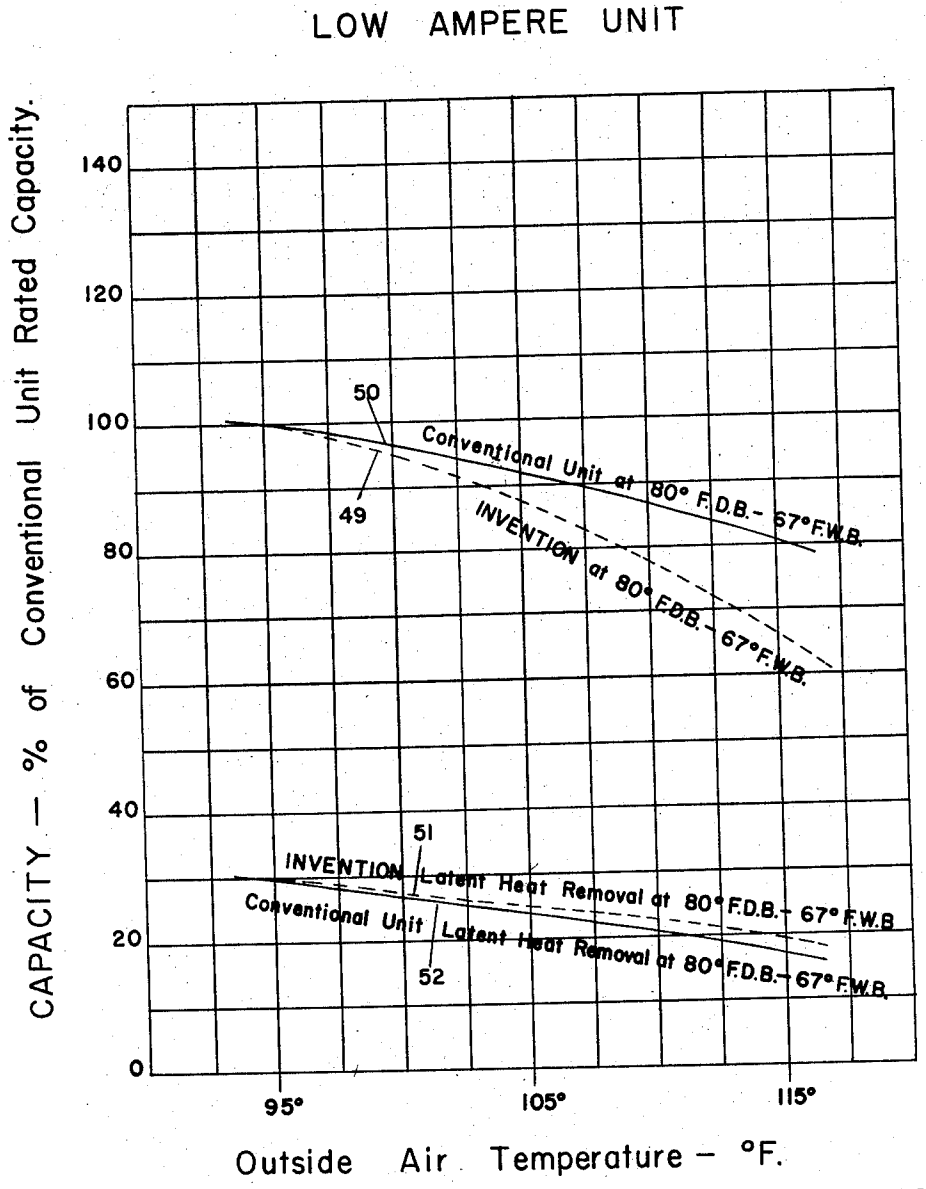


FIG. 4

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REFRIGERATION SYSTEM

Filed Sept. 17, 1956 5 Sheets-Sheet 5

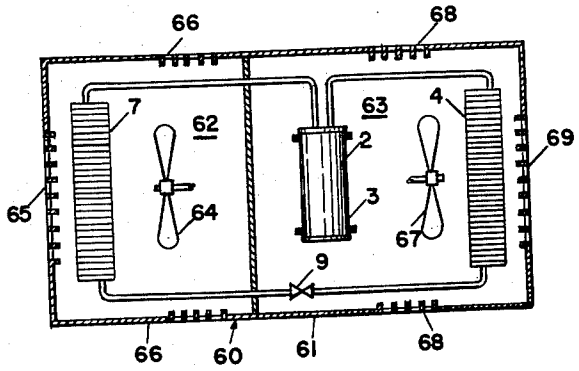


FIG. 5

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United States Patent Office 2,916,894 Patented Dec. 15, 1959

2,916,894

REFRIGERATION SYSTEM

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Application September 17, 1956, Serial No. 610,072

9 Claims. (Cl. 62—196)

This invention relates to refrigeration systems and, more particularly, to a refrigeration system and method of operation of the same as employed in self-contained air conditioning units, for example, of the room cooler type to provide units of greater refrigerating capacity as compared with conventional units employing the same compressor motor or units having substantially the same refrigerating capacity as compared with conventional units but employing a compressor motor requiring less current under maximum operating conditions. The term "conventional unit" is employed to describe a self-contained air conditioning unit embodying a refrigeration system including a reciprocating compressor having a cylinder clearance in the order of 3% to 5% and provided with a capillary tube or thermostatic expansion valve to regulate refrigerant supply to the evaporator.

Air conditioning units, for example, self-contained air conditioning units of the room air conditioner type will normally operate when in use with room temperatures in the order of 75° F. to 80° F. and with outside temperatures in the range of 90° F. to 100° F. However, the unit must also be designed to start up and pull down occasionally under very high load conditions, as for example, when the outside temperature is in the range of 105° F. to 115° F. and the inside temperature may be of the order of 95° F. The current drawn by the unit under such conditions is substantially higher than that normally required but, since such conditions can occasionally occur, the nameplate current of the unit is normally derived from the current drawn at the extreme condition.

Further, since such conditions will occasionally occur, the current input to the motor must, therefore, be larger than would be normally required and the fuses, branch wiring and the electrical service to the space being conditioned must all be proportionately larger than required under normal operating conditions. Where a number of such machines are in a particular locality, it also follows that the size of the wire in the distribution system and the capacity of the transformers and other equipment in the electrical distribution system must be substantially larger than is actually required under normal operating conditions.

It can be shown that if the work load imposed on the compressor can be maintained substantially constant as between the normal operating conditions and such maximum conditions, then the power input to the compressor motor may be prevented from rising when operated at the maximum conditions. This permits a self-contained air conditioning unit to be created which has a substantially lower maximum current requirement and the correspondingly lower nameplate current. This in turn will, in many cases, permit the installation of a unit with adequate capacity on existing wiring circuits where otherwise rewiring would be required. The use of air conditioning units of the type described will markedly reduce the total installed kva. capacity required of the electrical distribution system supplying such loads.

I have found that it is possible to provide a refrigeration system for air conditioning units preferably of the self-contained type for which the motor during normal operating conditions is actuated with a substantially constant full load thus providing a considerable increase in refrigerating capacity of the unit for a given motor size or, if desired, the current draw of the motor may be decreased to provide about the same refrigeration capacity as conventional units with larger motors thus providing a unit which draws less amperes during use at normal conditions.

In contrast to refrigeration systems heretofore employed in self-contained air conditioning units, the refrigeration system may be designed to provide a substantially constant load on the motor employed to actuate the reciprocating compressor of the system. In systems heretofore employed, the load imposed on the motor varied considerably depending upon variation in ambient air temperature and the evaporator load. The provision of a substantially constant load on the motor provides about the same refrigeration capacity during normal operating conditions as provided by conventional units heretofore employed, permitting a motor to be used which draws less amperes and obviating the excessive costs heretofore frequently required for new wiring upon installation.

The chief object of the present invention is to provide a refrigeration system for an air conditioning unit preferably of the self-contained type which obviates the disadvantages present in conventional units.

An object of the present invention is to provide a refrigeration system for a self-contained air conditioning unit in which a substantially constant load is imposed upon the motor employed to actuate the reciprocating compressor of the system so that a substantially equal power input to the motor is maintained at normal and maximum operating conditions.

A further object is to provide a refrigeration system of increased efficiency in which the increased efficiency may be employed to impart greater refrigerating capacity to an air conditioning unit in which it is employed or to provide substantially the same refrigerating capacity as a conventional unit but employing a compressor motor requiring less current under maximum operating conditions.

A still further object is to provide a refrigeration system employing a reciprocating compressor having a predetermined clearance and a control responsive to the saturation condition of the refrigerant in the low side of the system which permits a refrigerating capacity to be obtained materially greater under normal operating conditions than the refrigerating capacity provided by conventional units heretofore employed. Clearance may be defined as that volume of the compressor cylinder and discharge ports open directly to the cylinder which is filled with compressed gaseous refrigerant upon the termination of the compression stroke (upward stroke) of the compressor piston.

A still further object is to provide a refrigeration system employing a reciprocating compressor having a predetermined clearance and a constant pressure valve, the power input to the compressor motor being maintained substantially equal at normal and maximum operating conditions. Other objects of the invention will be readily perceived by reference to the following description.

This invention relates to a refrigeration system comprising in combination an electric motor adapted to actuate a reciprocating compressor, a reciprocating compressor having a predetermined cylinder clearance sufficient to contain an amount of compressed gaseous refrigerant such that, upon the downward stroke of the compressor piston, re-expansion fo the compressed gas-

eous refrigerant occurs to a degree that the total weight of gaseous refrigerant entering the compressor cylinder is varied in an amount to maintain the work load imposed on the motor substantially constant, a condenser, a control responsive to the saturated condition of the refrigerant in the low side of the system to substantially maintain a predetermined suction pressure at the inlet to the compressor, and an evaporator, said compressor, condenser, control and evaporator being placed in the system in such order.

The attached drawings illustrate a preferred embodiment of the invention, in which

Figure 1 is a diagrammatic view of a refrigeration system embodying the present invention;

Figure 2 is a graph illustrating the compressor motor power requirements as employed in the present invention compared with compressor motor power employed in conventional units;

Figure 3 is a graph illustrating capacity of the refrigeration system of the present invention compared to the capacity of a conventional unit;

Figure 4 is a graph illustrating capacity of the system of the present invention employing a compressor motor of lower current requirement than employed in the conventional unit with which it is compared; and

Figure 5 is a diagrammatic view of a self-contained air conditioning unit of the room cooler type including the refrigeration system of the present invention.

Referring to the attached drawings, there is shown the refrigeration system of the present invention which includes a reciprocating compressor **2** provided with a predetermined clearance as hereinafter explained, actuated by an electric motor **3** connected to a suitable source of current. It will be appreciated the reciprocating compressor and motor may be combined to form a hermetic unit if desired.

The compressor **2** is connected to a condenser **4** by line **5**, fan **6** passing air at an ambient temperature through the condenser to condense refrigerant therein. Condenser **4** is connected to evaporator **7** by line **8**. A control **9** responsive to the saturation condition of the refrigerant in the evaporator **7**, such as a constant pressure valve, is placed in line **8** to regulate flow of liquid refrigerant to evaporator **7** thereby maintaining a desired evaporator or suction pressure at the inlet of the compressor. It will be understood, of course, that valve **9** may be responsive to the saturated suction temperature instead of the saturated suction pressure since the conditions are equivalent. Hence, I have recited that the valve is responsive to the saturation condition of the refrigeratant in the evaporator.

It will be appreciated evaporator pressure and suction pressure are substantially equivalent since the pressures are the same except for any slight pressure drop through the evaporator coil.

A fan **10** passes air from an area being conditioned through evaporator **7**, the air being cooled by its heat exchange relation with refrigerant in the evaporator. Evaporator **7** is connected to the inlet **11** of compressor **2** by suction line **12**. If desired, to permit pressures in the system to equalize upon shutdown, thus permitting the compressor to start unloaded, a capillary tube bypass **13** may be provided about valve **9**. It will be appreciated a groove or hole may be provided in the valve seat of valve **9** to achieve the same result.

Valve **9** substantially maintains a predetermined suction or evaporator pressure at the inlet **11** of compressor **2**. Compressor **2** must be designed to provide a predetermined cylinder clearance, regardless of the number of cylinders employed, sufficient to contain an amount of compressed gaseous refrigerant such that, upon the downward stroke (suction stroke) of the compressor piston, re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant entering the compressor cylinder is varied in an amount to

maintain the work load imposed on the motor **3** substantially constant. In other words, the total weight of gaseous refrigerant drawn in by the compressor is varied inversely with the pumping head to an extent sufficient to maintain the work load imposed on the motor substantially constant. Under these circumstances, current is supplied to the motor directly proportional to the evaporator or suction pressure and inversely proportional to the amount of clearance provided in the compressor, thus maintaining the work load on the motor substantially constant throughout normal operating ranges.

A constant motor load is maintained by holding substantially constant the product of the refrigerant weight flow and the pumping head of the compressor. The clearance volume cooperates to do this. Hence the weight of refrigerant trapped there is proportional to the pumping head. When the refrigerant volume expands to the fixed suction pressure in the system the amount of space in the cylinder occupied by this refrigerant gas is proportional to the weight of refrigerant gas trapped and hence proportional to the original pumping head. The remaining volume in the cylinder to be filled with fresh refrigerant gas then becomes inversely proportional to the pumping head and the product of refrigerant weight flow and pumping head remains constant.

Compressor **2** as shown in Figure 1 includes a crankcase (not shown), a shaft **19** adapted to be connected to motor **3**, and a crank (not shown) mounted on the shaft connected to a piston **20** to reciprocate piston **20** in cylinder **21**. Cylinder **21** is provided with a suction manifold **22** having a port **23** to cylinder **21**. Suction valve **24** regulates flow of gaseous refrigerant into cylinder **21**. Ports **25** extend through valve plate **26** and are closed by discharge valve **27** which regulates flow of compressed, gaseous refrigerant from cylinder **21** into the head **28** of the compressor. Compressed gaseous refrigerant flows from head **28** of the compressor through outlet **29** and is supplied through line **5** to condenser **4**. Suction manifold **22** is connected to suction line **12** by inlet **11**, gaseous refrigerant or vapor formed in the evaporator being supplied to the cylinder therethrough.

Cylinder **21** is provided with a predetermined clearance shown at **30** sufficient to contain an amount of compressed gaseous refrigerant such that upon the downward stroke of the compressor piston **20** re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant entering the compressor cylinder **21** is varied in an amount to maintain the work load imposed on the motor **3** substantially constant at normal operating and maximum conditions thereby substantially maintaining equal power input to the motor at such conditions.

While it is necessary in all compressors to provide some amount of clearance to prevent impact between the compressor piston and the valve plate, practice in the industry has been to make such space as small as possible since in conventional units capacity of the system decreases upon an increase in clearance. In the present invention the optimum cylinder clearance is about 11% of the cylinder volume considering the cylinder volume as being the volume in the cylinder between the valve plate and the piston at the end of its downward stroke plus the volume of the valve ports open directly to the cylinder. The advantages of the present invention may be obtained if the clearance of the compressor cylinder falls within the range of about 7% to about 15%.

Constant pressure valve **9** comprises a housing **35** containing a diaphragm **36** which has evaporator pressure on the lower side of it and atmospheric pressure above it, these pressures being balanced and adjusted by means of springs **37**, **38**. For a given pressure setting, the valve opens just enough that the flow through it balances the vapor removed by the compressor. If the evaporator pressure starts to decrease, pressure below the diaphragm **36** is reduced and the adjusting spring **37** pushes down.

This motion is transferred to cage **39** and needle **40** by push rods **41** to move the needle toward an open position. This allows more high pressure refrigerant to flow into the evaporator thus increasing the pressure to that required.

Any increase in pressure similarly raises the diaphragm **36** and closing spring **38** moves the needle **40** towards a closed position permitting less refrigerant to flow into the evaporator thus decreasing the pressure to the pressure it is desired to maintain.

In Fig. 2 there is shown a graph which compares the compressor motor power of the refrigeration system of the present invention and a conventional unit heretofore used in the industry. In the graph, the motor employed in both units is the same and outside air temperature is plotted against compressor motor watts. The graph illustrates the differences between normal operating conditions (condensing temperature at 130° F.) and maximum load conditions (condensing temperature at 150° F.).

Referring to the graph, it is noted that the load imposed upon the motor used to actuate the compressor of the present invention is substantially constant as indicated at **42** while the load imposed upon the same motor employed in the conventional unit varies greatly as indicated at **43**. There is also plotted a conventional unit as shown at **44** at a pressure corresponding to 55° F. suction temperature as compared with units at pressures corresponding to 45° F. suction temperature as shown at **42, 43**. Under such conditions the present invention at normal operating conditions (130° F. condensing temperature) draws slightly less than 1580 watts as compared to the conventional unit which draws about 1280 watts under the same conditions. It is therefore clear that the capacity of the compressor of this invention may be increased approximately 23% over that of the conventional unit and still draw no more power under maximum load conditions. As a corollary a motor drawing less current may be employed in the present invention with a proportional decrease in maximum power requirements while affording substantially the same refrigerating capacity.

Figure 3 indicates this gain in refrigerating capacity as reflected in unit performance. Figure 3 is a graph in which capacity is plotted against outside air temperature. The unit of the present invention is compared with a conventional unit. In both cases, the same sizes of motors are used with the same quantities of heat exchange surface. The same air quantities were removed through the heat exchangers of both units. In both units, a suction temperature of 42° F. was maintained. Capacities of the units are plotted as the percentage of rated capacity of a conventional unit.

The units were tested at 80° F. dry bulb temperature and at 67° F. wet bulb temperature. As shown by comparison between lines **45, 46** there is an increase in capacity employing the present invention at code conditions roughly 14% greater than the conventional unit tested, and this increase is provided substantially throughout the range of operating conditions shown.

The graph also illustrates, as indicated by lines **47, 48** that the latent heat capacity of the unit of the present invention is considerably greater than the latent heat capacity of a conventional unit, thereby producing lower relative humidities, more comfortable conditions and a lower effective temperature in the area being conditioned.

In Figure 4 there is illustrated a graph depicting performance of a low ampere unit constructed in accordance with the present invention compared with a conventional unit. In the tests illustrated, in the unit embodying the present invention, a motor drawing approximately 17% less locked rotor amperes and 17% less maximum load amperes than the motor employed in the conventional unit was used. The coil surfaces and air quantities in both cases were the same. In each case, a suction temperature of 45° F. was employed. The tests were conducted at 80° F. dry bulb temperature and 67° F. wet bulb temperature.

Comparing a unit embodying the present invention with the conventional unit as shown by lines **49, 50** respectively it will be observed the capacity or performance of both units is the same for all practical purposes during normal operating conditions. An exception in performance of these units not indicated in the graph resides in the pull-down characteristic from a condition of, say, 95° F. in the area being conditioned when the units are first placed in operation.

With the conventional unit, a high suction pressure would prevail which will yield more sensible heat capacity during the pull-down, but less latent heat capacity. Therefore, it may be expected that the unit embodying the present invention would take somewhat longer to pull down the dry bulb temperature from a given condition in the area being conditioned but would pull down the absolute humidity faster. With a unit embodying the present invention, the area being conditioned will reach a comfortable humidity level faster than it would with a conventional unit since it normally takes longer to pull the moisture level down to equilibrium than it does to pull the temperature level down. Thus, under these conditions the pull down performance of the unit of the present invention may be deemed superior to that of the conventional system.

Considering the operation of the refrigeration system, compressor **2** compresses gaseous refrigerant and forwards the compressed gaseous refrigerant to condenser **4**. The compressed gaseous refrigerant is condensed in condenser **4** and is supplied to evaporator **7**, the amount of liquid refrigerant being supplied to evaporator **7** being regulated by constant pressure valve **9** to maintain a predetermined evaporator or suction pressure. Air to be cooled is passed in heat exchange relation with refrigerant in evaporator **7**, the refrigerant being evaporated, vapor so formed passing through suction line **12** to the compressor **2**.

It will be appreciated that in compressor **2** at the end of the compressor stroke (upward stroke of the piston) a substantial amount of compressed gaseous refrigerant is trapped in the clearance spaces between the piston and the discharge valve **27**. The weight of the gas varies with the head pressure with a particular clearance. As the piston begins its downward stroke, this compressed gaseous refrigerant re-expands and fills the cylinder chamber so that only a certain additional amount of gaseous refrigerant from the suction manifold may enter the cylinder. The amount of compressed gaseous refrigerant remaining in the cylinder varies with the clearance of the compressor; as recited above, the optimum clearance is about 11%. The optimum clearance may vary slightly depending upon the refrigerant used in the system.

It can be shown that the power requirements of a compressor of fixed displacement and utilizing a given refrigerant, will vary in proportion to this factor

$$\left(1+C-CR^{\frac{1}{n}}\right)\left(R^{\frac{n-1}{n}}-1\right)P_o$$

where

C=the clearance fraction expressed in percent of cylinder volume
R=the compression ratio
P_o=the suction pressure
n=compression exponent

In constructing the system which is the subject of this invention, it is required that the power requirements under two different conditions of operation namely the normal operating conditions and the maximum operating conditions be made substantially identical.

This is accomplished by the cooperative effect of two factors which are discreetly controlled. First, the use of

the pressure limiting expansion valve assures that the suction pressure will be substantially the same at both operating conditions. Having thus fixed the suction pressure, it is clear from inspection of the above expression that a value of clearance can be determined which will result in the above expression having identical value for two different values of compression ratio. It has been found from such analysis and confirmed by extensive testing that for the normal air conditioning applications involving air cooled condensers, the value of the clearance factor in the range of .08 to .15 will give substantially equal power at both the normal operating condition and the maximum operating condition as previously described.

Let us assume this system is being operated with a constant evaporator load. Assuming condensing pressure rises due to an increase in ambient temperature, the weight of trapped gas in the clearance space increases.

Since condenser pressure has increased it will be appreciated less total weight of refrigerant is circulated. Since the compressor must pump against a higher head pressure a greater weight of refrigerant is trapped in the cylinder clearance spaces. As the compressor piston is retracted (moved downward in the suction stroke) the trapped gas expands so that a less weight of refrigerant is drawn into the compressor cylinder. It will be appreciated the piston has been retracted to a greater extent before the pressure of the trapped gas reaches the constant suction pressure leaving a reduced cylinder volume for accommodation of suction gas. Even though the volume of refrigerant being compressed on a particular stroke of the compressor piston is substantially the same less total weight of refrigerant is being circulated in the system, so that the power input to the motor remains substantially equal.

If the cylinder clearance were not designed to contain an adequate amount of compressed gaseous refrigerant, then a substantially constant work load could not be imposed on the electric motor actuating the compressor. Under these conditions, however, the clearance provided is such that re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant entering the compressor cylinder is varied in an amount to maintain the work load imposed on the electric motor substantially constant.

Let us consider operation of the system assuming the evaporator load remains constant but that condensing temperature and pressure decreases. Since condenser pressure has decreased it will be appreciated a greater total weight of refrigerant is circulated. The compressor now pumps against the lower head pressure so that a less total weight of refrigerant is trapped in the cylinder clearance spaces. As the compressor piston is retracted (moved downward in the suction stroke) the trapped gas expands. However, since less gas by weight had been trapped a greater weight of refrigerant is drawn into the compressor cylinder even though the total weight of refrigerant being compressed on a particular stroke of the compressor piston is substantially the same, a greater total weight of refrigerant is being circulated, the compressor does substantially the same amount of work so that the power input to the motor remains substantially constant, the load imposed on the motor being substantially constant.

Under some circumstances, it may be desirable to place an accumulator in the suction line since the use of an accumulator assures that no liquid refrigerant would be supplied to the compressor under extreme conditions of light evaporator load.

The refrigerant employed in the system is preferably the azeotrope of dichlorodifluoromethane and unsymmetrical difluoroethane. It will be appreciated, if desired, the refrigeration system, including the compressor, may be designed to employ dichlorodifluoromethane as a refrigerant and in such case the use of the azeotrope described above in such system increases the capacity of

the system, increases effective displacement of the compressor to compensate for the increased clearance and thus provides substantially the same capacity with the decrease in power requirement described.

It will be appreciated that in all cases within normal operating ranges the motor actuating the compressor has substantially the same load imposed thereon. It is impossible for an overload to be placed on the motor since valve 9 maintains a substantially constant evaporator or suction pressure. While the system will not operate as efficiently when the load imposed on the evaporator is below normal operating ranges, this is not important since under those circumstances the system itself will be shut down.

An advantage of this present system resides in the fact that as outside temperature rises above 115° F. the power consumed may actually drop off slightly rather than increasing as occurs with a conventional unit. Thus, the system of the present invention is particularly desirable for use in desert areas, for example, where excessive temperatures are encountered. Due to the use of the constant pressure valve, the suction pressure is held substantially constant so that possibility of freeze-ups is greatly reduced. The system of the present invention provides superior control of relative humidity since under conditions of high dew point the suction pressure does not rise and therefore greater dehumidification and lower relative humidity is obtained.

In Figure 5 I have illustrated a self-contained air conditioning unit 60 of the room cooler type embodying the refrigeration system of the present invention. Referring to Figure 5 there is shown a housing 61 forming an evaporator compartment 62 and a machine compartment 63. The evaporator 7 is placed in compartment 62; the compressor 2, compressor motor 3, condenser 4 and constant pressure valve 9 are placed in compartment 63. It will be noted that compressor 2 and motor 3 have been combined to form a hermetic compressor-motor unit. A fan 64 is provided to pass air from the area being conditioned through the evaporator in heat exchange relation with refrigerant therein to cool the air. The fan discharges the cool air through outlet 65 into the area being cooled. It draws air from the area being conditioned into compartment 62 through inlet 66. A condenser fan 67 is provided to pass exterior air through condenser 4 to remove heat from the system. Fan 67 draws the exterior air into compartment 63 through inlet 68 and discharges the heated air through outlet 69 exteriorly of the room being conditioned. The refrigeration system functions as described previously.

The present invention provides a self-contained air conditioning unit which may utilize the same compressor motor as used in conventional units and will provide a greater refrigerating capacity in the unit than is capable of being provided by the conventional unit. If desired, in the present invention, a compressor motor drawing less current may be provided which will permit substantially the same capacity to be obtained as is obtained in conventional units.

While I have described a preferred embodiment of the present invention, it will be understood the present invention is not limited thereto since it may be otherwise embodied within the scope of the following claims.

I claim:

1. In a refrigeration system, the combination of an electric motor adapted to actuate a reciprocating compressor, a reciprocating compressor having a predetermined cylinder clearance sufficient to contain an amount of compressed gaseous refrigerant such that, upon the downward stroke of the compressor piston, re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant entering the compressor cylinder is varied in an amount to maintain the work load imposed on the motor substantially constant, a condenser, a control responsive to the satura-

tion condition of the refrigerant in an evaporator to substantially maintain a predetermined pressure at the inlet to the compressor, an evaporator, said compressor, condenser, control and evaporator being placed in the system in such order, and means bypassing the port of the control to permit pressure equalization in the system during shutdown.

2. In a refrigeration system, the combination of an electric motor adapted to actuate a reciprocating compressor, a reciprocating compressor having a fixed cylinder volume and having a predetermined cylinder clearance, the weight of gaseous refrigerant trapped in the clearance being proportional to the pumping head, the cylinder clearance being such that, upon the downward stroke of the compressor piston, re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant drawn in by the compressor is varied inversely with the pumping head to maintain the work load imposed on the motor substantially constant, the total volume of gaseous refrigerant being compressed on each stroke of the piston being substantially the same, a condenser, a constant pressure valve to substantially maintain a predetermined pressure at the inlet to the compressor, and an evaporator, said compressor, condenser, constant pressure valve and evaporator being placed in the system in such order, the compressor clearance falling within the range of 7–15% of the volume of the cylinder and discharge ports.

3. A refrigeration system according to claim 2 in which the optimum compressor clearance is about 11% of the volume of the cylinder and discharge ports.

4. A refrigeration system according to claim 3 in which the refrigerant consists of the azeotrope of dichlorodifluoromethane and unsymmetrical difluoroethane.

5. In a self-contained air conditioning unit, the combination of a housing comprising a first compartment and a second compartment separated by a partition, a refrigeration system in the housing, said refrigeration system including a compressor having a fixed cylinder volume and having a predetermined cylinder clearance the weight of gaseous refrigerant trapped in the clearance being proportional to the pumping head, the cylinder clearance being such that, upon the downward stroke of the compressor piston, re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant drawn in by the compressor is varied inversely with the pumping head to maintain the work load imposed on the electric motor substantially constant, an electric motor to actuate the compressor, a condenser, a constant pressure valve to substantially maintain a predetermined pressure at the inlet to the compressor and an evaporator, the compressor, condenser, constant pressure valve and evaporator being placed in the system in such order, the evaporator of the system being placed in the first compartment and the condenser, compressor and electric motor being placed in the second compartment, means to pass a stream of air to be conditioned through the evaporator in heat exchange relation with refrigerant therein to condition the air and to discharge the stream of conditioned air into an area being treated, and means in said second compartment to pass a stream of air through the condenser in heat exchange relation with refrigerant passing therethrough to remove heat from the system and to discharge the hot air exteriorly of the area being treated.

6. In a refrigeration system, the combination of an electric motor adapted to actuate a reciprocating compressor, a reciprocating compressor having a fixed cylinder volume and having a predetermined cylinder clearance, the weight of gaseous refrigerant trapped in the clearance being proportional to the pumping head, the cylinder clearance being such that, upon the downward stroke of the compressor piston, re-expansion of the compressed gaseous refrigerant occurs to a degree that the total weight of gaseous refrigerant drawn in by the compressor is varied inversely with the pumping head to maintain the work load imposed on the motor substantially constant, the total volume of gaseous refrigerant being compressed on each stroke of the piston being substantially the same, a condenser, a constant pressure valve to substantially maintain a predetermined pressure at the inlet to the compressor, and an evaporator, said compressor, condenser, constant pressure valve and evaporator being placed in the system in such order, the optimum compressor clearance being that value which yields the same power requirement at condensing temperatures in the range of 125° F.–130° F. as is required at condensing temperatures within the range of 145° F.–150° F.

7. A self-contained air conditioning unit according to claim 5 in which the compressor of the refrigeration system possesses a clearance falling within the range of 7–15% of the volume of the cylinder and discharge ports.

8. A self-contained air conditioning unit according to claim 7 in which the optimum compressor clearance is about 11% of the volume of the cylinder and discharge ports.

9. A self-contained air conditioning unit according to claim 8 in which the refrigerant utilized in the refrigeration system consists of the azeotrope of dichlorodifluoromethane and unsymmetrical difluoroethane.

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