Nov. 8, 1966 J. T. BYRON 3,283,524

REFRIGERATION SYSTEM

Filed March 17, 1964

2 Sheets-Sheet 1

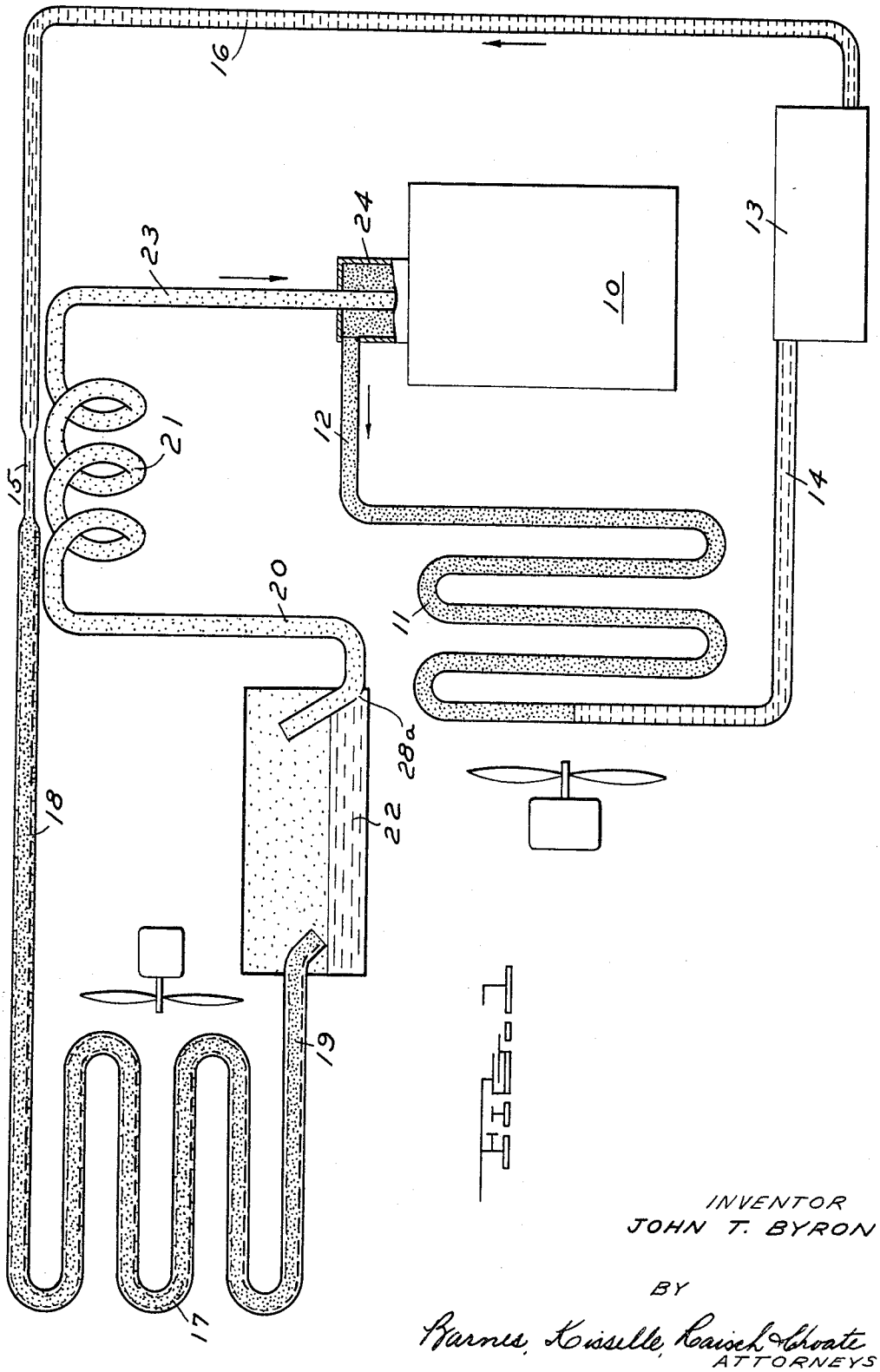


FIG. 1

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REFRIGERATION SYSTEM

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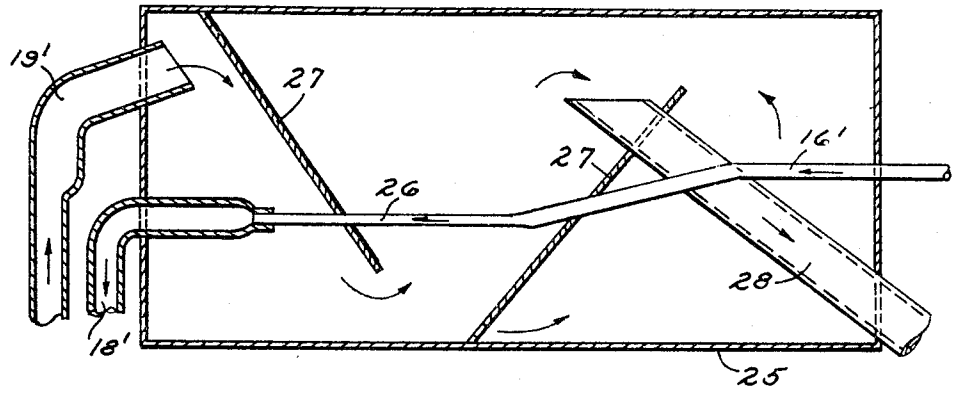


FIG. 4

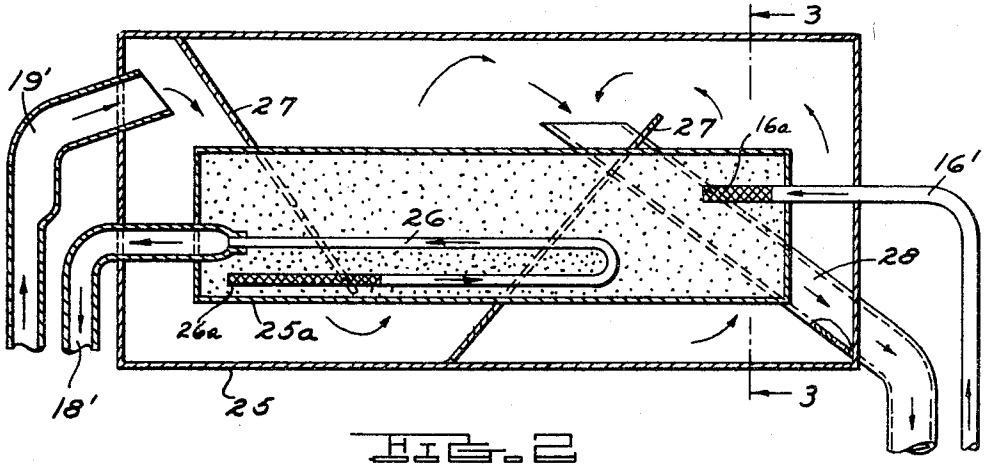


FIG. 2

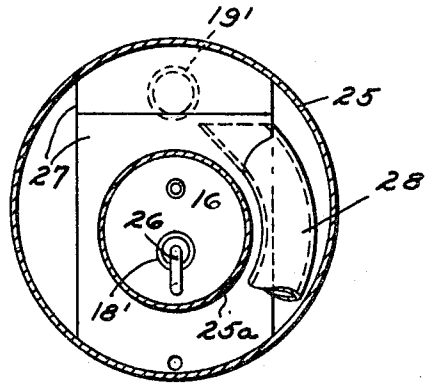


FIG. 3

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REFRIGERATION SYSTEM

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5 Claims. (Cl. 62—115)

This application is a continuation-in-part of my prior application Serial No. 241,937, filed December 3, 1962, now abandoned.

This invention relates to refrigeration systems and particularly to commercial systems of the expansion valve or capillary tube type.

It is a further object of the invention to provide such a refrigeration system which utilizes a flow controlling device having a larger opening than has been heretofore thought possible for an equivalent capacity system.

It is a further object of the invention to provide a refrigeration system which utilizes a compressor of much less capacity for refrigeration spaces of the same size.

It is a further object of the invention to provide such a refrigeration system wherein the compressor is operated at a much lower head pressure, and a much higher suction pressure.

It is an object of the invention to provide a refrigeration system which has a greater efficiency than has been heretofore possible with conventional systems.

It is a further object of the invention to provide such a system which is self-defrosting during the off cycle when refrigerating a space at above 32° F. and defrosts in a much shorter time than conventional systems utilizing some conventional defrosting arrangement when refrigerating a space below 32° F.

It is a further object of the invention to provide such a system which produces a mist type oiling for all internal parts of the compressor—pistons, valves, seals, crankshaft, wrist-pins, eccentrics, and bearings.

It is a further object of the invention to provide such a refrigeration system which is less likely to slugging of the compressor than conventional systems.

It is a further object of the invention to provide such a system which makes it possible to use a motor compressor unit which is less than half in size, less than half in cost, and less than half in power requirements than motor compressors heretofore thought to be needed for the same refrigerating capacity.

It is a further object of the invention to provide a refrigeration system which operates on a substantially lower pressure differential between the high side and the low side than has heretofore been thought possible.

Basically, the refrigeration system embodying the invention comprises a compressor, a condenser connected to the compressor, a liquid receiver connected to the condenser, a flow controlling device having a relatively large restriction preferably in the form of a restrictive tube or special "capillary" connected to the liquid receiver, a continuous coil evaporator connected to the restrictive tube, and an accumulator connected to the evaporator and to the inlet of the compressor. The accumulator has a volumetric capacity equal to or greater than the evaporator. When a tube is used as a restrictive device, it is much shorter in length and greater in cross-sectional area than conventional capillaries for equivalent capacity refrigeration systems. A heat exchanger may optionally be provided between the accumulator and compressor in heat exchange relationship to the restrictive device. It has been found that such a system permits the use of a compressor of much less capacity for the same refrigerating requirements. The compressor operates as a result of the system at a much lower head pressure and a higher back pressure so that the compressor does not become

so hot but, on the contrary, can be readily touched by hand. The system is completely selfdefrosting during the off cycle when used to refrigerate spaces to temperatures above 32° F. It has further been found that in such a system there is less likelihood of slugging. In addition, it appears that a fine mist of oil is passed to the compressor to lubricate the compressor so that the compressor tends to operate more uniformly.

While my system requires about three times the amount of refrigerant as that required in comparable systems, the combination of a relatively large restrictor, an accumulator, and a low pressure differential, results in an evaporator which, in operation, has substantially all its internal wall surface coated with liquid refrigerant uniformly throughout its length.

In the drawings:

FIG. 1 is a diagrammatic view of a refrigeration system embodying the invention.

FIG. 2 is a part sectional view of a combined restrictive device in the form of a tube, accumulator, heat exchanger and oil separator which may be used in the system shown in FIG. 1.

FIG. 3 is a sectional view taken along the line 3—3 in FIG. 2.

FIG. 4 is a part sectional view of a modified type of combined restrictive device in the form of a tube, accumulator, heat exchanger and oil separator which may be used in the system shown in FIG. 1.

Referring to FIG. 1, the refrigeration system embodying the invention comprises a compressor **10** which has an inlet and an outlet. A condenser **11** is connected to the outlet by a line **12** and a liquid receiver **13** is connected to the outlet of the condenser by a line **14**. The outlet of the liquid receiver is, in turn, connected to the inlet of a flow controlling device in the form of a tube or special "capillary" **15** by a line **16**. The outlet of the restrictive tube **15** is connected to the inlet of the evaporator **17**. The outlet of the evaporator is, in turn, connected to the inlet of an accumulator **22** by a line **19** and the outlet of the accumulator **22** is connected to the inlet of the compressor **10** by a line **20**. Optionally, a heat exchanger **21** is provided in the line **20**.

The accumulator **22** has a volumetric capacity equal to or greater than the evaporator. The liquid receiver **13** has a volumetric capacity sufficient to keep the condenser **11** free of most liquid at all times. My flow control device such as a restrictive tube or special "capillary" **15** is of larger capacity than conventional capillaries for use with comparable systems, that is, comparable size compressors. More specifically, the restrictive tube **15** is preferably much shorter in length or much greater in diameter than conventional capillaries. With this combination of a large flow control unit that flows liquid refrigerant into the evaporator, the evaporator, which is substantially a flooded continuous coil type, and the accumulator, the system requires a charge of refrigerant substantially three times that required in conventional systems of comparable refrigerating capacity.

In my system with a charge of refrigerant double and preferably triple in weight of the normal charge in the system, the compressed substantially saturated gas, as later understood, from compressor **10** flows through line **12** to the condenser **11** and collects in the receiver **13** as a liquid. From the receiver **13**, the liquid under relatively low head pressure is fed to the restrictive tube **15** which is at least twice as large as the conventional capillary heretofore thought to be needed for a system of comparable size. The liquid from the receiver **13** is fed to and through the restrictive tube **15** as a continuous and uninterrupted mass of liquid. From the restrictive tube **15** the liquid flows into the continuous coil evaporator **17**

where there will be some liquid refrigerant present throughout its entire length and, in fact, some liquid will flow into the accumulator **22**. There will be liquid in the bottom of each coil of the evaporator and there will be liquid backing substantially all the surfaces of the evaporator. Some compressed liquid refrigerant will be exploding throughout the length of the continuous tube evaporator so that the refrigerating effect is substantially uniform from inlet to outlet. Because the refrigerant in the evaporator is in the form of a highly saturated vapor, and the internal surfaces are backed with liquid, the heat transfer is at a maximum efficiency and results in an efficiency about five times as great as that obtained when the refrigerant in the evaporator is a dry gas. A highly but slightly less saturated vapor passes from the evaporator **17** to the accumulator **22** which collects oil and liquid refrigerant and prevents the liquid refrigerant from entering the suction pipe of the compressor **10**. From the accumulator **22** the highly but slightly less saturated vapor minus any slugs of liquid is returned to the compressor **10**.

Due to the fact that there is only a relatively moderate change of state or condition from a relatively low pressure liquid in the high side to a lower pressure saturated vapor in the entire low side, the back pressure remains relatively high and the head pressure remains relatively low when compared to prior art systems. Because other systems change the liquid to a dry gas of relatively low saturation, more work is required by the compressor to return the refrigerant to the liquid state. Such additional work requires more energy and develops more heat. Since less heat is required in my system, the head remains relatively cool and can be touched by hand.

Expansion valves which are, in essence, throttle valves, cause heat due to molecular friction. Standard capillary tubes, wherein the refrigerant changes to vapor within the capillary, produce molecular friction which in turn produces heat. This heat is added to the evaporator and must be dissipated to the condenser. The larger bore and shorter length of the flow controlling device of the present invention does not produce extra heat, thus a smaller motor compressor unit will produce a given refrigerating effect.

Because of the increased efficiency of my system, it is possible to use a motor compressor unit which is substantially less in capacity for the same refrigerating results. Since the cost of weight of motor compressors constitutes the greatest portion of the cost in weight of a refrigeration system, the resultant savings in cost and weight by utilizing my system are at least one half of present refrigeration systems.

When the system is being used to refrigerate spaces at above 32° F. and the compressor stops on the "off cycle," hot liquid Freon continues to flow easily through the relatively large special capillary into the evaporator and effectively defrosts the coil in approximately ten percent of the time required using auxiliary defrosting devices.

Due to the fact that the defrost period is relatively short, the space being refrigerated has less opportunity to rise in temperature and therefore there is less overall load on the motor compressor unit. In addition, the capacity of the motor compressor unit is available for refrigerating a greater portion of the time.

Where the system is used for refrigerating spaces below 32° F., the flow of hot liquid refrigerant during the off cycle tends to minimize the need for defrosting and, also, assist in the defrosting when an auxiliary defrosting device is used. As a result, a system used for maintaining a space below 32° F. requires less frequent defrosting.

Since the refrigerant in the cooling coil (evaporator) is in the form of a highly saturated vapor, which has a greater capacity for absorbing heat, a smaller temperature differential is maintained between the cooling coil and the air in the space being cooled. As a result, very little frost accumulates on the evaporator thus shortening the defrost period and also this small temperature difference and little frost accumulation results in very little moisture removal from the products, such as food, being cooled, thus maintaining a very high relative humidity of 80% to 98%. In this way, food can be kept fresh and moist much longer than in conventional coolers of the present day.

Medium pressure vapor is brought into the compressor suction side. This vapor consists of a mixture of Freon globules of vapor and globules or droplets of oil interspersed and interdissolved. The pressure in the crankcase is less than in the suction line outside the compressor and due to the decreased pressure in the crankcase, the Freon and oil droplets expand and "explode," thus causing an oil mist to lubricate the pistons and cylinder walls and the wrist pins, crankshafts, eccentrics, valve discs, reeds, and all moving internal parts of the compressor as the mixture moves through on its way to the condenser. This assures long trouble-free operation of the compressor.

As shown in FIGURE 1, the return line **23** from heat exchanger **21** passes through an enlarged portion **24** of the outlet **12** so that the returning vapor is in heat exchange relationship with the compressed refrigerant passing out of the compressor **10**.

Due to the fact that with my method the head pressure remains relatively low and therefore the temperature of the discharge vapor keeps far below the critical temperature for the Freons (that is: 205° F. for Freon 22 and 233° F. for Freon 12). The critical temperature for a vapor is that temperature above which it is impossible to return the vapor to a liquid regardless of how great pressure is put on it.

Therefore, with other methods, the vapor reaches far higher than critical temperatures and will not liquefy, and thus causes so called "flash gas" in the liquid line.

My method uses vapor, i.e., wet in the form of fog and/or liquid to carry heat from the area to be cooled and discharges this heat at the condenser. Thus, the heat carrying power is many times greater using liquid and vapor by my method than by dry gas using other methods.

Due to the fact that my method operates with a small temperature differential between evaporator and the surrounding air, the relative humidity of the refrigerated space is held very high—near the dew point—regardless of the temperature. Thus, there is very little dessicating of stored products—a feature very desirable and, by present methods, very costly and difficult to obtain.

The size of the restrictive device used in my system is much larger than has been heretofore thought possible for restrictions such as capillaries in comparable systems utilizing comparable size compressors. Thus, it has been conventional to use with a two horsepower compressor a capillary, for example, having an internal diameter of 0.050" and eleven foot length. In accordance with my invention, for a two horsepower compressor, a restrictive tube is used which is 0.100 inch internal diameter and four feet long. The lengths and bores of the tubes, vary according to their applications but when compared with standard capillary tubes used to achieve similar results, are much shorter in length and/or much larger in bore than those presently used to obtain similar B.t.u. removal, i.e., similar refrigerating effects. The restrictive effect of the restrictive device used in my system is less than one half the restrictive effect of capillaries and other expansion devices heretofore used. Even in household refrigeration systems when capillaries are usually used with an internal diameter of 0.0156 or less and substantial length, in my system, it is desirable to keep the internal diameter of the restrictive device above 0.020 inch and with a much lesser length.

An example of a system and the results achieved is as follows: A combined compressor, condenser and liquid receiver, fan driven, air cooled and of one-half horse-

power was connected to an accumulator having a capacity greater than the evaporator. The restrictive device was in the form of a tube having an internal diameter of 0.050 inch and three to four feet in length. This arrangement was used to cool a walk-in refrigeration enclosure which was fifteen feet by nine feet by ten feet and had a concrete floor and four inch cork board walls and ceiling The evaporator used was fifty-five inches in length of five-eighths inch tubing two tubes wide and three tubes high, having fins one-half inch apart of a size three by three inches slipped on the tubes.

When such a system was operated, it was found that the one-half horsepower compressor was able to hold the temperature of the enclosure at 34° F. with intermittent operation.

When a similar system was used, utilizing a thermostatic expansion valve, compressor, condenser, and receiver, it was found that a two horsepower compressor condenser was needed to maintain the temperature at 35° F. with the compressor operating at all times when used in the same box.

The operation of the system can also be more fully understood by the following example:

A pail of boiling (212° F.) water of fifty pounds capacity was inserted in a 22 cubic foot enclosure utilizing the refrigeration system of the present invention which system had a one-quarter horsepower compressor which maintained the enclosure at 33° F. The ambient temperature outside the enclosure and surrounding the unit was 80° F. Within forty minutes, the temperature of the enclosure reached 50°. In the next forty minutes, the temperature of the enclosure (refrigerator) went down to 33° which was the temperature of the air in the enclosure.

Subsequently, a similar enclosure was connected to a conventional system utilizing one-third horsepower compressor, and a thermostatic expansion valve. With fifty pounds of boiling water in position in the enclosure, it was found that it took ninety minutes for the system to reach 52–53 degrees and one hundred and twenty minutes more to go down to 33° F.

A comparison of the above two systems indicated that the dome of the compressor in the system embodying the invention was warm and could be readily touched by hand whereas, the dome of the compressor of the prior art was so hot that it could not be touched by hand.

It was also found that the system was self-defrosting on the off cycle and no frost accumulated.

An improved efficiency of the system heretofore described is achieved by combining a restrictive tube, accumulator, heat exchanger and an oil separating device in a single enclosure such as shown in FIG. 2. As shown in FIG. 2, a container **25** encloses a cylinder **25***a* which, in turn, encloses the restrictive tube or special capillary **26**. Liquid refrigerant from the liquid receiver passes through pipe **16'** and screen **16***a* to the interior of cylinder **25***a*. The liquid refrigerant then enters restrictive tube **26** through screen **26***a*. The container **25** itself forms the accumulator to which the refrigerant flows from the evaporator **17**, some liquid refrigerant collecting in the bottom of the cylinder **25***a*. The refrigerant from the evaporator comes into heat exchange relationship with the cylinder **25***a* and restrictive tube **26** so that, in effect, a heat exchanger is produced. The refrigerant vapor then passes by successive baffles **27** and into the top of an outlet tube **28** to the compressor. The passage past the baffles **27** separates any oil globules from the refrigerant and the oil collects at the bottom of the container **25** and is removed as drops through a small oil take off **28***a*. The Freon liquid and vapor moves with sufficient velocity through the evaporator and accumulator to sweep oil globules along with it instead of allowing the globules to fall out and collect as a liquid in the lowside as is the case where there is insufficient refrigerant vapor velocity to displace oil in the lowside. Thus, the refrigerant passing

to the outlet **28** contains oil mostly in the misty form which bathes the compressor's internal parts, and effectively lubricates all internal parts—even the valve reeds or discs in the discharge plate, which, with present systems, after a few weeks operation, are invariably found dry and worn. The process is continuous and thus continuous lubrication is provided to all internal moving parts.

In the combined unit shown in FIG. 4, container **25** encloses and forms the accumulator in which the restrictive tube or special capillary **26** is positioned. The refrigerant from the liquid receiver passes from tube **16'** through the restrictive device **26** and thereafter through tube **18'** to the evaporator. The refrigerant from the evaporator passes into the container **25** through tube **19'** and is subjected to the baffles **27** as in the previous combined unit shown in FIGS. 2 and 3. By this arrangement, the enclosure **25** serves the dual function of an accumulator and a heat exchanger with the restrictive tube **26**.

Although I do not wish to be bound by the theory involved, in my opinion, the beneficial advantages obtained by the present invention are due to an operation of the system at lower pressure so that less heat is lost by radiation and a higher efficiency is achieved, due also to the fact that vapor and/or liquid transports many times the B.t.u.'s that a similar volume of dry "gas" will transport. My method uses vapor and/or liquid refrigerant to remove the B.t.u.'s from the space to be cooled, whereas the prior art systems use relatively dry gas to remove the B.t.u.'s and thus their carrying capacity is much less.

The following is the thermodynamic calculations showing the proper B.t.u. availability in a refrigeration cycle.

T—temperature
P—pressure

With T and P constant, we have $W=-p\delta v$ and the condition of equilibrium becomes

$$\delta\left(\phi-\frac{U+pV}{T}\right)=0$$

or $\delta\psi=0$, since

$$\left(\phi-\frac{U+pV}{T}\right)=\psi,\phi=\text{entropy}$$

If energy U and volume V are independent variable, then the entropy is the characteristic function and we find directly from the equation:

$$d\phi=\frac{dU+pdV}{T}=\frac{\partial\phi}{\partial U}dU+\frac{\partial\phi}{\partial V}dV$$

that

$$\left(\frac{\partial\phi}{\partial U}\right)_v=\frac{1}{T}$$

and

$$\left(\frac{\partial\phi}{\partial V}\right)U=\frac{P}{T}$$

If ϕ is a known function of U and V, then:

$$T-\frac{1}{\frac{\partial\phi}{\partial U}}P-\frac{\frac{\partial\phi}{\partial V}}{\frac{\partial\phi}{\partial U}}$$

and hence ϕ and U can be found as functions of T and P.

Likewise, if V and T are chosen as independent variable, then the free energy F is the characteristic function and

$$dF=dU-Td\phi-\phi dT$$

or

$$dF=-pdV-\phi dT$$

Hence

$$\left(\frac{\partial F}{\partial V}\right)_T=-p$$

and

$$\left(\frac{\partial F}{\partial T}\right)_V=-\phi$$

If F is a known function f of V and T, then we have the following expressions:

$$p=-\frac{\partial F}{\partial V}$$

$$\phi=-\frac{\partial F}{\partial T}$$

$$U=F+T\phi=F-T\frac{\partial F}{\partial T}$$

If p and T are chosen as independent variables, then the characteristic function is

$$\psi=\phi-\frac{U+pV}{T}$$

It follows that:

$$d\psi=d\phi-\frac{dU+pdV+Vdp}{T}+\frac{U+pV}{T^2}dT$$

or

$$d\psi=-\frac{V}{T}dp+\frac{U+pV}{T^2}dT$$

Hence

$$\left(\frac{\partial\psi}{\partial p}\right)_T=-\frac{V}{T}$$

and

$$\left(\frac{\partial\psi}{\partial T}\right)_P=\frac{U+pV}{T^2}$$

If ψ is a known function of p and T, then we have

$$V=-T\frac{\partial\psi}{\partial P}$$

$$U=T\left(T\frac{\partial\psi}{\partial T}+p\frac{\partial\psi}{\partial p}\right)$$

and

$$\phi=\psi+T\frac{\partial\psi}{\partial T}$$

From these all thermodynamic properties of a system are determined uniquely.

Entropy for a perfect gas is zero and if we consider that for all practical purposes the "Freons" are near perfect gases, then we may assume that their entropies are zero.

Then, taking Freon 12 in a 1 H.P. compressor as manufactured by Tecumseh Products Co., Tecumseh, Michigan, we find the enthalpy for Freon 12.

R.P.M.=1750
Displacement=5.78 cubic inch per revolution
Time=60 minutes or 1 hour

At −40° F. vapor density of F 12 is .2557 lbs./cu. ft.
Latent Heat at −40° F.=73.50 B.t.u. per lb.
Therefore: enthalpy per H.P. per hour for

$$\text{F 12}=\frac{5.78\times1750\times60\times.2557\times73.50}{1728}=$$

6600.72 B.t.u./hr. per H.P.

Tecumseh rating for 1 H.P. at −40° F. is 2170 B.t.u./hr.

Therefore, my system gives 6600.72/2170.00=3+ or three times the refrigerating effect obtained from present systems. One of the main reasons is the fact that cold liquid and vapor is present in the evaporator using my device. Thus, the temperature difference between the refrigerant in the evaporator and the air in the room being cooled is very small perhaps less than 5° F., i.e., it is not necessary to have the metal of the evaporator much lower in temperature than the temperature of the air or product in the refrigerated room. Also, due to "mist oiling" of the valves and pistons and due to low head pressure, there is not the usual heat from friction or the usual heat of compression.

Similarly, for Freon 22 as per Tecumseh specifications: Displacement=3.59 cubic inches. At −40° F., vapor density of F 22 is .3050 lb./cu. Latent heat at −40° F.=100.46 B.t.u. per lb. R.P.M.=1750.

Total refrigerating effect=

$$\frac{3.59\times1750\times60\times0.3050\times100.46}{1728}=$$

6685.7 B.t.u./hr. per H.P.

Tecumseh rating for 1 H.P. is 1920 B.t.u./hr. per H.P.

As heretofore discussed, in systems embodying my invention, the head pressure is substantially lower and the back pressure is substantially higher than in conventional refrigerating systems. As a general rule, if systems embodying my invention are compared with comparable systems utilizing the same size compressors, the head pressure in my system is less than half the head pressure in conventional systems and the back pressure in my system is more than double the back pressure in conventional systems. Moreover, if the same size compressors are used, my system produces a much greater refrigerating capacity as heretofore described.

As an example, a system embodying my invention utilizing a two horsepower compressor for refrigerating a space above 32° F. may have a head pressure of approximately 90 pounds per square inch and a back pressure of approximately 15 pounds per square inch. This may be contrasted to a conventional refrigerating system utilizing a two horsepower compressor which under the same conditions may have a head pressure of approximately 180 pounds plus per square inch and a back pressure of approximately 5 pounds plus per square inch.

It can thus be seen that there has been provided in accordance with my invention a refrigeration system which has a greater efficiency than has heretofore been possible with conventional systems. In accordance with my invention a compressor is used having a capacity much less than has been heretofore thought necessary for refrigerating spaces of the same size. The compressor is operated at a substantially lesser head pressure and a much higher suction pressure. The system is self-defrosting during the off cycle when refrigerating a space to temperatures above 32° F. and defrosts in a much shorter time than conventional systems utilizing some defrosting arrangement when refrigerating a space to a temperature below 32° F. Furthermore, the system provides a mist type oiling for all internal parts of the compressor.

The refrigerating system embodying my invention is thus more efficient and less costly than prior art systems.

In a standard system of the prior art, it has been customary where, for example, a 20° F. air temperature is desired adjacent an evaporator coil, to design the system so that the temperature of the evaporator coil is about 0° F. In a system embodying my invention, in order to have a 20° F. air temperature adjacent the evaporator coil, it is recommended that the temperature of the evaporator coil be 15° F. Similarly, in walk-in refrigerator boxes maintained at a temperature of 35° F. it is conventional, utilizing a standard system, to use an evaporator coil which is at a temperature of 15° F. For a comparable temperature of 35° F. in the walk-in box, in my system, the evaporator coil is recommended to be at 32° F.

Summarizing the most important features of my invention:

(1) The combination of a relatively low pressure differential between the high side and the low side.

(2) A flow control device which feeds the liquid refrigerant to the evaporator as a liquid refrigerant.

(3) An accumulator that has a volumetric capacity equal to the condenser or evaporator.

(4) In the operation of the system, because of the combination between the large flow control and the accumulator, one has what might be called a continuous flooded evaporator, in that substantially all of the internal surfaces of the coil are backed with liquid, substantially uniformly throughout its length, with refrigerant exploding into gas bubbles throughout its length.

(5) The accumulator will be partially filled with refrigerant liquid.

(6) During off cycles the relatively large control unit will feed hot refrigerant liquid to the evaporator.

(7) Because of the combination of the relatively large flow control unit, wet surface evaporator, and accumulator, my system requires initial charging of approximately, preferably, three times the amount of refrigerant used in a conventional system of comparable refrigerating effect.

(8) Furthermore, the system permits a misting type oiling for all internal parts of the compressor.

(9) Because my system puts more liquid through the flow control device than a standard system, the flow rate of liquid and gas throughout my entire system is much faster than in standard systems using the same size compressor.

(10) Although some oil droplets may collect at the bottom parts of the evaporator and the bottom of the accumulator when the system is shut off, when in operation, the oil droplets will be swept clear and into the compressor.

What I claim is:

1. The method of improving the thermodynamic efficiency of refrigeration systems of the restrictive device type and utilizing a continuous type evaporator and accumulator which comprises

charging the system with at least twice the normal amount of refrigerant used with such a system of the same refrigerating capacity,

continuously circulating said abnormal volume through the system,

decreasing the restrictive effect of the restrictive device to at least one half that of the standard type for an evaporator of the same capacity such that the refrigerant passes in a continuous and uninterrupted liquid mass through the restrictive device and thereafter is converted by expansion to a highly saturated, relatively wet vapor and remains as a relatively wet vapor in passing through the evaporator and accumulator to the compressor whereby substantially all the internal surfaces of the evaporator are backed with liquid refrigerant,

moving said refrigerant through the system in a condition such that a rapid transfer of heat occurs through the evaporator walls and the surface temperature of the evaporator is maintained relatively near the temperature of the area being refrigerated so that a relatively high humidity is provided in the area being refrigerated,

insuring a continuous supply of liquid refrigerant on the high side by the use of a receiver positioned below the condenser,

and continuously accumulating and trapping liquid flowing from the evaporator toward the compressor.

2. In a refrigeration system comprising a compressor, a condenser, a liquid receiver, an evaporator and an accumulator connected between the evaporator and the compressor, the improvement wherein a restrictive device is provided between the liquid receiver and the evaporator,

said refrigerating system being charged with at least twice the refrigerant used with such a system of the same refrigerating capacity,

the size of said restrictive device being such that the abnormal volume of refrigerant passes in a continuous and uninterrupted liquid mass through the restrictive device and thereafter is converted by expansion to a highly saturated, relatively wet vapor and remains as a relatively wet vapor in passing through the evaporator and into the accumulator,

said restrictive device comprising a tube which is shorter in length or greater in diameter than a conventional capillary tube for the refrigeration components of comparable size,

said accumulator having a volume not less than the volume of the evaporator,

said liquid receiver having a capacity sufficient to maintain the condenser substantially free of liquid at all times,

said restrictive device being in heat exchange relationship with the interior of said accumulator.

3. The method set forth in claim 1 including the step of periodically interrupting the operation of the compressor such that hot liquid refrigerant passes through the restrictive device substantially uninterrupted and facilitates the defrosting of the evaporator.

4. In a refrigeration system comprising a compressor, a condenser, a liquid receiver, an evaporator and an accumulator connected between the evaporator and the compressor, the improvement wherein a restrictive device is provided between the liquid receiver and the evaporator,

said refrigerating system being charged with at least twice the refrigerant used with such a system of the same refrigerating capacity,

the size of said restrictive device being such that the abnormal volume of refrigerant passes in a continuous and uninterrupted liquid mass through the restrictive device and thereafter is converted by expansion to a highly saturated, relatively wet vapor and remains as a relatively wet vapor in passing through the evaporator and into the accumulator,

said restrictive device comprising a tube which is shorter in length or greater in diameter than a conventional capillary tube for the refrigeration components of comparable size,

said accumulator having a volume not less than the volume of the evaporator,

said liquid receiver having a capacity sufficient to maintain the condenser substantially free of liquid at all times.

5. The combination set forth in claim 4 wherein said system includes connecting means between the accumulator and the inlet of the compressor and connecting means between the outlet of the compressor and the restrictive device, said connecting means being in heat exchange relationship to one another.

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