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REFRIGERATING APPARATUS

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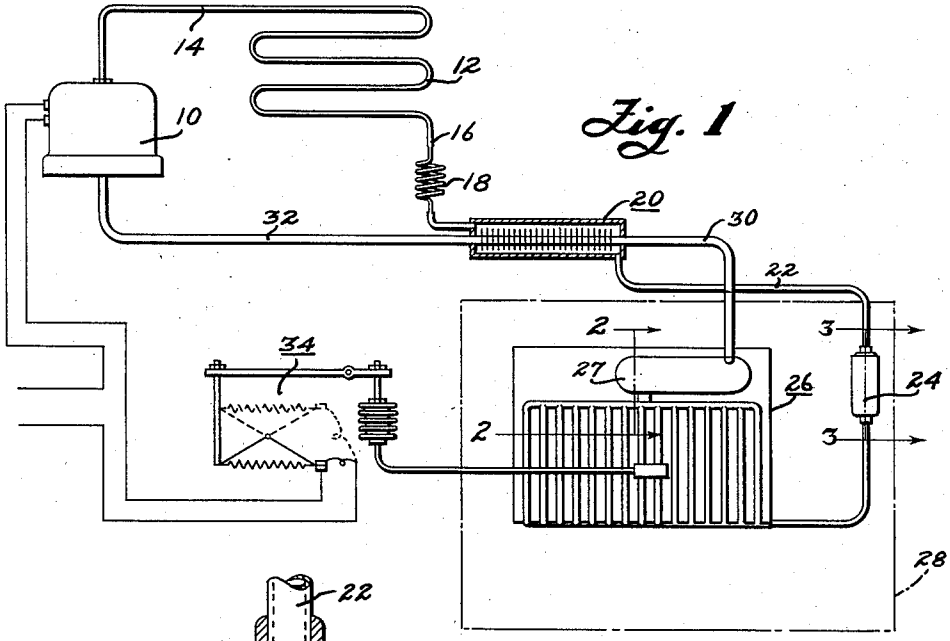


Fig. 1

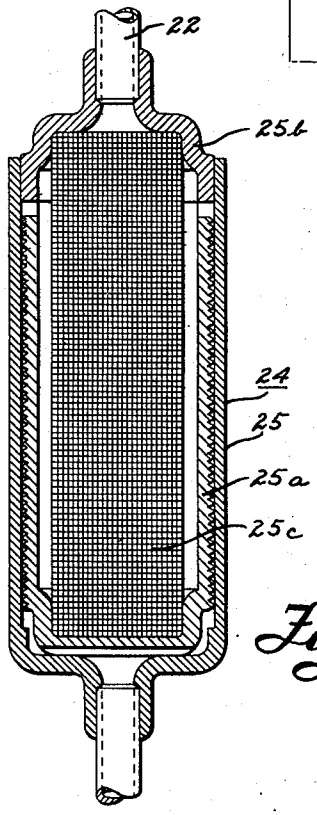


Fig. 3

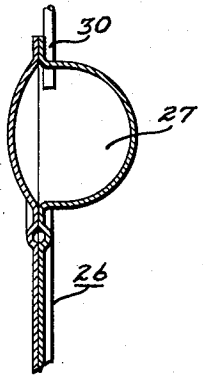


Fig. 2

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REFRIGERATING APPARATUS

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2 Claims. (Cl. 62—115)

This invention relates to refrigerating apparatus and more particularly to controlling means for the refrigerant circuit of a compression refrigerating machine. In refrigerating apparatus of the compression type designed for low cost, it is essential, if satisfactory operation is to be secured, that the refrigerating cycle take place with maximum efficiency at all times since with a low powered motor, which cost limitations impose, the unit must be operated as near as possible to its theoretical maximum efficiency if satisfactory box temperature and ice freezing characteristics are to be maintained.

It has been possible heretofore to construct low cost refrigerating apparatus which will operate with satisfactory efficiency at full load by proper design of the system and proper correlation of the relative capacities of its various elements. Many such low cost systems employ a fixed restrictor to regulate expansion of the refrigerant in order to take advantage of the low cost and freedom from service difficulties inherent in that type of expansion device. However, the natural characteristics of a fixed restrictor are such that the changes in rate of flow therethrough produced with changes in pressure differential across the same do not produce the optimum flow rates for all possible load conditions. It has been necessary in designing a system of this character, therefore, to choose some one load condition, usually the maximum, under which the system is designed to operate at maximum efficiency and to accept operation at considerably less efficiency under all other load conditions.

If a refrigerating apparatus of the character described is designed to operate at maximum efficiency under maximum load conditions, such, for example, as a high room temperature, together with a large freezing load of water to be cooled and frozen, the considerably reduced efficiency under less than full load conditions causes unnecessarily large current consumption at times when the apparatus is not operating under maximum load.

Heretofore, it has been possible to reduce the high current consumption at partial loads only by sacrificing efficiency at full load, and it is an object of the present invention to provide a mechanical refrigerating apparatus which is designed to operate at maximum efficiency under full load, and in which the loss of efficiency and consequent current consumption are materially reduced when operating under partial load conditions.

In order to meet these requirements, the present invention provides, in addition to the usual fixed restrictor or other control device which regulates the expansion of liquid refrigerant into the evaporator, an additional restricting means located at the outlet of the condenser for the purpose of providing additional restriction during operation at full load and insuring that the refrigerant delivered from the condenser will be substantially all in liquid form. This is particularly advantageous in systems utilizing a heat interchanger for improved efficiency at full load, but in which the interchanger introduces a substantial loss of energy when the refrigerant entering the same has not been completely liquefied as happens under full load operation.

It is a further object, therefore, to provide means at the outlet of the condenser in a refrigerating apparatus for preventing the passage of gaseous refrigerant, particularly in a refrigerating apparatus employing a heat interchanger for equalizing temperatures between the liquid refrigerant entering the evaporator and the gaseous refrigerant leaving the same.

Further objects and advantages of the present invention will be apparent from the following description, reference being had to the accompanying drawing, wherein a preferred form of the present invention is clearly shown.

In the drawing:

Fig. 1 is a diagrammatic view of a refrigerating apparatus embodying the preferred form of the present invention;

Fig. 2 is a fragmentary cross section of line 2—2 of Fig. 1; and

Fig. 3 is a fragmentary cross section of line 3—3 of Fig. 1.

Referring now to the apparatus shown in Fig. 1, there is illustrated diagrammatically, a compression refrigerating apparatus which includes a motor-compressor unit 10, preferably of the hermetically sealed type and which is adapted to deliver compressed refrigerant to a condenser 12 by means of a conduit 14. Located at the outlet 16 to the condenser 12, there is a liquefying restrictor 18 which preferably takes the form of a helical coil of tubing of small diameter and considerable length. It will be understood that the liquefying restrictor may take other forms, for example, the liquid line itself may be made sufficiently small in diameter and of sufficient length to act as a restrictor. In either event the restrictor 18 is in the present invention preferably of the type which is immovable and continuously open as distinguished from weighted, float or pressure operated valves movable into open and

closed position. The liquefying restrictor 18 feeds liquid refrigerant to a heat interchanger 20, whence cooled liquid refrigerant is delivered by conduit 22 to an expanding restrictor 24, which may be of any well known construction and which is shown in the drawing as a helical groove of small diameter and of considerable length. This may be formed of an outer cylinder 25 and an inner cylinder 25A having a thread like groove cut on its outer surface. The two cylinders are assembled together sufficiently tight to prevent leakage across the contacting surfaces and to cause refrigerant to follow the helical path provided by the thread like groove. A cap 25B closes the open end of the outer cylinder and retains a filter screen 25C in position to filter the refrigerant before entering the groove.

The evaporator 26 receives from the restrictor 24 refrigerant which expands in the evaporator to withdraw heat from the refrigerator cabinet or other device to be cooled, indicated by the dotted rectangle 28. The evaporator is preferably of the type embodying a plurality of refrigerant conduits connected in parallel and is illustrated as formed from embossed sheet metal in a manner well known in the art. Adjacent the outlet of the evaporator there is provided a reservoir 27 for liquid refrigerant to permit the evaporator to hold varying quantities of liquid refrigerant without materially varying the area of surface contact between the liquid and the evaporator. A suction conduit 30 delivers expanded refrigerant to the interchanger 20, whence it is delivered by a conduit 32 to the compressor 10. During conditions of partial refrigerating load, the motor-compressor unit 10 is operated intermittently under the control of a suitable thermostatic switch, generally designated as 34, which controls the circuit of the motor in the unit 10. It will be understood, of course, that the parts illustrated in Fig. 1 are represented only diagrammatically for the sake of simplicity, and that they may be of any well known structural form and that the usual accessories to a complete refrigeration system may be provided. For example, a liquid refrigerant receiver may be provided in conjunction with the condenser 12, and the evaporator 26 may be disposed in any position in the cabinet, for example, at the top thereof. Likewise, the condenser may be at a lower level than the evaporator if desired.

The design of the refrigerating apparatus and the correlation of the various parts thereof with each other and with the refrigerator cabinet to be refrigerated may follow generally that described in the copending application of Andrew A. Kucher, Serial No. 668,771, although it is to be understood that many of the advantages of the present invention may be derived by its incorporation in refrigerating systems of other design.

Considering the operation of the apparatus described, first under conditions of maximum load namely: at the highest room temperature normally encountered and with a maximum freezing load at the evaporator, it will be noted that due to the high room temperature the head pressure in the condenser 12 is at a high value. Also, since the freezing load and the load at the evaporator due to heat leakage through the cabinet are both large, the vaporization of the liquid refrigerant in the evaporator will take place at a most rapid rate, causing a comparatively high back pressure in the evaporator and suction conduits 30 and 32. Under these conditions, the compressor 10 is compressing a maximum amount of refrigerant by weight due to the high back pressure at which refrigerant is taken into the compressor. The restricting effect of the restrictors 18 and 24 are such that under the conditions described, they pass refrigerant at a much greater rate than it is compressed by the compressor 10.

Due to the lag in response of certain portions of the system to the relatively sudden application of a maximum load at the evaporator, it will be seen that a certain amount of refrigerant is retained in the condenser in liquid form before the head pressure builds up sufficiently to cause the restrictor 24 to pass refrigerant at the same rate that it is being compressed. Under these conditions, the refrigerant leaving the condenser 12 is entirely in liquid form. For some time after the freezing load is first imposed on the evaporator 26, the system operates in this manner, producing refrigeration at substantially the maximum rate possible from a compressor of the size incorporated in the system. As soon as the freezing load begins to taper off, however, the system operates in a somewhat different manner.

Due to a decrease in the rate of vaporization at the evaporator 26, the back pressure in the evaporator 26 and the suction lines 30 and 32 will be lowered, thus increasing momentarily the pressure differential across the restrictor 24 and causing liquid refrigerant to be delivered to the evaporator at a faster rate than it is evaporated therefrom. Due to the high head pressure which is maintained in the condenser as a result of the high room temperature, the accumulated liquid refrigerant in the condenser becomes gradually exhausted since the combined restricting effect through restrictors 18 and 24 is such that under the large pressure difference across the restrictors, refrigerant passes at a greater rate than it is being compressed. In other words, soon after the freezing load has been applied, liquid refrigerant tends to accumulate in the evaporator 26. The reservoir 27 acts to accommodate the additional liquid refrigerant without material change in area of surface contact between the liquid refrigerant and the evaporator and gives room for violent ebullition to occur without causing liquid particles to be drawn into conduit 30. The accumulation of liquid refrigerant in the evaporator results eventually in exhausting all the liquid refrigerant accumulated in the condenser 12 or in the receiver which may be associated therewith. Under these conditions of operation, the restrictor 18 comes into play.

It is known that any restrictor of the so-called capillary passage type which comprises a passage of small cross section, but great length and which derives its restricting action principally from fluid friction, offers many times as much resistance to a given quantity of refrigerant in gaseous form as it offers to the same refrigerant in liquid form. For example, with some refrigerants in common use, a given restrictor will pass roughly fifty times as much liquid refrigerant by weight per unit of time as will pass through the restrictor in gaseous form. Therefore, as soon as the gaseous refrigerant begins to enter the restrictor 18 a very great resistance to its passage is offered, thereby resulting in materially slowing up the delivery of refrigerant from the condenser. This increases the condensing capacity of condenser 12 not only by building up the back pressure therein, but by decreasing the rate of flow

therethrough, and consequently increasing the length of time during which a given quantity of refrigerant remains in contact with its condensing surfaces. It will also be seen that with a liquefying restrictor of the construction illustrated, namely: a helical coil of tubing, that the restrictor itself being located outside the cabinet and being cooled by the room air or other medium utilized to cool the condenser 12, acts to condense any gaseous refrigerant that enters the same before it can leave the opposite end of the restrictor 18. Thus, the liquefying restrictor under the conditions now being considered insures that no gaseous refrigerant will enter the interchanger 20. In so doing, a serious loss of efficiency is avoided by preventing the condensation of refrigerant in the interchanger 20. If liquefaction of refrigerant before entering the interchanger were not insured, there would result a condition in which cold gaseous refrigerant entering the interchanger 20 from the evaporator would absorb latent heat from the gaseous component of the refrigerant delivered to the interchanger 20 from the condenser 12, thus absorbing some of the work done by the compressor within the system itself and resulting in a loss of efficiency. This loss in some types of refrigerating apparatus amounts to as much as 15 to 20 percent of the total load capacity of the system. It thus becomes a major factor in reducing the refrigerating output delivered from a given quantity of electrical energy input to the unit under partial load conditions.

The liquefying restrictor is also useful under other conditions of less than full load operation, for example, during conditions of high heat leakage load as when the room temperature is comparatively high. Thus, at high room temperature, it will be seen that the head pressure in the condenser 12 is high, since refrigerant condenses at a higher pressure in a higher room temperature. The design of the system being such that the back pressure does not increase as fast as the head pressure with higher room temperatures, there results a greatly increased pressure differential across the restrictors 18 and 24. Since the restrictors 18 and 24, in order to insure efficient operation at low load, have been calibrated to pass more than the quantity of refrigerant required under the load conditions now being considered, there will be a tendency for liquid refrigerant to accumulate in the evaporator, thus starving the condenser 12 and tending to deliver gaseous refrigerant therefrom. As soon as the condenser 12 is emptied of liquid refrigerant, the liquefying restrictor acts to greatly reduce the flow of refrigerant upon the first entry of any gaseous refrigerant and thus back up refrigerant in the condenser for liquefation as described heretofore.

It will be understood, of course, that at times when gaseous refrigerant enters the liquefying restrictor 18, there is not a sudden change from all liquid refrigerant to all gaseous refrigerant, but the change will take place gradually, starting with small bubbles of gas being delivered from the condenser 12. This action is entirely gradual and any gas bubbles which enter the restrictor 18 tend to reduce the rate of flow therethrough, thus backing up refrigerant in the condenser and tending to increase the condensing action in the condenser. It will be seen that the functioning of the device under these conditions is inherently self-balancing and under any given set of load conditions, a balance will be set up automatically such that the rate of flow through the restrictor is held down to a value at which practically all the condensing takes place in the condenser 12. The condensing action in the restrictor 18 takes place only to the extent necessary to condense before leaving the restrictor 18 that quantity of gaseous refrigerant which is required to thus reduce the rate of flow through the restrictor.

It should be noted that many of the benefits provided by the restrictor 18 do not depend on condensation of the gaseous component of the refrigerant being completed before leaving the restrictor 18. The proportion of gaseous refrigerant necessary to produce the required amount of restriction at restrictor 18 is very small compared to the proportion of gaseous refrigerant which would be fed to the interchanger 20 if the restrictor 18 were omitted, and consequently, even if part or all of the gaseous component were not condensed before leaving restrictor 18, the latent heat loss in the interchanger would be but a small fraction of that which would be incurred without the liquefying restrictor 18. In other words, many advantages of the present invention may be obtained from a liquefying restrictor which is so formed or situated that little or no condensation takes place at the restrictor since the proportion of gaseous refrigerant required to make it effective on the system is very small. Obviously, therefore, any restrictor having the ability to permit comparatively unrestricted flow as long as the refrigerant enters in a completely liquefied state to greatly restrict the flow upon entrance of refrigerant having a portion thereof in the gaseous state will provide a marked improvement in efficiency and is within the purview of the present invention.

It will thus be seen that the present invention provides a refrigerating system capable of operating at maximum efficiency and consequent low current consumption not only under conditions of maximum load on the system, but which also operates intermittently during periods of less than full load requirements and which during the intermittent periods of running also operates at substantially maximum efficiency. This result is achieved by provision of means which permits the total restriction between the condenser and the evaporator to be reduced to a value which permits efficient operation at the low load conditions when the pressure differential across the restrictor is low, without incurring the losses which would otherwise result at higher loads from the delivery of refrigerant from the condenser in gaseous form due to the exceedingly fast rate at which a restrictor of that value would pass refrigerant under the high pressure differential existing under heavy load.

From the foregoing, it will be seen that the provision of the liquefying restrictor permits certain refinements in the design of the system which could not be achieved without it. For example, with a single fixed restrictor located between the interchanger and the evaporator, it is impossible to calibrate the restrictor to give the proper flow rates for more than one load condition. If the restrictor is designed to pass refrigerant at the proper rate for full load when the pressure difference across it is a maximum, then at low loads the much lower pressure difference results in too small a rate of flow and greatly reduced efficiency for that reason. On the other hand, if the restrictor is calibrated to pass enough refrigerant under the low load conditions, then at high load it passes too much refrigerant resulting in incomplete condensation

at the condenser and reduced efficiency from the latent heat loss at the interchanger. The liquefying restrictor, on the other hand, permits the total restriction to be reduced to the proper value for low load conditions, when complete liquefaction in the condenser is not a problem anyway, and acts under higher load conditions to automatically increase the restriction, in effect, by its action in holding back or slowing up the passage of refrigerant upon the entry of gaseous refrigerant from the condenser. In other words, the liquefying restrictor permits the use of a small enough total restriction to cause operation at maximum efficiency at low load and at all higher loads holds the rate of flow down to the proper value for any given load by its greatly increased resistance to gaseous refrigerant.

While the form of embodiment of the invention as herein disclosed, constitutes a preferred form, it is to be understood that other forms might be adopted, all coming within the scope of the claims which follow.

What is claimed is as follows:

1. In a mechanical refrigerating apparatus, the combination of a refrigerant liquefying means including a refrigerant condenser, a refrigerant evaporator having an inlet and an outlet, means for interchanging heat between the refrigerant entering the evaporator and that leaving the evaporator, a first continuously open restrictor between the condenser and the interchanging means, a second continuously open restrictor between the interchanging means and the evaporator, and a liquid and gaseous refrigerant reservoir at the top of the evaporator adjacent the outlet thereof.

2. In a mechanical refrigerating apparatus, the combination of a refrigerant liquefying means including a refrigerant condenser, a refrigerant evaporator having an inlet and an outlet, a continuously open restrictor at the inlet of the evaporator for expanding liquid refrigerant into the evaporator, means for interchanging heat between the refrigerant entering the evaporator and that leaving the evaporator, a continuously open restrictor for imposing a comparatively great restriction to the passage of gaseous refrigerant from the condenser to the interchanger, and means in the evaporator for receiving variable quantities of liquid refrigerant in the evaporator without materially varying the area of surface contact between the liquid refrigerant and the evaporator.

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