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EVAPORATOR FEED CONTROL MEANS IN
REFRIGERATING APPARATUS

2,871,679

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2 Sheets-Sheet 1

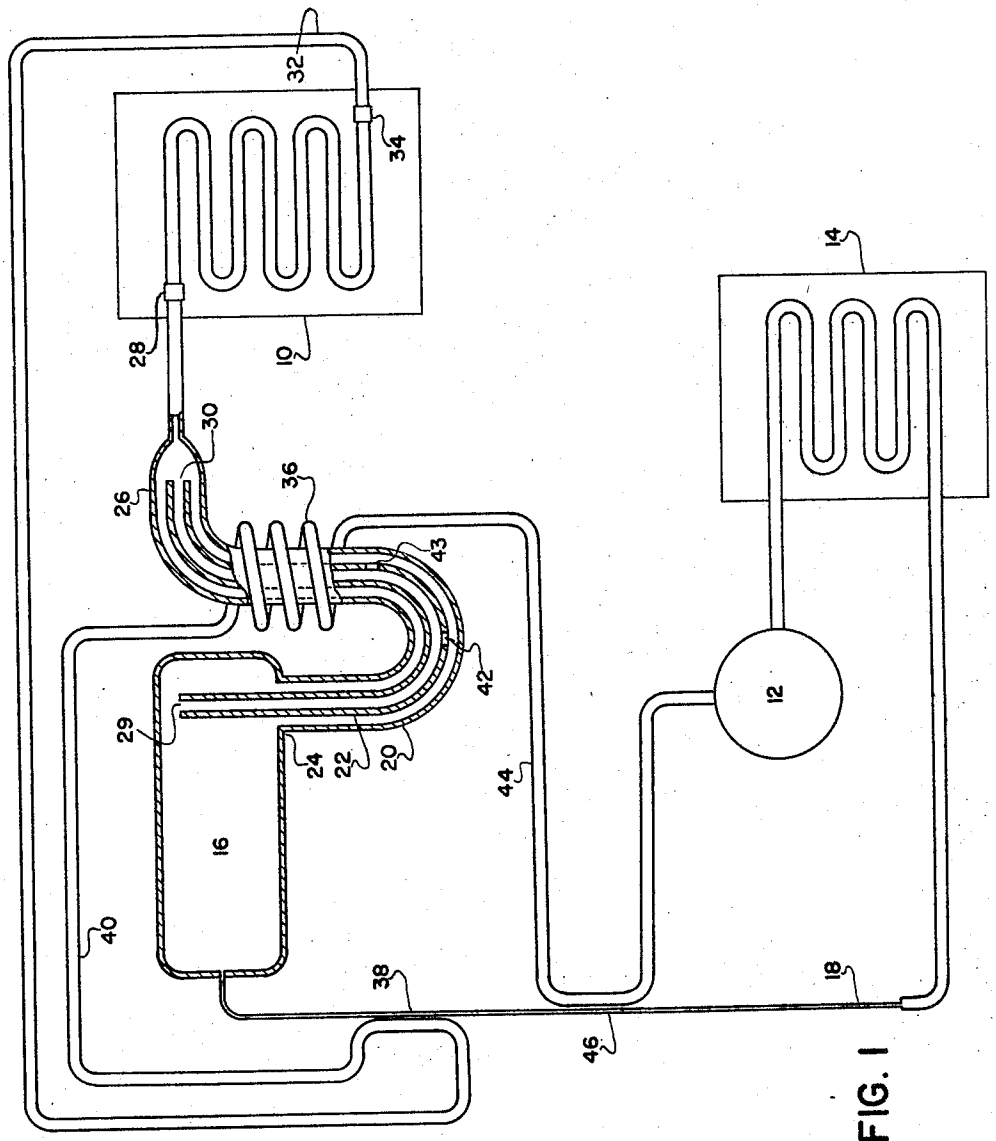


FIG. 1

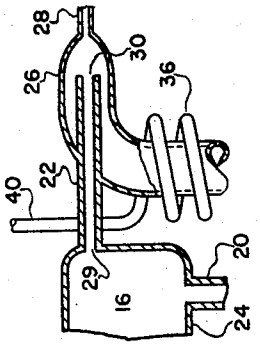


FIG. 3

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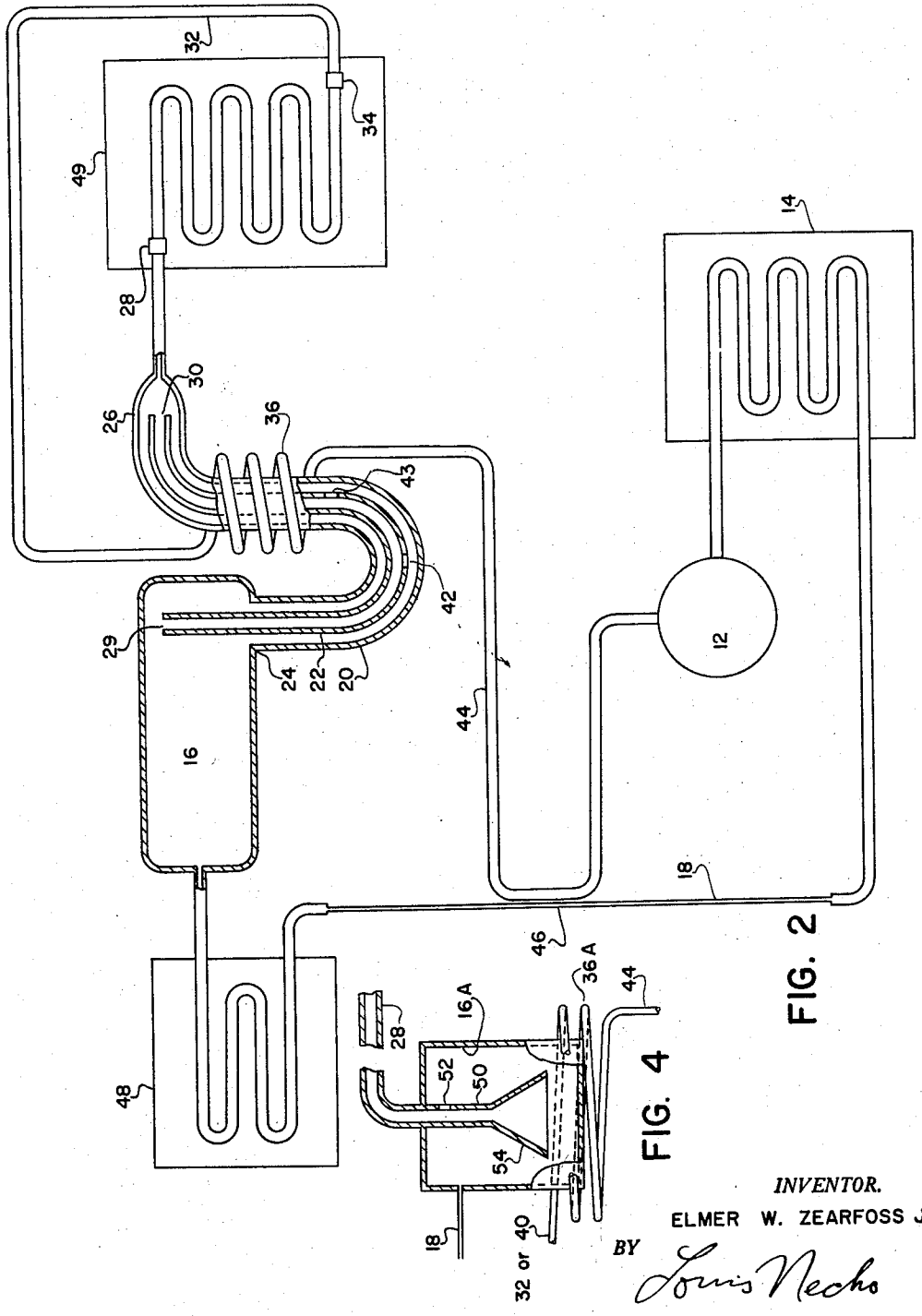


FIG. 2

FIG. 4

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EVAPORATOR FEED CONTROL MEANS IN REFRIGERATING APPARATUS

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Application January 19, 1955, Serial No. 482,820

14 Claims. (Cl. 62—504)

My invention relates to a refrigerating apparatus of the type in which a capillary tube is employed as the pressure reducing means between the high and low sides of the apparatus.

One object of the invention is to produce an improved refrigerating apparatus of the type set forth.

Under theoretically ideal, controlled operating conditions and given a constant refrigeration load, it is possible to design a system in which all components are balanced and in which the flow of refrigerant through the evaporator is at optimum so that a high degree of efficiency is attained.

But, in practice, the refrigeration load and the temperature-pressure conditions which affect the balance of the system, vary and thus create problems in design and in control.

To solve this problem, it has heretofore been proposed to provide an evaporator circuit having a refrigerant storage capacity such as to suffice for maximum requirements. This expedient is not altogether desirable in a flooded type system because the practicable configurations of the evaporator are limited and because a relatively high refrigerant charge is required. In the dry, or accumulator type of system, the expedient referred to is not desirable because it will result in undesirable peak load and onset cycle performance which reflect in proper refrigerant distribution and flow control. Furthermore, the inclusion of an accumulator in the system creates oil logging problems.

It is therefore a further object of the invention to produce a system in which all of the foregoing problems are overcome.

More specifically, the object of the invention is to produce a wholly self-regulating refrigerating apparatus in which the amount of refrigerant which reaches the evaporator is automatically regulated by the demand imposed on the evaporator to the end that the evaporator will, at all times, receive a supply of refrigerant which is, in effect, a function of the temperature of the spent refrigerant gas leaving the evaporator.

A still further object is to produce an improved refrigerating apparatus in which the regulation of the supply refrigerant reaching the evaporator is effected without any moving parts and in a manner which does not appreciably increase the cost or weight of the apparatus and which involves no maintainance cost whatever.

In most types of apparatus involving the use of a capillary tube it is necessary that the refrigerant charge introduced into the sealed circuit be accurately or, at least, very closely estimated. Otherwise, for well known reasons, satisfactory and efficient operation may become impossible.

It is therefore a still further object of the invention to produce an improved refrigerating apparatus in which the volume of the refrigerant charge initially introduced into the system is not so critical and need not be accurately predetermined.

In all refrigerating machines, over-rides or extremes

of temperature, be they on the cold or on the warm side are not desirable. In other words, uniform operation, within predetermined limits is desirable and, to that end, the cycling of the apparatus should be such as uniformly to maintain the desired temperature.

It is therefore a still further object of the invention to produce an improved refrigerating apparatus which has a cycling performance without "hunting" and undue override.

These and other objects are attained by my invention as set forth in the following specification and as shown in the accompanying drawings in which:

Fig. 1 is a diagrammatic representation of a refrigerating apparatus embodying my invention.

Fig. 2 is a similar view showing a second embodiment of the invention.

Figs. 3 and 4 are reduced, fragmentary and diagrammatic views showing slight modifications which can be made in the apparatus shown in Figs. 1 and 2.

In all embodiments the apparatus includes an evaporator **10**, a compressor **12**, a condenser **14**, and an accumulator **16** which is connected through the condenser, to the high, or discharge, side of the compressor by means of a capillary tube **18**. The structure and operation of the evaporator, compressor and condenser are conventional and are therefore not shown nor described in detail. For the purpose of this disclosure, it is thought sufficient to say that the spent refrigerant is withdrawn from the evaporator by the suction side of the compressor; that the compressed gas is discharged into the condenser where it gives up heat and changes to the liquid state, and that the liquid refrigerant, or a mixture of liquid and gaseous refrigerant, is delivered through capillary tube **18**, with or without the inter-position of accumulator **16**, to the evaporator where, due to reduced pressure, the liquid refrigerant evaporates and, in so doing, absorbs heat from the evaporator.

The improved flow control means of the embodiment of my invention as shown in Fig. 1 includes an outer tube **20**, having a relatively large internal diameter, and an inner tube **22** having a relatively small internal diameter and disposed within the outer tube. It will be noted that the intake end **24** of outer tube **20** leads from near the bottom of accumulator **16** and that its discharge end **26** is connected to the intake end **28** of the evaporator. It will also be noted that the intake end **29** of inner tube **22** leads from near the top of the accumulator and that its discharge end **30** leads, but is not physically connected to, the intake end of the evaporator. The inner tube **22** thus provides a passage for the flash gas from the accumulator to the evaporator an dtube **22** serves to receive refrigerant liquid. The combined volumetric capacity of outer tube **20** and of the accumulator is in excess of that needed to accommodate all the liquid refrigerant that could collect in these parts at any one time, so that neither the intake end **29**, nor the discharge end **30**, of inner tube **22**, are ever submerged in liquid refrigerant.

The parts thus far described operate as follows:

As the liquid gas moves through the capillary tube, its pressure is gradually decreased and increasing quantities of flash gas are produced so that a mixture of liquid and gas refrigerant is ultimately delivered to the accumulator. The gaseous component, which rises to the top of the accumulator, flows to the evaporator through inner tube **22** and since this tube is virtually non-restrictive, there will be no appreciable pressure drop across its ends and the liquid refrigerant in outer tube **20**, except as hereinafter set forth, can be said to be in the hydro-static state. Under these conditions the evaporator can be said to be inactive in that its temperature will be substantially that of the gas flowing through it, and to produce refrigeration

it is necessary to cause liquid refrigerant to flow into the evaporator under evaporating pressure.

In order to cause liquid refrigerant to flow from outer tube 20 to the evaporator at a rate which will be a function of changing conditions to which the system is subjected at any given time, I provide means for applying heat to a portion of outer tube 20 in advance of the inlet end 28 of the evaporator. According to my invention I do not use any extraneous source of heat because such heat would have to be accurately controlled in exact correlation with temperature-pressure conditions in the system. To do this manually is difficult, if not impossible, and to do it automatically will involve controls which are so prohibitively complicated and expensive is to be out of the question. Instead, I utilize the superheat of the gas leaving the evaporator, or the heat of the condensed refrigerant, or both, to cause the refrigerant to flow from outer tube 20 into the evaporator according to the demands of the evaporator. Thus, and as shown in Fig. 1, I take a portion of pipe 32, see Fig. 1 (which normally would lead from thte discharge end 34 of the evaporator directly to the compressor) and I bring it into heat exchange relation with a portion of the outer tube 20 as exemplified by coils 36.

If the temperature of the gas in coils 36 is above the saturation point, that is, if the gas is superheated, the sensible heat of the gas in coils 36 causes the saturated refrigerant to "boil" and "percolate," and thus delivers a supply of refrigerant liquid from tube 20 to the evaporator where it evaporates. The evaporation of the liquid refrigerant cools the evaporator and, as the frost point progresses toward the discharge end of the evaporator, it reduces the superheat of the gas reaching coils 36 so as to reduce, or nullify, the heat exchange between coils 36 and the refrigerant in tube 20 and thus reduce, or stop, the flow of refrigerant liquid to the evaporator. By this arrangement, during the cycling of the apparatus, no liquid refrigerant will flow from tube 20 to the evaporator as long as the temperature of the gas in coils 36 is at the saturation point, and vice versa.

This arrangement will suffice for some applications, but, for other applications, it may be necessary to increase the heat exchange between coils 36 and the refrigerant in outer tube 20 so as to provide an adequate supply of liquid refrigerant reaching the evaporator. To this end, instead of wrapping pipe 32 directly around tube 20 as at 36, I first bring the tube 32 into heat exchange relation with a portion of the capillary tube 18, as at 38, or at some other point in advance of the accumulator 16. Since the refrigerant flowing through the capillary tube is at a higher temperature than the gas flowing in tube 32, the temperature of the gas reaching coils 36 through return pipe 40 will be higher than the temperature of the gas initially flowing out of the evaporator. The increased heat thus applied to tube 20 correspondingly increases the propulsion of liquid refrigerant from outer tube 20 into the evaporator.

In another method of increasing the flow of refrigerant liquid to the evaporator, I provide one or more holes 42 through which refrigerant liquid flows from the lower portion of outer tube 20 into inner tube 22 to be carried to the evaporator by the flash gas flowing through inner tube 22. The flow of refrigerant through inner tube 22 correspondingly decreases the amount of refrigerant which, in the absence of hole or holes 42, will have to be propelled through outer tube 20 by the action of coil 36, and thus lowers the thermal inertia of the heat exchange between coil 36 and tube 20 with resultant improved refrigerant flow characteristics. Alternately, or additionally tube 22 may be provided with a hole or holes 43 at a level adjacent coil 36 for removing oil which may collect in tube 20 during the off cycle and which might impair the flow of refrigerant through tube 20. Holes also serve to admit refrigerant liquid into tube 22 in the same manner as hole 42.

It is to be noted that the flow capacity of tube 22 and hole or holes 42 or 43, is such that the flow of refrigerant to the evaporator will be below the minimum rate demanded by the evaporator and therefore coil 36 will retain control of the system at all times.

In Fig. 2, the system is adapted for use in connection with a very long evaporator circuit, that is one in which there will be objectionable "hunting" and "over-ride" or fluctuation in the operation of the system. According to my invention, I split the evaporator into two sections and I interpose one section 48 between the capillary tube 18 and the accumulator 16 and I interpose section 49 between the accumulator and the compressor in exactly the manner in which evaporator 10 is connected in Fig. 1. In this embodiment, a mixture of liquid refrigerant and flash gas is delivered to evaporator section 48 and the partly spent refrigerant is delivered to the accumulator from which it flows to evaporator section 49 which, as stated, takes the place of evaporator 10. In Fig. 2, I show evaporator section 48 as being smaller than evaporator section 49. This is preferable but conditions are conceivable under which the reverse situation may be desired or under which both evaporator sections should be of equal capacity, and these adjustments, can be made without departing from the spirit or scope of the invention. In order to avoid criss-crossing of lines, evaporator sections 48 and 49 are shown far apart but, in practice, sections 48 and 49, except for interposition of the accumulator and tubes 20 and 22 there-between, may be close together and, in effect, a continuation one of the other.

If desired tube 44 leading from coils 36 to the compressor, can be brought into heat exchange relation with capillary tube 18 as at 46. This is conventional practice and forms no part of the present invention.

In both embodiments, accumulator 16 does not trap and hold the refrigerant as in the case of a conventional accumulator from which the refrigerant can only be removed by the suction of the compressor pump. In my invention the refrigerant liquid reaching the accumulator flows by gravity to tube 20.

In Fig. 3, I have shown tube 22 as leading directly from the top of the accumulator to the end 26 of tube 20 instead of being disposed within tube 20 as in Figs. 1 and 2. This arrangement is satisfactory for some applications but it will lack the advantages attendant upon the use of holes 42 and/or 43.

In Fig. 4 I show a modification in which outer tube 20 is omitted and in which a single tube 50 is used. Tube 50 has a bottom bell shaped member 54 disposed near the bottom of accumulator 16A, which corresponds to accumulator 16, and has a hole 52 disposed near the top of the accumulator. In this construction, flash gas in the top of the accumulator flows through hole 54 to the inlet end of the evaporator in the manner above described and a coil 36A, corresponding to coil 36 is applied to the lower portion of accumulator 16A to "boil" and "percolate" the liquid refrigerant in the same manner as coil 36. In other words, the arrangement of Fig. 4 is also self regulating in that the amount of refrigerant percolated by coil 36A is a function of the temperature of the gas flowing there-through.

It will be seen from the foregoing that my invention has the advantages inherent in a dry system without the disadvantages inherent in the use of an accumulator placed between the evaporator and the compressor in which oil tends to collect and from which refrigerant can only be removed by the action of the compressor pump; that the flow of refrigerant to the evaporator is controlled by the temperature-pressure conditions in the evaporator; that because the supply of liquid refrigerant is close to the inlet end of the evaporator there is no appreciable delay in satisfying the demand of the evaporator as would be the case in a system in which the refrigerant must be withdrawn from the accumulator, compressed, condensed and de-

livered to the evaporator, that all of the foregoing is accomplished without any moving parts; that the refrigerant charge need not be unduly large; that the amount of refrigerant introduced into the system is not critical; that, as shown in Fig. 2, the system is readily adaptable for installations using a relatively long refrigerant circuit, and that the invention, as shown in any of the embodiments can be incorporated at no appreciable increase in cost or in weight, or in overall size or displacement.

What I claim is:

1. Refrigerating apparatus of the kind including a compressor and a condenser for supplying refrigerant to a capillary tube, an evaporator, and a suction conduit for delivering gaseous refrigerant from said evaporator to said compressor, flow control means comprising; an accumulator disposed to receive a mixture of liquid and gaseous refrigerant from said capillary, a first passageway defining a flow path for liquid refrigerant from said accumulator to said evaporator, a second passageway defining a flow path for gaseous refrigerant from said accumulator to said evaporator, and means for effecting heat exchange between portions of said first passageway and said suction conduit, whereby, when said suction conduit heats said first passageway, the flow of liquid from said accumulator toward said suction conduit is increased.

2. The structure recited in claim 1 and further characterized by a second heat exchange between portions of said suction conduit and said capillary tube, said second heat exchange being adapted to intensify the action of said first heat exchange.

3. The structure recited in claim 1 in which said second passageway is disposed inside of said first passageway.

4. The structure recited in claim 1 therebeing at least one hole in the wall of said second passageway, intermediate its ends, for establishing communication between the interiors of said first and second passageways.

5. The structure recited in claim 1 in which said second passageway is disposed within said first passageway and in which said passageways are bent into a U-shape with at least the bight portion of the U disposed below the accumulator and below the intake end of the evaporator.

6. The structure recited in claim 1, there being a metering opening formed in a wall of said second passageway at a point below said accumulator for establishing communication between the interiors of said first and said second passageways.

7. The structure recited in claim 1, there being a plurality of openings formed in a wall of said second passageway for establishing communication between the interiors of said first and second passageways, with at least one of said openings in the said lower portion of said second passageway and at least one of said openings in an upper portion of said second passageway.

8. The structure recited in claim 1 in which a portion of said evaporator is interposed between said accumulator and the discharge end of said capillary tube.

9. A refrigerating apparatus including a first evaporator section, a compressor-condensor unit for withdrawing refrigerant from the exhaust side of said first evaporator section for compressing and liquifying the refrigerant, a second evaporator section, a capillary tube leading from the discharge side of said compressor-condensor unit to the intake side of said second evaporator section, a refrigerant accumulator, a passageway leading from the discharge end of said second evaporator section to said accumulator, and means leading from said accumulator to the intake side of said first evaporator section for delivering refrigerant from said accumulator to said first evaporator.

10. The structure recited in claim 12 and means for bringing the refrigerant withdrawn from said first evaporator section into heat exchange relation with the refrigerant flowing toward said first evaporator section.

11. The structure recited in claim 9 in which said means includes a first passageway for delivering refrigerant gas from the upper portion of the accumulator to said first evaporator section, and a second passageway for delivering refrigerant liquid from the lower portion of the accumulator to said first evaporator.

12. Refrigerating apparatus including an accumulator adapted to receive liquid refrigerant at evaporating temperature, an evaporator for evaporating said liquid refrigerant therein, a passageway leading from said accumulator to the intake end of the evaporator for delivering liquid refrigerant from said accumulator to said evaporator, means including a condenser for withdrawing refrigerant from the discharge side of the evaporator, and for compressing it, for condensing it, a capillary leading from said condenser to said accumulator for delivering condensed refrigerant to said accumulator, and means for bringing the refrigerant flowing from the discharge end of the evaporator into heat exchange relation with a portion of said passageway.

13. The structure recited in claim 12 in which the refrigerant flowing from the discharge end of the evaporator is brought into heat exchange relation with a portion of said capillary tube before it is brought into heat exchange relation with the refrigerant in said passage.

14. The structure recited in claim 12 in which the end of the passageway which leads from the accumulator is at a lower level than the intake end of the evaporator.

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