

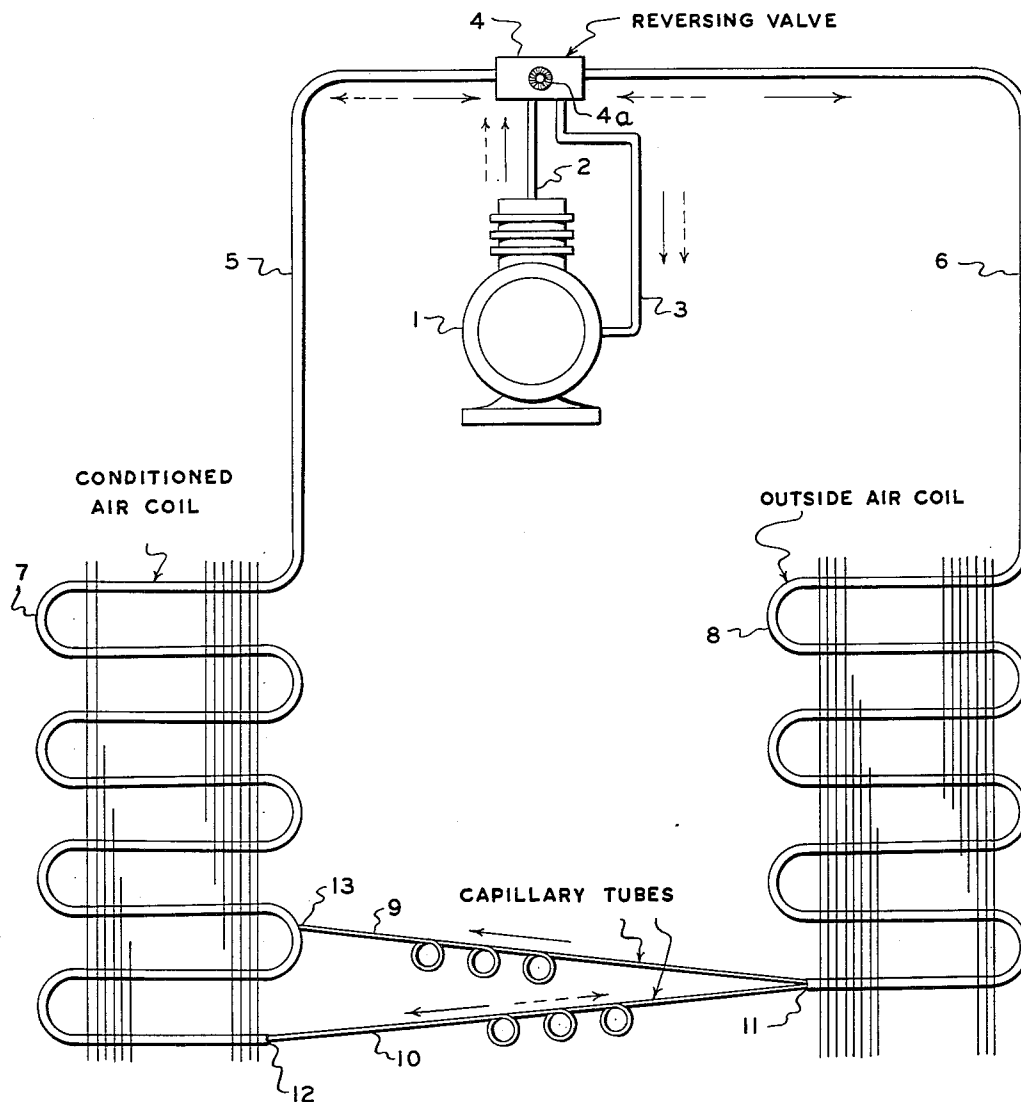
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AIR CONDITIONING APPARATUS

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AIR CONDITIONING APPARATUS

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My invention relates to air conditioning apparatus and more particularly to reverse cycle refrigeration systems for use in such apparatus.

Reverse cycle refrigeration systems may be included in air conditioners so that the air in the enclosure to be conditioned may be either heated or cooled for comfort. During the cooling operation the indoor coil or heat exchanger acts as an evaporator and the outdoor coil or heat exchanger acts as a condenser. Conversely during the heating operation the indoor coil acts as a condenser and the outdoor coil as an evaporator. This change in function is, of course, accomplished by reversing the direction of the refrigerant flow through the system.

A problem is, however, presented by the switchover of the system from the one operation or cycle to the other. During the summer cooling operation or cycle there is a considerable temperature differential between the indoor coil acting as an evaporator and its surrounding atmosphere, the room air; whereas during the winter heating operation or cycle there is ordinarily a much smaller differential between the outdoor coil acting as an evaporator and its surrounding atmosphere, the cold outside air. Thus the coil acting as an evaporator cannot pick up as much heat during the heating cycle as is absorbed during the cooling cycle. As a result the system cannot be operated with good efficiency at the same rate of refrigerant flow for both operations. A rate of refrigerant flow which allows a substantially complete vaporization of the refrigerant in the indoor coil during the cooling operation results in a "flooding through" of the outdoor coil during the heating operation. In other words, with the most efficient rate of flow for the cooling cycle also flowing during the heating cycle, the outdoor coil would be unable to vaporize all the refrigerant flowing therethrough so that liquid refrigerant would pass or "flood through" to the compressor. Since as is well known in the art, a "flood through" condition of the evaporator results in a lessened efficiency for the system than when all the refrigerant is vaporized in the evaporator, it is therefore desirable that the system have a lower rate during the heating cycle than during the cooling cycle.

In many refrigerating systems, however, capillary expansion tubes are used to regulate the flow of refrigerant from the condenser to the evaporator. These tubes are quite inexpensive compared to expansion valves and due to their extreme simplicity and total lack of moving parts are extremely satisfactory in operation. In fact they are generally used in refrigerating systems whenever possible. But since these tubes are of a fixed resistance to flow depending upon their length and cross-sectional area, they have not been readily usable with reverse cycle systems wherein it is necessary or desirable to pass a different rate of refrigerant flow through the system in its heating operation than in its cooling operation. Due to their fixed resistance to flow they will, if connected in the ordinary manner between the indoor coil and the outdoor coil, pass the same amount of refrigerant in both

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operations. If the capillary tube is designed for correct operation during the cooling cycle, an inefficient operation will result during the heating cycle. Or if the capillary tube should be designed for an efficient heating cycle, an inefficient cooling cycle will be caused. Thus in spite of their many advantages capillary tubes have not proven particularly satisfactory in refrigeration systems designed for both heating and cooling operations, as are used in many air conditioners.

Accordingly it is a primary object of my invention to provide a new and improved reversed cycle refrigeration system utilizing capillary tube expansion means, in which a different rate of refrigeration flow is obtained during the heating cycle than during the cooling cycle.

It is another object of my invention to provide a new and improved reverse cycle refrigeration system utilizing capillary tube expansion means, in which more restriction is offered to refrigerant flow during the heating cycle than during the cooling cycle whereby this system operates efficiently during both cycles.

In carrying out my invention in one preferred form thereof I provide a reverse cycle refrigeration system adapted for incorporation in an air conditioner for either heating or cooling the air within an enclosure. This refrigeration system includes a compressor and indoor and outdoor coils or heat exchangers. The system further includes a reversing valve for selectively connecting the discharge and the suction of the compressor to the outdoor and the indoor coils respectively during the cooling cycle and to the indoor and outdoor coils respectively during the heating cycle. For expanding the refrigerant of the system from condensing pressure to evaporating pressure a plurality of capillary tubes are provided; and in accordance with my invention these capillary tubes are connected in a novel manner so as to provide for efficient operation of the system during both the heating and the cooling cycles. Specifically at least one of the capillary tubes is connected between the ends of the coils remote from the reversing valve, and another of the capillary tubes is connected between the remote end of the outdoor coil and a point on the indoor coil intermediate the ends thereof. By their connection to the remote end of the outside coil both of these capillary tubes join the coil below the level of refrigerant liquefaction therein when the coil is acting as a condenser during the cooling cycle. Therefore, liquid refrigerant is fed to both the tubes during the cooling cycle and the maximum possible flow rate of the system is obtained. But the point at which the second capillary tube is connected to the indoor coil is above the level at which refrigerant liquefaction occurs in the indoor coil during the heating cycle when the indoor coil is acting as a condenser. As a result gaseous refrigerant is fed to the second capillary tube during the heating cycle and only the first capillary tube connected to the end of the coil receives liquid refrigerant. Since it receives gaseous refrigerant, the second capillary is not nearly so effective to pass refrigerant as during the cooling cycle and consequently only the first capillary connected to the end of the indoor coil passes an appreciable amount of refrigerant. Thus with only one capillary effective to pass refrigerant a smaller refrigerant flow rate occurs during the heating cycle than during the cooling cycle, which of course, is the desired result in order to permit complete vaporization of the refrigerant by the outdoor coil during the heating operation. In other words as a result of my novel capillary tube arrangement the operation of the system is automatically modified so that it has a lower refrigerant flow rate during the heating cycle than during the cooling cycle thereby allowing it to operate efficiently during both cycles.

The novel features which I believe to be characteristic of my invention are set forth with particularity in the ap-

pendent claims. My invention itself, however, both as to its organization and method of operation may be best understood by reference to the following description taken in conjunction with the accompanying drawing, the single figure of which is a diagrammatic view of a reverse cycle refrigerating system embodying my invention.

Referring now to the diagram I have illustrated therein a refrigerating system including a compressor 1 having a discharge line 2 and a suction line 3. The discharge and suction lines are both connected to a reversing valve 4. Also connected to the reversing valve 4 are a pair of conduits 5 and 6 which lead respectively to indoor and outdoor heat exchangers or coils 7 and 8. The indoor coil 7 is arranged for heating or cooling the air in the enclosure to be conditioned and the outdoor coil 8 is arranged for either rejecting heat to or picking it up from the outside atmosphere. The reversing valve 4 which in my preferred embodiment is manually controlled by means of a knob 4a is operative to selectively connect the discharge and suction of the compressor to conduits 5 and 6 and thus to the inside and outside coils 7 and 8 respectively. Or it may be operated to reverse that connection and connect the discharge line 2 to the outside heat exchanger 8 and the suction line 3 to the inside or conditioned air coil 7. More specifically if it is desired to set the system for a heating cycle the compressor discharge is connected to the inside coil 7 and the suction to the outside coil 8 whereas if it is desired to initiate a cooling cycle the discharge is connected to the outside coil 8 and the suction to the conditioned air coil 7. My invention is, however, not limited to a system in which the reversing valve is manually actuated for I contemplate that it may also be used in systems wherein the reversing valve is automatically actuated from one position or cycle to the other in response to a controlling condition. For example the reversing valve could be electrically actuated in response to a thermostat sensitive to room temperature.

Further included in my preferred system for the purpose of expanding the refrigerant from condensing pressure to evaporating pressure are a plurality of capillary tubes 9 and 10. In accordance with my invention these tubes are so connected and sized that an efficient flow rate is obtained both during the heating cycle and during the cooling cycle. More specifically, these capillary tubes are so connected that they offer more restriction to the flow of refrigerant during the heating cycle than during the cooling cycle whereby a lesser amount of refrigerant flows during the heating cycle. As explained hereinbefore it is desirable that less refrigerant flow within the system during the heating cycle than during the cooling cycle in order that the system may operate efficiently during both cycles. Because the temperature differential between the outdoor coil acting as an evaporator during the heating cycle and its surrounding atmosphere, the outside air, is normally much smaller than the differential between the inside conditioned air coil acting as the evaporator during the cooling cycle and its surrounding atmosphere, the room air, the outside coil cannot pick up as much heat during the heating cycle as the inside coil during the cooling cycle for the same amount of refrigerant passing there-through. Thus if the inside coil is to be run most efficiently during the cooling cycle which, as is well known in the art, means running the coil full, the outside coil must be run at some lesser amount during the heating cycle if "flood through" is to be avoided. By running the inside coil full during the cooling cycle I, of course, mean supplying the inside coil with the particular amount of refrigerant which will result in all the refrigerant being boiled off, i. e. evaporated, but not super-heated by the time it leaves the evaporator. The same amount of refrigerant supplied to the outdoor coil during the heating cycle would, however, not be all boiled off and this "flood through" of some liquid refrigerant would result. In other words if the coil acting as the evaporator is to be supplied with the proper amounts of refrigerant for maximum sys-

tem performance during both the heating and the cooling cycle some means must be provided for causing a lesser rate of flow during the heating cycle than during the cooling cycle.

As above mentioned, it is through my novel arrangement of the capillary tubes 9 and 10 that I accomplish this result. In other words it is through the manner of connection of the capillary tubes 9 and 10 within the refrigerating system that I cause the refrigerant flow during the cooling cycle to be such that the inside coil 7 then acting as an evaporator is run full and also cause a lower rate of flow to occur during the heating cycle so that the outside air coil then acting as an evaporator is also run substantially full but not flooded through. In my novel capillary tube arrangement both the capillaries 9 and 10 are connected at their one ends to the end 11 of the outside air coil remote from the reversing valve 4. At its other end the one capillary 10 is connected to the end 12 of indoor coil 7 remote from the reversing valve 4. But the capillary 9 is connected at its other end to the coil 7 at a point or tap 13 intermediate its ends. This point 13 may vary somewhat along the coil 7 according to the design of the system but it should be above the level at which liquid refrigerant begins to form when the coil is acting as a condenser, i. e., above the level of refrigerant liquefaction. In other words, by my invention the one capillary tube 10 taps the coil 7 at a point wherein the refrigerant flow is still gaseous rather than liquid when the coil 7 is acting as a condenser.

Although I have not shown it in the diagram, it should be understood that either or both of the capillary tubes 9 and 10 may be placed in heat exchange relation with the compressor suction line 3. This heat exchange relation, which, for example, may be accomplished by soldering the tubes or portions thereof to line 3 increases the efficiency of the system since it results in heat being picked up from the warm refrigerant flowing in the capillary tubes by the cool suction refrigerant flowing in line 3. The heat so extracted from the refrigerant in the capillary tubes increases by like amount the capacity of the refrigerant to absorb heat in the coil acting as the evaporator.

The manner in which my novel arrangement of capillary tubes operates to cause a lesser flow during the heating cycle than during the cooling cycle may be best understood by reference to the arrows shown in the diagram, wherein the refrigerant flow during the cooling cycle is indicated by the solid arrows and the flow during the heating cycle by the dotted arrows. As there shown, during the cooling cycle the discharge line 2 of the compressor is connected by the reversing valve 4 to the line 6 leading to the outside air coil 8 and the suction line 3 of the compressor is connected to the line 5 leading from the indoor coil 7. This of course causes a circulation of gaseous refrigerant from the compressor to the outside air coil 8 wherein it is condensed. The coil 8 is so designed that for normal ambient temperatures of the outside air surrounding the coil, the refrigerant is substantially all liquefied by the time it reaches the end 11 remote from the compressor. The liquid refrigerant then passes through both the capillary tubes 9 and 10 being expanded and greatly reduced in pressure. In other words the capillaries 9 and 10 cooperate to reduce the refrigerant from the condensing temperature and pressure to the evaporating temperature and pressure. Since the capillaries are both fed with liquid and both discharge into the same vessel, the coil 7, they in effect operate in parallel even though they do not discharge into coil 7 at exactly the same point. And with the capillaries so acting in parallel they are effective to allow a maximum predetermined amount of refrigerant to flow through the system. In fact the system will normally be designed so that with both capillaries passing refrigerant the coil 7 will be run full for maximum system performance. The refrigerant boiled off in the coil 7 will of course be returned through the line 5 to the suction 3 of the compressor.

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However with the system thus designed for efficient operation during the cooling cycle with both capillaries passing refrigerant, an inefficient operation would result if the same amount of refrigerant were passed during the heating cycle. The outside air coil 8 would not be able to boil off as much refrigerant as the indoor coil 7 during the cooling cycle and therefore an inefficient "flood through" condition of the outdoor coil would likely result. As indicated by the dotted arrows however my novel capillary tube arrangement operates so as to cause a lessened flow during the heating cycle. To place the system in the heating cycle the reversing valve 4 is operated to its other position wherein the discharge line 2 is connected to the line 5 leading to the indoor coil 7 and the suction line 3 is connected to the line 6 leading from the outside coil 8. With those connections made the hot gas leaving the refrigerant is fed into the inside or conditioned air coil 7. As it progresses through the coil 7 it cools off and as it reaches the lower end of the coil it liquefies. This point or level of liquefaction is however below the point 13 at which the capillary 9 taps the coil. In other words the operation of the coil 7 when acting as a condenser, is such that only gaseous refrigerant can enter the capillary tube 9. The refrigerant however does liquefy within the coil between the point or tap 13 and the lower end 12 of the coil to which the capillary 10 is joined, so that the capillary 10 is fed with liquid refrigerant.

Since as is well known in the art a capillary cannot pass as much gas as liquid, the result is that although the capillary tube 10 still passes approximately the same amount of refrigerant as during the heating cycle, the capillary 9 passes materially less refrigerant. In fact the action of capillary 9 when so fed with gaseous refrigerant is such that substantially the flow characteristics of the lower capillary 10 alone are obtained during the heating cycle. In other words the flow through the capillary 9 is so small due to the fact that gaseous rather than liquid refrigerant is fed thereto that the capillary 10 practically acts alone in passing refrigerant to the outside coil 8 now acting as an evaporator. With only one capillary acting effectively, the flow through the system is much restricted resulting in the flow during the heating cycle being very appreciably less than during the cooling cycle. In fact this flow is preferably such that the outer coil is run just full without either flooding through or superheating during the heating cycle. In other words due to the lessened flow during the heating cycle effected by my novel capillary tube arrangement the reduced pressure liquid entering the outside coil 8 through the capillary 10 flows at a rate that it is substantially boiled off without superheating by the time it reaches the end of the coil connected to the line 6 leading to the compressor.

As well as decreasing the system flow during the heating cycle my novel capillary tube arrangement also provides an additional advantageous result. Due to the smaller flow caused by only one rather than two capillaries conducting liquid, the system must of course move to a different steady state condition during the heating cycle than during the cooling cycle, and this difference in steady state conditions causes a different evaporator pressure to exist during the two cycles. Specifically, since less refrigerant is flowing during the steady state condition of the heating cycle, the pressure in the evaporator, outside coil 8, is lower than it is in the evaporator, inside coil 7, during the cooling cycle when both capillaries are conducting liquid. Because of the increased resistance to flow between the condenser, indoor coil 7, and the evaporator, outdoor coil 8, during the heating cycle the refrigerant backs up for some distance in the condenser short of tap 13 until the steady state condition is reached. Since the condenser thus holds more refrigerant than during the cooling cycle, the evaporator must hold less, and the less refrigerant therein the lower is

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the evaporator pressure. This decrease in evaporator pressure is advantageous because as a direct corollary it means a lower evaporator temperature; and the lower the evaporator temperature the more heat may be picked up by the evaporator for a given rate of flow. Thus not only does my improved capillary tube arrangement provide for a decreased flow during the heating cycle than occurs during the cooling cycle but also it results in a lower evaporator temperature during the heating cycle for more effective heat transfer between the cold outside air and coil 8.

Thus it will be seen that through my invention I have provided a new and improved reverse cycle refrigeration system in which a novel capillary tube arrangement results in efficient operation during both the cooling and the heating cycles. As a result of this capillary tube arrangement a greater flow of refrigerant is obtained during the cooling cycle than during the heating cycle whereby the system may be designed for efficient operation during the cooling cycle without causing "flood through" of the evaporator and the attendant inefficient compressor operation during the heating cycle. Since it utilizes capillary tubes as the expansion and flow controlling means, this system is inexpensive to produce but yet is practically fool-proof in operation.

While in accordance with the patent statutes I have described what is at present considered to be the preferred embodiment of my invention, it will be understood to those skilled in the art that various changes and modifications may be made therein without departing from my invention, and I, therefore, aim in the appended claims to cover all such changes and modifications as fall within the true spirit and scope of my invention.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. A reverse cycle refrigerating system for heating and cooling air for an enclosure comprising a compressor, first and second heat exchangers, means including a reversing valve for selectively connecting the discharge and the suction of said compressor to said first and second heat exchangers respectively during the heating cycle and to said second and first heat exchangers respectively during the cooling cycle, and a plurality of capillary tubes for expanding the refrigerant of said system from condensing pressure to evaporating pressure, at least one of said capillary tubes being connected between the ends of said heat exchangers remote from said reversing valve and at least a second of said capillary tubes being connected between said remote end of said second heat exchanger and a point on said first heat exchanger intermediate the ends thereof, said remote end of said second heat exchanger being below the level of refrigerant liquefaction in said second heat exchanger when said second heat exchanger is acting as a condenser during said cooling cycle thereby causing both said one tube and said second tube to be fed with liquid refrigerant during said cooling cycle, and said point being above the level of refrigerant liquefaction in said first heat exchanger when said first heat exchanger is acting as a condenser during said heating cycle thereby causing said one tube to be fed with liquid refrigerant and said second tube to be fed with gaseous refrigerant during said heating cycle, whereby said second capillary tube is less effective to pass refrigerant during said heating cycle than during said cooling cycle and a lower refrigerant flow rate is effected throughout said system.

2. A reverse cycle refrigerating system for heating and cooling air for an enclosure comprising a compressor, first and second heat exchangers, means including a reversing valve for selectively connecting the discharge and the suction of said compressor to said first and second heat exchangers respectively during the heating cycle and to said second and first heat exchangers respectively during the cooling cycle, and a pair of capillary tubes

for expanding the refrigerant of said system from condensing pressure to evaporating pressure, one of said capillary tubes being connected between the ends of said heat exchangers remote from said reversing valve and the other of said capillary tubes being connected between said remote end of said second heat exchanger and a point on said first heat exchanger intermediate the ends thereof, said remote end of said second heat exchanger being below the level of refrigerant liquefaction in said second heat exchanger when said second heat exchanger is acting as a condenser during said cooling cycle thereby causing both of said tubes to be fed with liquid refrigerant during said cooling cycle, and said point being above the level of refrigerant liquefaction in said first heat exchanger when said first heat exchanger is acting as a condenser during said heating cycle thereby causing said one tube to be fed with liquid refrigerant and said other tube to be fed with gaseous refrigerant during said heating cycle, whereby said other capillary tube is less effective to pass refrigerant during said heating cycle than during said cooling cycle and a lower refrigerant flow rate is effected throughout said system.

3. A reverse cycle refrigerating system for heating and cooling air for an enclosure comprising a compressor, an indoor coil and an outdoor coil, means including a reversing valve for selectively connecting the discharge and the suction of said compressor to said indoor and outdoor coils respectively during the heating cycle and to said outdoor and indoor coils respectively during the cooling cycle, and a plurality of capillary tubes for expanding the refrigerant of said system from condensing pressure to evaporating pressure, at least one of said capillary tubes being connected between the ends of said coil remote from said reversing valve and at least a second of said capillary tubes being connected between said remote end of said outdoor coil and a point on said indoor coil intermediate the ends thereof, said remote end of said outdoor coil being below the level of refrigerant liquefaction in said outdoor coil when said outdoor coil is acting as a condenser during said cooling cycle thereby causing both said one tube and said second tube to be fed with liquid refrigerant during said cooling cycle, and said point being above the level of refrigerant liquefaction in said indoor coil when said indoor coil is acting as a condenser during said heating

cycle thereby causing said one tube to be fed with liquid refrigerant and said second tube to be fed with gaseous refrigerant during said heating cycle, whereby said second capillary tube is less effective to pass refrigerant during said heating cycle than during said cooling cycle and a lower refrigerant flow rate is effected throughout said system.

4. A reverse cycle refrigerating system for heating and cooling air for an enclosure comprising a compressor, an indoor coil and an outdoor coil, means including a reversing valve for selectively connecting the discharge and the suction of said compressor to said indoor and outdoor coils respectively during the heating cycle and to said outdoor and indoor coils respectively during the cooling cycle, and a pair of capillary tubes for expanding the refrigerant of said system from condensing pressure to evaporating pressure, one of said capillary tubes being connected between the ends of said coils remote from said reversing valve and the other of said capillary tubes being connected between said remote end of said outdoor coil and a point on indoor coil intermediate the ends thereof, said remote end of said outdoor coil being below the level of refrigerant liquefaction in said outdoor coil when said outdoor coil is acting as a condenser during said cooling cycle thereby causing both of said tubes to be fed with liquid refrigerant during said cooling cycle, and said point being above the level of refrigerant liquefaction in said indoor coil when said indoor coil is acting as a condenser during said heating cycle thereby causing said one tube to be fed with liquid refrigerant and said other tube to be fed with gaseous refrigerant during said heating cycle, whereby said other capillary tube is less effective to pass refrigerant during said heating cycle than during said cooling cycle and a lower refrigerant flow rate is effected throughout said system.

References Cited in the file of this patent

UNITED STATES PATENTS

2,183,343	Alsing	Dec. 12, 1939
2,388,314	Eisinger	Nov. 6, 1945
2,404,010	Urban	July 16, 1946
2,589,384	Hopkins	Mar. 18, 1952
2,716,868	Biehn	Sept. 6, 1955