

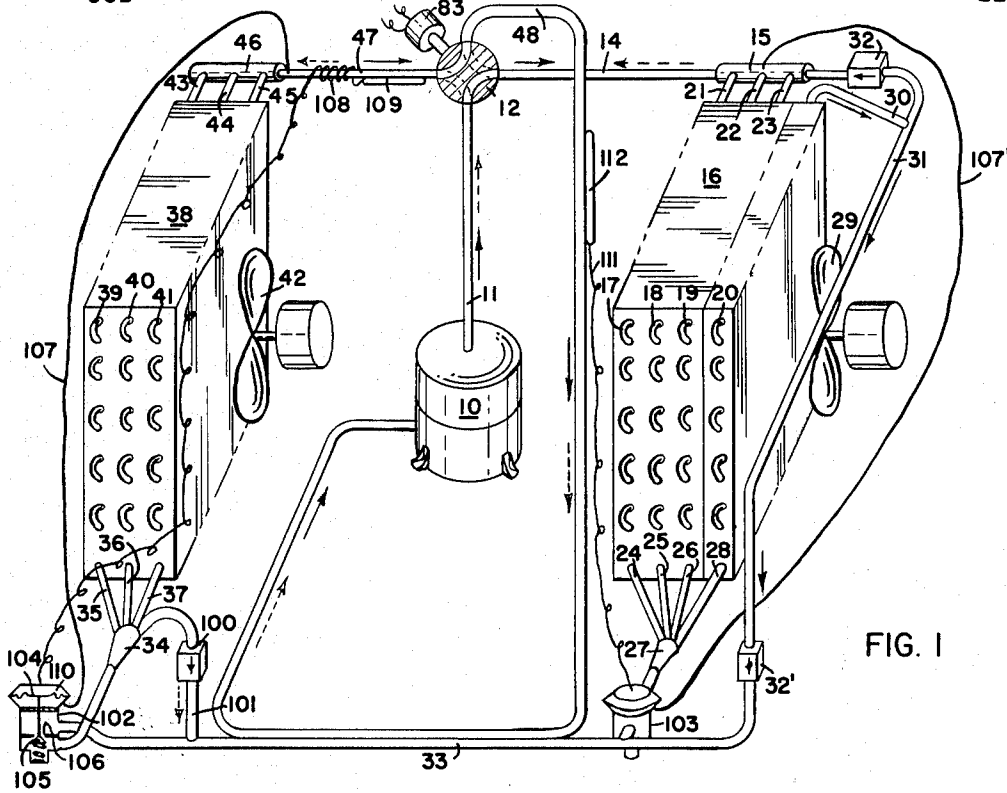
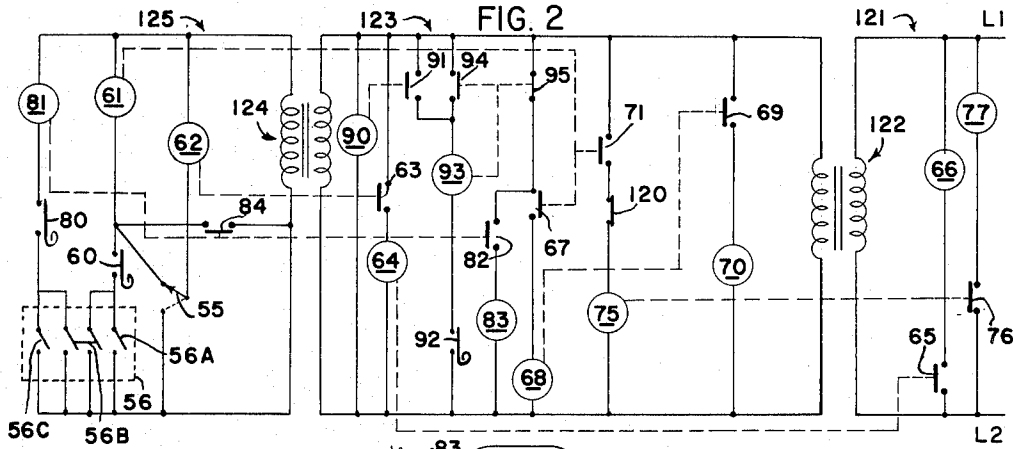
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REVERSE CYCLE HEAT PUMP

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REVERSE CYCLE HEAT PUMP

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This invention relates broadly to air conditioning apparatus and more particularly to air conditioning apparatus employing a refrigeration system operable under the reverse cycle principle. Apparatus of this type, generally known as heat pumps, are operable to supply either cool or warm air to an area to be treated. This invention is most particularly concerned with a novel heat pump having improved refrigerant metering means therein.

In a common commercial heat pump, heat is extracted from one source of air and rejected to another. When the heat pump is employed to supply heated air to an enclosure, heat extracted from a source of air flowing over an outdoor heat transfer coil is rejected to a stream of air flowing over a heat transfer coil located inside the enclosure being treated. Similarly when it is desired to cool air to an enclosure, heat extracted from a source of air flowing over the indoor heat transfer coil is rejected to a stream of air flowing over the outdoor heat transfer coil.

During heating operation in geographical areas wherein low outdoor temperatures are attained, the outdoor heat exchange coil often becomes coated with an insulating layer of frost which impedes the efficiency of the refrigerating system by reducing the heat transfer characteristics of this coil. Means are commonly provided for periodically reversing the refrigerant flow so that the unit reverts to cooling cycle operation. The outdoor coil temporarily acts as a condenser and the indoor coil as an evaporator to remove the coating of frost from the outdoor coil. In present heat pumps, there has been observed a transient phenomenon, commonly referred to as flood-back, at such time as the system temporarily reverts to the defrost cycle of operation. This phenomenon of flood-back which may last for but a moment or two is repeated each time the system switches to the defrost cycle and ultimately the compressor is damaged unless the flood-back is minimized.

When defrost is required, the reversing mechanism is actuated, permitting hot gaseous refrigerant from the compressor to flow through the discharge line, the reversing mechanism, the outdoor coil, the refrigerant metering means, the indoor coil and then back through the reversing mechanism to the compressor. The hot gaseous refrigerant which was in the indoor heat exchange coil flows from the indoor heat exchange coil to the reversing mechanism for a brief time. The expansion valve having a single thermal responsive control continues to sense the temperature of the hot gaseous refrigerant and tends to move toward a wide open position. Thus a slug of liquid refrigerant condensed in the outdoor heat exchange coil may surge through the thermal expansion valve into the indoor heat exchange coil in a liquid state. The excess of liquid refrigerant can flood back or can flood through to the compressor. After several such flood-backs the compressor may be damaged. This liquid refrigerant is relatively incompressible and if drawn into the cylinder head of the compressor, may cause broken valves. The liquid refrigerant may mix with the oil in the crankcase of the compressor. This oil-refrigerant mixture may foam, resulting in improper lubrication of the bearings and further damage to the compressor. Therefore, it is desirable that the expansion valve be provided with a thermal responsive control which rapidly

senses the temperature change and assumes proper control of the thermal expansion valve as soon as possible when the heat pump reverts from the heating cycle to the defrost cycle of operation.

5 An object of this invention is to provide an air conditioning apparatus having a reverse cycle refrigeration system wherein the disadvantages and defects in prior constructions are obviated.

10 Another object of this invention is to provide air conditioning apparatus having a reverse cycle refrigeration system with improved refrigerant metering means in such system.

15 Still another object of this invention is to provide a heat pump having improved thermal expansion valve means therein.

20 Still another object of this invention is to provide a heat pump with an improved thermal expansion valve means for minimizing the problem of flood-back when the heat pump changes from the heating cycle of operation to the defrost cycle of operation. Other objects and features of the invention will be readily apparent hereafter.

25 This invention relates to an improved heat pump comprising a refrigeration system including a compressor, reversing means, an outdoor heat exchange coil, refrigerant metering means and an indoor heat exchange coil interconnected in refrigerant flow relationship and means for defrosting the outdoor heat exchange coil during heating operation, the refrigerant metering means comprising an expansion valve adjacent the indoor coil, the expansion valve having control means rapidly responsive to reduced temperature in the line connecting the reversing means and the indoor heat exchange coil for substantially alleviating flood-back of refrigerant through the expansion valve when the reversing valve moves from heating position to defrost position.

30 This invention further relates to a method of operating a heat pump of the type having a reverse cycle refrigeration system comprising the steps of condensing refrigerant medium in an outdoor coil while evaporating refrigerant medium in an indoor coil to cool air passed over the indoor coil, condensing refrigerant medium in an indoor coil while evaporating refrigerant in the outdoor coil to heat air passed over the indoor coil, defrosting ice formed on the outdoor coil during heating operation by temporarily reversing the refrigeration system to cooling operation, controlling the initial flow of refrigerant medium from the outdoor coil to the indoor coil during defrost operation in response to the reduced temperature of the refrigerant medium substantially at the outlet from the indoor coil sensed by a fast acting control in heat exchange relationship with the refrigerant medium, and controlling the remaining flow of refrigerant medium from the outdoor coil to the indoor coil in response to fluctuations of the temperature of the refrigerant medium substantially at the outlet sensed by a slow acting control in heat exchange relationship with the refrigerant medium.

35 The present invention will be more fully understood when the following specification is read in conjunction with the accompanying drawing wherein:

40 FIGURE 1 is a diagrammatic view of the novel heat pump forming the subject of this invention, and

45 FIGURE 2 is a schematic wiring diagram of the electrical circuitry which is used to control the heat pump of FIGURE 1.

50 Referring now to FIGURE 1, the heat pump system illustrated comprises a motor compressor unit 10 which forwards hot gaseous refrigerant to the refrigerant flow reversing means 12, preferably a 4-way valve, through discharge line 11. The valve 12 is depicted in FIGURE 1 in cooling position. Refrigerant is routed from the dis-

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charge line through line 14 and header 15 to the outdoor heat exchange coil 16. The solid line arrows along the refrigerant lines indicate the direction of the flow of the refrigerant during the cooling cycle; the dotted line arrows indicate the direction of the flow of refrigerant during the heating cycle.

Outdoor air heat exchange coil 16, which may be provided with three vertical rows 17, 18, 19 of finned tubes, forms the main outdoor air heat exchange coil. In addition a vertical row 20 of finned tubes located upstream with respect to the outdoor air flow over the rows 17, 18 and 19 may be provided.

Check valve 32 prevents the flow of refrigerant into line 31, therefore refrigerant flows from header 15 through lines 21, 22 and 23, respectively, to each of the rows 17, 18 and 19, respectively, of the main outdoor heat exchange coil. One manner of circuiting the outdoor heat exchange coil and the indoor heat exchange coil is illustrated. However, it will be readily apparent to those skilled in the art that the refrigerant may be otherwise passed through the coils.

The refrigerant passes from the rows 17, 18 and 19, respectively, through distributor lines 24, 25 and 26 and distributor 27 to the thermal expansion valve 103. Thermal expansion valve 103, which is of conventional construction, acts as a check valve during the cooling cycle, therefore refrigerant flows through oversized line 28 to the row 20 which acts as a subcooler. Line 28 is oversized by comparison to lines 24, 25 and 26 to prevent flashing of refrigerant in subcooler row 20.

Fan 29 passes ambient air first over row 20 and then over rows 19, 18 and 17, respectively. The high pressure vaporous refrigerant flowing through the outdoor heat exchanger 16 is condensed in rows 17, 18 and 19 and is subcooled in row 20 to increase the cooling capacity of the heat pump system.

Outdoor heat exchange coil 16 is connected to the indoor heat exchange coil 38 by refrigerant metering means. Refrigerant flows from subcooler row 20 through line 30 into line 31. During cooling operation refrigerant flows through check valve 32' and line 33 to thermal expansion valve 102. From thermal expansion valve 102 refrigerant flows through distributor 34, and distributor lines 35, 36 and 37 into rows 39, 40 and 41 of indoor heat exchange coil 38.

Air passed over indoor heat exchange coil 38 by fan 42 is cooled by the evaporating liquid refrigerant in coil 38, thus providing the cooling effect of the heat pump system. The refrigerant flows through lines 43, 44 and 45 respectively to header 46 and then through line 47, reversing valve 12, and suction line 48 back to compressor 10.

During heating operation the solenoid-actuated reversing means 12 is moved to the dotted line position shown in FIGURE 1. Hot gaseous refrigerant flows from compressor 10 via discharge line 11, reversing valve 12 and line 47 to header 46. The refrigerant passes into the rows 39, 40 and 41 of the indoor heat exchange coil 38, which acts as a condenser during the heating cycle. The indoor air passing over coil 38 absorbs the latent heat of vaporization and some sensible heat from the refrigerant which condenses from a gas to a liquid. This warmed air heats the space to be conditioned. Thermal expansion valve 102 functions as a check valve during the heating cycle of operation, therefore refrigerant flows through check valve 100, line 101 and line 33 to thermal expansion valve 103. It is noted that check valve 32' prevents the flow of refrigerant through line 33 to line 31.

Refrigerant is metered through thermal expansion valve 103, distributor 27 and distributor lines 24, 25, 26, and 28 to the rows 17, 18, 19 and 20, respectively, of outdoor coil 16. The outdoor coil now acts as an evaporator to extract heat from the outdoor air passing over the coil. Refrigerant passes from rows 17, 18 and 19 through lines 21, 22 and 23 to header 15. The refrigerant passing

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through row 20 flows through line 30, line 31 and check valve 32 to header 15. Refrigerant then passes from the header through line 14, reversing valve 12 and line 48 back to compressor 10.

Referring again to FIGURE 1 it will be noted that the expansion valves 102 and 103 are of similar construction, however, the thermal responsive means for each differs. It will be understood that the description of valve 102 likewise applies to the description of valve 103.

Expansion valve 102 includes a diaphragm 104 adapted to move valve member 105 toward and away from port 106 to regulate the passage of refrigerant through valve 102. Pressure is imposed upon one side of diaphragm 104 by means of equalizer line 107 connected to line 47. Pressure is imposed against the opposite side of diaphragm 104 by means of a two-element control comprised of a fast-acting element and a slow-acting element.

The fast-acting element or capillary tube 108 is in intimate heat exchange relationship with line 47 and in the presently preferred embodiment of the invention is wrapped around line 47 to provide maximum heat transfer between the capillary tube and line 47. It will be understood that the capillary tube may be otherwise placed in contact with line 47 over a substantial portion of its length. The bulb 109, which has a heavier mass than capillary tube 108, may be clamped to the line 47.

The chamber 110, fast-acting element 108 and slow-acting element 109 are filled with a limited liquid charge. The charge may be of the same refrigerant as that used in the refrigeration system. The nature of the charge is such that a minor quantity of liquid is present in the tube and bulb during operation at normal temperature ranges. Above a predetermined temperature, the charge is all vapor. The volume of chamber 110 will be varied in response to temperature changes in line 47 sensed by the tube and the bulb. Upon a decrease in temperature in line 47, condensation of the charge occurs in tube 108. The pressure within the closed system corresponds to the temperature of tube 108. The valve responds relatively quickly to the decrease in temperature and corresponding pressure decrease. When the temperature in line 47 increases, tube 108 being of smaller mass, heats more quickly than bulb 109 and condensation of the charge occurs in the slow-acting element, the bulb, thus retarding the response of the valve.

Expansion valve 103 is provided with the usual capillary tube 111 and bulb 112 for sensing the temperature in suction line 48. Equalizer line 107' connects valve 103 with line 14.

Referring to FIGURE 2 there is shown an electrical schematic diagram for the presently preferred embodiment of the invention.

Electric current is supplied to a first circuit 121 via leads L1 and L2. Provided across the first circuit are indoor fan motor 66 and indoor fan starter contact 65. Also disposed across the first circuit are compressor motor 77 and compressor starter contact 76.

Step down transformer 122 may be provided to supply power to second circuit 123. If desired transformer 122 may be omitted and the first and second circuits may be merged and supplied from a common power source via leads L1 and L2.

Across the second circuit are outdoor fan motor 70 and outdoor fan relay contact 69. In parallel with motor 70 and contact 69 are compressor relay 75, safety switch 120 and compressor relay contact 71; defrost timer motor 90; and indoor fan starter relay 64 and indoor fan relay contact 63. Normally closed switch 120 represents one of the conventional safety devices, for example, a high pressure cutout. Also provided in circuit 123 are outdoor fan relay 68, compressor relay contact 67 and defrost relay contact 95. Reversing valve solenoid 83 and reversing valve relay contact 82 are in parallel with relay 68 and contact 67. In series across circuit 123 are defrost thermostat 92, defrost relay 93 and defrost relay contact

94. Defrost timer contact 91, periodically closed by a cam (not shown) actuated by motor 90, is in parallel with contact 94.

Control circuit 125 is connected to circuit 123 by step-down transformer 124. Within control circuit 125 are selector switch 56 having contacts 56A, 56B, and 56C; cooling thermostat 60; heating thermostat 80; compressor relay 61; reversing valve relay 81; indoor fan relay 62; fan selector switch 55; and reversing valve relay contact 84.

Operation

The electrical circuit will first be traced for cooling operation of the heat pump. Fan selector switch 55 may be manually placed in the solid line position for automatic operation or in the dotted line position for continuous operation. Assume switch 55 is in the solid line position. Selector switch 56 is placed in either the "cool" position (closing contact 56A) or the "auto" position (closing contact 56B). For purposes of illustration assume switch 56 is in the "cool" position and contact 56A is closed. Where there is a demand for cooling in the area to be treated, cooling thermostat 60 closes, energizing the compressor relay 61 and the indoor fan relay 62. The indoor fan relay contact 63 closes, energizing indoor fan starter relay 64. The indoor fan starter contact 65 closes and the indoor fan 66 starts.

With compressor relay 61 energized, compressor relay contact 67 also closes, energizing outdoor fan relay 68; outdoor relay contact 69 closes, and outdoor fan motor 70 starts. The compressor relay contact 71 closes energizing compressor starter relay 75. Compressor starter contact 76 closes and compressor 77 starts.

The compressor motor and outdoor fan motor stop when the cooling thermostat 60 opens. The heat pump system reverts to the condition as shown in the electrical schematic diagram.

During heating operation the fan switch 55 may be set in the automatic position as shown in the solid line and the selector switch 56 may be either in the "heat" position (closing contact 56C) or the "auto" position (closing contact 56B). It will be noted that with fan switch 55 in the dotted line position, indoor fan relay 62 will be continuously energized and the indoor fan will operate continuously.

In response to a demand for heating in the area to be treated, heating thermostat 80 closes, energizing reversing valve relay 81. Reversing valve relay contact 82 closes. This energizes reversing valve solenoid 83 moving the reversing valve to the position shown in dotted lines in FIGURE 1. Reversing valve relay contact 84 closes energizing compressor relay 61 and indoor fan relay 62. The indoor fan 66 is energized as above noted and the compressor relay contact 67 closes energizing the outdoor fan relay and outdoor fan motor 70 starts. Compressor relay contact 71 closes, energizing compressor relay 75 which closes contact 76 and energizes compressor motor 77. If the heating thermostat 80 is satisfied the thermostat opens and the compressor motor and outdoor fan motor stop.

The operation during defrost will now be described. Defrost timer motor 90 is always in operation. Periodically, in a presently preferred embodiment every 90 minutes, defrost timer contact 91 closes for approximately 20 seconds. When the defrost thermostat 92 senses a predetermined temperature, for example 45° F., the defrost thermostat 92 closes indicating that defrost is required. If the defrost thermostat 92 is closed and the timer contacts are closed, defrost relay 93 is energized and defrost relay contact 94 closes. Defrost relay contact 95 opens and reversing valve solenoid 83 shifts to the cooling position.

When the defrost thermostat senses a predetermined temperature, for example 65° F., the defrost thermostat 92 opens, defrost relay 93 is deenergized, reversing valve

solenoid 83 is energized and the unit shifts back to the heating cycle of operation.

It will be noted that during heating operation the hot gaseous refrigerant flows through line 47 to indoor coil 38. The temperature of the refrigerant may be on the order of from 200 to 225° F. Upon a demand for defrost the reversing valve 12 will be actuated as above noted. The hot refrigerant in line 47 is connected to the suction line and is drawn back to the compressor. Similarly the hot refrigerant contained within indoor coil 38 tends to be drawn toward header 46 and line 47 for return to the compressor. The thermal responsive control for expansion valve 102 is therefore sensing a high temperature and is tending to maintain the expansion valve wide open. The hot condensed liquid now flowing from outdoor coil 16 passes through line 33 and expansion valve 102 into indoor coil 38. Some liquid is vaporized in coil 38, however, because valve 102 is wide open, some refrigerant will tend to flood through coil 38. The charge within capillary tube 108 wound about line 47 responds rapidly to the changed condition in line 47. Tube 108 is now colder than bulb 109, thus any liquid in bulb 109 vaporizes and condenses in tube 108. The pressure within chamber 110 corresponds to the temperature of tube 108. Valve 102 thus moves quickly toward the closed position to minimize the quantity of liquid refrigerant flooding through the valve. Within a short time all the refrigerant leaving coil 38 is vaporized. As the load changes the suction temperature may rise. Bulb 109 being of greater mass, warms more slowly than tube 108. The liquid within the closed system evaporates and condenses at bulb 109. The action of valve 102 is retarded. Thus control of valve 102 is now provided by slow-acting element or bulb 109. By providing expansion valve 102 with the thermal responsive control as shown in FIGURE 1 the transient phenomenon of flood-back which results when the heat pump system temporarily reverts from heating operation to defrost is greatly minimized.

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

I claim:

1. In a heat pump, a compressor, a reversing mechanism connected to said compressor by means of a discharge line and a suction line respectively, a first coil, first conduit means connecting said reversing mechanism and said first coil, a second coil, refrigerant metering means connecting said first coil and said second coil, second conduit means connecting said second coil and said reversing mechanism, said refrigerant metering means including an expansion valve having a two-element control therefor, one element responding rapidly to temperature change, the second element responding more slowly to temperature change than said first element, both elements being in heat exchange relationship with the second conduit means, and defrost control means including means for actuating said reversing mechanism for removing frost formation from the first coil, said two-element control actuating said expansion valve to prevent the transient phenomenon of flood-back of refrigerant which occurs when the defrost control means actuates the reversing mechanism temporarily to the cooling position during heating operation to defrost the first coil.

2. In a heat pump, the combination of a compressor, reversing means, a discharge line and a suction line connecting said compressor and said reversing means, a first coil, first conduit means connecting said first coil and said reversing means, a second coil, refrigerant metering means connecting said first coil and said second coil, second conduit means connecting said second coil and said reversing means, whereby during the cooling operation refrigerant is passed from said compressor through

said reversing means, first conduit means, first coil, refrigerant metering means, second coil, second conduit means and reversing means back to said compressor and during heating operation, refrigerant is passed from said compressor through said reversing means, second conduit means, second coil, refrigerant metering means, first coil, first conduit means and reversing means back to said compressor, and means for periodically defrosting the ice formation from said first coil during heating operation including means for actuating said reversing means, said refrigerant metering means including an expansion valve having a control mechanism responsive to the reduced temperature of the refrigerant in said second conduit means when the reversing means is actuated to permit defrosting of the first coil during heating operation to minimize the flood-back of refrigerant through said expansion valve, said control mechanism including a slow-acting element and a fast-acting element, each in intimate heat exchange relationship with the second conduit means.

3. A heat pump as in claim 4 wherein said slow acting element comprises a capillary bulb and said fast-acting element comprises a capillary tube wrapped around the second conduit means.

4. The method of operating a heat pump of the type having a reverse cycle refrigeration system comprising the steps of condensing refrigerant in a first coil while evaporating refrigerant in a second coil to cool the area to be conditioned, condensing refrigerant in the second coil while evaporating refrigerant in the first coil to heat the area to be conditioned, periodically reverting to cooling cycle operation during heating cycle operation to remove frost which may form on the first coil, controlling the initial flow of refrigerant from the first coil to the second coil in response to a greatly reduced temperature of the refrigerant in the line connecting the second coil and a reversing means sensed by a fast-acting element, and controlling the normal flow of refrigerant from the first coil to the second coil in response to fluctuations of the temperature of the refrigerant in the line connecting the second coil and the reversing means sensed by the fast-acting element and a slow-acting element.

5. The method of operating a heat pump of the type having a reverse cycle refrigeration system comprising the steps of condensing a refrigerant medium in a first coil while evaporating refrigerant medium in a second coil to cool air passed over the second coil, condensing refrigerant medium in the first coil to heat air passed over the second coil, defrosting ice formed on the first coil during heating operation by temporarily reversing the refrigeration system to cooling operation, controlling the initial flow of refrigerant medium from the first coil to the second coil during defrost operation in response to the reduced temperature of the refrigerant medium substantially at the outlet from the second coil sensed by a fast-acting control in heat exchange relationship with the refrigerant medium, and controlling the remaining flow

of refrigerant medium from the first coil to the second coil in response to fluctuations of the temperature of the refrigerant medium substantially at the outlet of the second coil sensed by the fast-acting control and a slow-acting control in heat exchange relationship with the refrigerant medium, whereby the flood-back of refrigerant medium resulting when the heat pump temporarily reverts to cooling operation to defrost the first coil is greatly minimized.

6. In a heat pump, the combination of a compressor, reversing means, a discharge line and a suction line connecting said compressor and said reversing means, a first coil, first conduit means connecting said first coil and said reversing means, a second coil, refrigerant metering means connecting said first coil and said second coil, second conduit means connecting said second coil and said reversing means whereby during the cooling operation refrigerant is passed from said compressor through said reversing means, first conduit means, first coil, refrigerant metering means, second coil, second conduit means and reversing means back to said compressor and during heating operation, refrigerant is passed from said compressor through said reversing means, second conduit means, second coil, refrigerant metering means, first coil, first conduit means and reversing means back to said compressor, and means for periodically defrosting the ice formation from said first coil during heating operation including means for actuating said reversing means, said refrigerant metering means including first and second expansion valves, control mechanism for said first expansion valve including a slow acting element and a fast acting element, each in intimate heat exchange relationship with said second conduit means, said fast acting element being operable to initially override said slow acting element in response to reduced refrigerant temperature in said second conduit means when said reversing means is actuated to defrost said first coil during heating operation to regulate said first expansion valve to minimize flood-back of refrigerant through said first expansion valve, and control mechanism for said second expansion valve including an element in intimate heat exchange relationship with said suction line and operable during heating operation to regulate flow of refrigerant through said second expansion valve in response to temperature changes in said suction line.

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