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(54) Method and apparatus for satisfying heating and cooling demands and control therefor.

(57) Apparatus (10) for satisfying heating and cooling demands comprising a cooling circuit (12) for satisfying the cooling demand and including a high pressure side (60) and a low pressure side (62); a heating circuit (14) for satisfying the heating demand and including a booster compressor (26) for drawing and compressing refrigerant vapor from the high pressure side (62) of the cooling circuit (12), and return means (72, 74, 32) for returning refrigerant from the heating circuit (14) to the cooling circuit (12); characterized by a sensor (Th.S.) for sensing the temperature of vapor discharged from the booster compressor (26); and means (38, 40, 42, 44, 46) responsive to the sensor (Th.S.) for terminating the heating action of the heating circuit (14) when the temperature of the vapor discharged from the booster compressor (26) exceeds a preset temperature.

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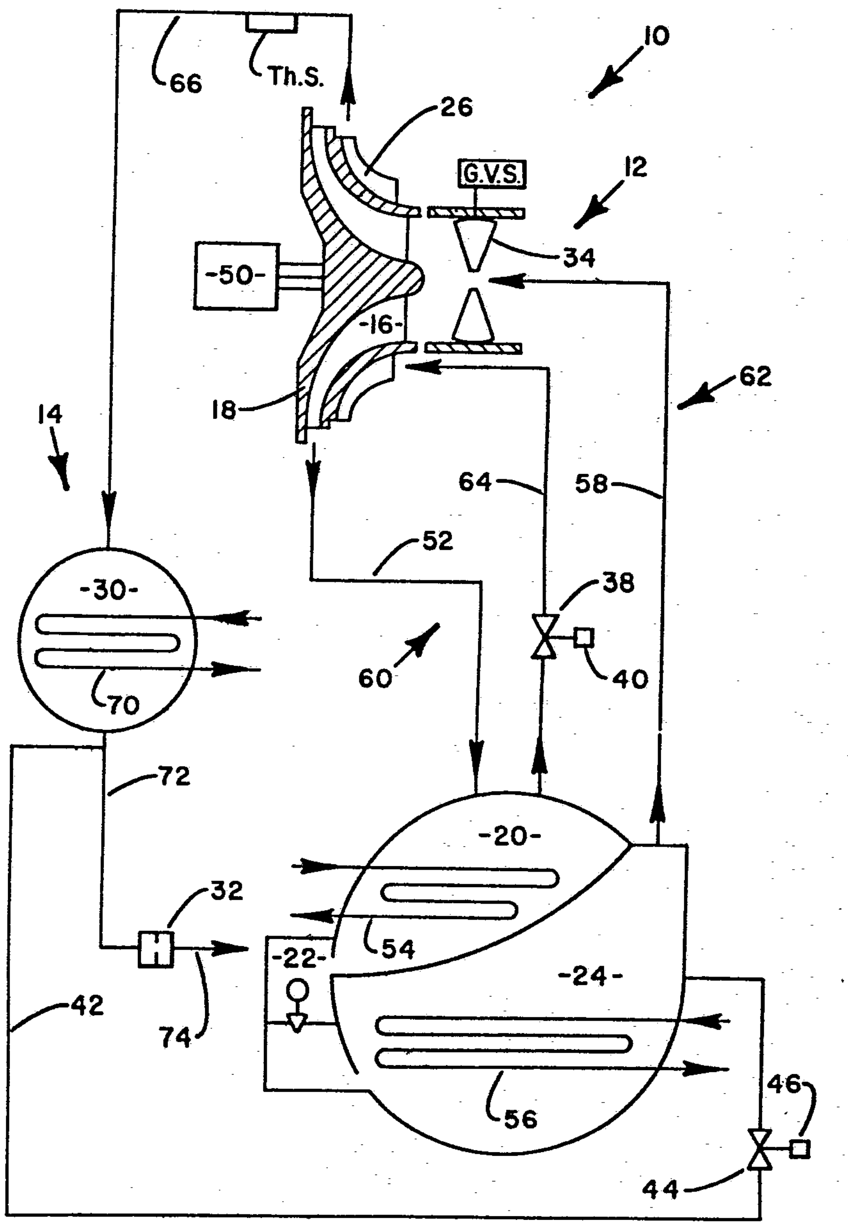
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FIG. 1

Method and Apparatus for Satisfying Heating and Cooling Demands and Control Therefor

This invention relates generally to refrigeration, and more specifically to refrigeration methods and apparatus for simultaneously satisfying heating and cooling demands.

Refrigeration apparatus or machines are frequently employed to cool a fluid such as water which is circulated through various rooms or enclosures of a building to cool these areas. Often, the refrigerant of such machines rejects a relatively large amount of heat at the condenser of the machine. This rejected heat is commonly dissipated to the atmosphere, either directly or via a cooling fluid that circulates between the condenser and cooling tower. Over a period of time, the rejected heat represents a substantial loss of energy, and much attention has been recently directed to reclaiming or recovering this heat to satisfy a heating load or demand.

One general approach to reclaiming this heat is to employ a booster compressor to draw and further compress a portion of the refrigerant vapor passing through the condenser of the refrigeration machine. This further compressed vapor is then passed through a separate, heat reclaiming condenser.

A heat transfer fluid is circulated through the heat reclaiming condenser in heat transfer relation with the refrigerant passing therethrough. Heat is transferred from the refrigerant to the heat transfer fluid, heating the fluid and condensing the refrigerant. The heated heat transfer fluid may then be used to satisfy a present heating load or the fluid may be stored for later use, and the condensed refrigerant is returned to the refrigeration circuit for further use therein.

With refrigeration machines having both a refrigeration, or cooling, circuit and a heating circuit as described above, it is desirable to vary the capacities of the heating and cooling circuits to meet changing heating and cooling loads, and typically this is done by varying the refrigerant flow rates through the circuits. Difficulties may arise, though, when the refrigerant flow rate through the heating circuit is very low. More particularly, under such conditions, the booster compressor may significantly raise the temperature of the refrigerant vapor passing therethrough, and the refrigerant may approach temperature levels which cause the refrigerant to chemically breakdown. Such a chemical breakdown of the refrigerant may produce acidic compounds which can damage the structure of the refrigeration machine. Preventing excessive vapor temperature in the heating circuit is complicated by a number of facts. First, it is preferred to vary the capacities of the heating and cooling circuits substantially independent of each other. Thus, the capacity of the cooling circuit may be anywhere between its minimum and maximum values when excessive vapor temperatures are approached in the heating circuit. Second, with certain refrigeration machines of the general type described above, the specific manner for preventing excessive vapor temperatures in the heating circuit will vary in accordance with the actual capacity of the cooling circuit when these excessive temperatures are approached.

In addition to the foregoing, as alluded to above, the heating load on refrigeration machines of the general type described above will not remain constant, but rather will vary with changes in various factors such as ambient temperature. If the heating load falls below a certain value, it may be preferred to terminate heating action of the heating circuit of the refrigeration machine and satisfy the heating load in some other manner. With many refrigeration machines having both heating and cooling circuits, it is necessary to maintain a continuous flow of refrigerant vapor through the heating circuit, however, even when the heating action thereof is terminated, to prevent the heating circuit from overheating. If, when the heating action of the heating circuit is terminated, the vapor supplied thereto is at normal supply pressure for that circuit -- that is, substantially at the pressure of the condenser of the refrigeration unit -- then a relatively large vapor mass flow through the heating circuit is needed to maintain satisfactory temperatures therein, and the booster compressor uses a relatively large quantity of power to compress this vapor while no useful work is being accomplished.

Although it is known that reducing the mass flow through the booster compressor will reduce the power requirements thereof, if the compressor is a centrifugal compressor, a mere reduction in the vapor mass flow therethrough, without a simultaneous decrease in the pressure differential across the compressor, will cause the compressor to operate near surge conditions. As is well recognized, it is undesirable to operate a centrifugal compressor at or near surge conditions due to the high discharge temperatures and mechanical vibrations that are generated at such times. If the refrigerant vapor is supplied at a much lower than normal supply pressure when the heating load on the heating circuit is terminated, the mass or weight flow of refrigerant vapor may be concomitantly reduced thereby decreasing the consumption of wasted energy. Further, by lowering the pressure differential across the compressor, while simultaneously lowering the mass flow therethrough, the compressor will be prevented from operating at or near surge conditions.

In view of the above, a first aspect of the present invention relates to preventing excessive vapor temperatures in the heating circuit of a booster type, heat reclaiming refrigeration machine. More particularly, this first aspect of the present invention relates to apparatus for satisfying heating and cooling demands comprising a cooling circuit having a high pressure side and a low pressure side, and a heating circuit including a booster compressor for drawing and compressing refrigerant vapor from the high pressure side of the cooling circuit. The apparatus also comprises a sensor for sensing the temperature of the vapor discharged from the booster compressor, and a control responsive to the sensor for terminating the heating action of the heating circuit when the temperature of the vapor discharged from the booster compressor exceeds a preset temperature.

A second aspect of the present invention relates to terminating the heating action of a heating circuit of a booster type, heat reclaiming refrigeration machine when the heating load on the heating circuit falls below a predetermined level. More specifically, this aspect of the present invention relates to apparatus for satisfying heating and cooling demands comprising a cooling circuit having a high pressure side and a low pressure side, and a heating circuit including a booster compressor for drawing and compressing refrigerant vapor from the high pressure side of the cooling circuit to satisfy a heating load. The load on the heating is monitored, with a continuous flow of refrigerant through the heating circuit being maintained regardless of changes in the heating load thereon. When the heating load upon the heating circuit decreases below a predetermined level, the pressure differential across the heating circuit is substantially equalized to reduce the power consumed by the booster compressor. Further, in a preferred embodiment, when the heating action of the heating circuit is terminated, the pressure of the refrigerant vapor delivered to the booster compressor is reduced to decrease the vapor mass flow through the heating circuit required to maintain satisfactory temperature therein.

This invention will now be described by way of example, with reference to the accompanying drawings in which:

Figure 1 is a schematic representation of a vapor compression heating reclaiming refrigeration machine uniquely designed to prevent excessive vapor temperatures in the heating circuit of the machine;

Figure 2 is a schematic drawing of an electric control circuit for the refrigeration machine shown in Figure 1; and

Figure 3 is a schematic representation of a heat reclaiming refrigeration machine uniquely designed to terminate heating action when the heating load on the machine falls below a preset level.

Referring to Figure 1, there is depicted refrigeration machine 10 employing teachings of the present invention. Machine 10 includes, generally, cooling circuit 12 and heating circuit 14. Cooling circuit 12, in turn, includes primary compressor such as first stage 16 of two stage compressor 18, primary condenser 20, primary expansion means 22, and evaporator 24. Heating circuit 14 includes booster compressor means such as second stage 26 of compressor 18, heat reclaiming condenser 30, and auxiliary expansion means such as orifice 32. Inlet guide vanes 34 are provided to control the refrigerant flow through first stage 16 of compressor 18 and, thus, through cooling circuit 12. Positioning means (not shown) are provided to move guide vanes 34 between minimum and maximum flow positions. Valve 38 is utilized to regulate the refrigerant flow through second stage 26 of compressor 18 and, hence, through heating circuit 14. Positioning means such as reversible electric motor 40 is provided for moving valve 38 between minimum and maximum flow positions. Vent line 42 connects heating circuit 14 with a low pressure region such as evaporator 24, vent line valve 44 regulates refrigerant flow through the vent line, and positioning means such as electrically actuated solenoid 46 moves the vent line valve between open and closed positions. Drive means such as electric motor 50 is employed to simultaneously drive first and second stages 16 and 26 of compressor 18.

An electric control circuit for motors 40 and 50 and solenoid 46 is shown in Figure 2. To simplify references to Figure 2, the Figure includes numerical references 1-16 at the left thereof to indicate various lines in the Figure. Solenoid 46 is shown in line 8 of Figure 2 while motors 40 and 50 are shown, respectively, in lines 13 and 16 of the Figure. Solenoid 46 is connected to a first source of electrical energy represented by line L-1 and L-2 in Figure 2. Further, Figure 2 shows motors 40 and 50 connected, respectively, to second and third electrical energy sources, with lines L-3 and L-4 representing the second source and lines L-5 and L-6 representing the third source of electrical energy. As will be apparent to those skilled in the art, numerous types of electrical energy sources may be used with the circuit shown in Figure 2. One suitable set of sources, for example, provides approximately a 115 volt alternating current between lines L-1 and L-2, about a 28 volt alternating current between lines L-3 and L-4, approximately a 460 volt alternating current between lines L-5 and L-6, with each of the above currents having a frequency of about 60 hertz.

The circuit shown in Figure 2 includes numerous relay coils and relay contacts controlled thereby, and attention is directed to the right-hand side of Figure 2 where adjacent to each line having a relay coil there are identified the lines containing relay contacts controlled by that coil. Also, the symbol "K" designates the relay coil while the symbol "CR" designates the contacts controlled thereby. For example, coil K3 in line 1 controls contacts CR3 in lines 1 and 3, and timer relay coil KT1 in line 11 controls contacts CRT1 in line 12. As is customary in the art, the relay contacts shown in Figure 2 are illustrated in their inactive or de-energized position. Further, it should be understood that the controls for refrigeration machine 10 include a variety of switches and other devices not shown in Figure 2. For example, the controls include a water pump switch and a plurality of indicator lights. The addition of these devices is well within the purview of those skilled in the art, and they have been omitted from Figure 2 for the sake of clarity.

Program Timer PT is schematically shown in line 5 of Figure 2. Program Timers are well known in the art and are used to produce a sequence of events. Program Timer PT of machine 10 controls switches PT-1, PT-2, PT-3, and PT-4 located, respectively, in lines 5, 6, 4, and 1 of Figure 2, and the Program Timer runs these switches through an ordered series of steps. If the Program Timer is de-energized at some point in its sequence, when re-energized the timer will restart at the point in its sequence where it was de-energized. Furthermore, as is well known in the art, the Program Timer will run for a period of time between each step in its sequence, and each time period may be individually adjusted.

Under initial conditions, switches PT-1 and PT-2 are in the positions shown in full lines in Figure 2, switch PT-3 is open, and switch PT-4 is closed. At the same time, thermostatic switch Th.S. in line 9 of Figure 2 is closed and, hence, relay coil K1 in line 9 is energized. Because coil K1 is energized, contacts CR1 in line 4 are closed and contacts CR1 in line 10 are open. With contacts CR1 open in line 10, timer relay KT1 (discussed in greater detail below) in line 11 is de-energized; and with relay KT1 de-energized, contacts CRT1 in line 12 are closed. Because contacts CRT1 are closed, relay coil K2 in line 12 is energized. As a result of this, contacts CR2 in line 13 are closed, and contacts CR2 in lines 8 and 14 are open.

To initiate operation of machine 10, start switch St.S. in line 2 of Figure 2 is manually closed. Referring to Figure 2, current passes through closed switch PT-4 in line 1 and through start switch St.S., energizing relay coils K3 and KT2 in lines 1 and 2 respectively. Coil KT2 is a delay timer relay which closes contacts CRT2 in line 7 after a short time delay such as one minute, and coil KT2 maintains these contacts closed thereafter so long as the coil is energized. The energization of coil K3 closes contacts CR3 in lines 1 and 3. Closed contacts CR3 in line 1 are in parallel with start switch St.S. and thus provides a holding current for relay coils K3 and KT2, allowing release of the start switch. When contacts CR3 in line 3 close, current is conducted through switch PT-4, through closed contacts CR1 in line 4, through closed contacts CR3 in line 3, through switch PT-1, and through normally closed contacts CRT3 in line 5, energizing Program Timer PT.

After Program Timer PT is energized, switch PT-1 moves to the position shown in broken lines in Figure 2. This provides a holding current for Program Timer PT via line 5 and normally closed contacts CR4 and CRT3 therein. Next, switch PT-2 moves to the position shown in broken lines in Figure 2, energizing oil pump relay coil o.p. which then starts an oil pump (not shown) for compressor motor 50. After a short time delay to allow oil pressure in compressor motor 50 to increase to an acceptable level, Program Timer PT opens switch PT-4 and then the Program Timer closes switch PT-3 to start compressor motor 50. With switch PT-4 open, the process of starting compressor motor 50 will continue only if safety switch Saf.S. in line 2 of Figure 2 is closed. Safety switch Saf.S. schematically represents a plurality of safety switches which prevent or terminate operation of compressor motor 50 upon the development of undesirable conditions such as low oil pressure in the compressor motor. Additional safety devices are well known in the art and may be easily used with machine 10 by those skilled in the art.

If all of the parameters sensed by safety switch Saf.S. are within acceptable ranges, the safety switch is closed. Current passes through safety switch Saf.S., through closed contacts CR1 in line 4, through closed contacts CR3 in line 3, and through switch PT-3, energizing relay coil K4 in line 3. When relay coil K4 is energized, relay contacts CR4 in lines 3 and 16 close and contacts CR4 in line 5 open. Contacts CR4 in line 3 are in parallel with switch PT-3 and provide a holding current for relay coil K4, allowing switch PT-3 to open. Contacts CR4 in line 5 are in series with Program Timer PT; and when these contacts open, the program timer is de-energized. Contacts CR4 in line 16 are in series with compressor motor 50; and when these contacts close, the compressor motor is activated. In practice, a motor starter (not shown) may be activated in response to the energization of coil K4 and employed to facilitate starting compressor motor 50. Thus, compressor motor 50 is started, refrigeration machine 10 is put into operation, and Program Timer PT is de-energized. As will be appreciated, if safety switch Saf.S. is open when switch PT-3 closes, then coil K4 is not energized and motor 50 is not started until the safety switch closes. Similarly, if safety switch Saf.S. opens while motor 50 is operating, coil K4 is de-energized, contacts CR4 in line 16 open, and motor 50 is deactivated until the safety switch recloses.

Referring back to Figure 1, in operation, first stage 16 of compressor 18 discharges hot, compressed refrigerant vapor into primary condenser 20 via line 52. Refrigerant passes through primary condenser 20, rejects heat to an external heat exchange medium such as water circulating through heat exchange coil 54 located therein and condenses. The condensed refrigerant flows through primary expansion means 22, reducing the temperature and pressure of the refrigerant. The expanded refrigerant enters and passes through evaporator 24 and absorbs heat from an external heat transfer medium such as water passing through heat exchange coil 56 which is positioned within the evaporator. The heat transfer medium is thus cooled and the refrigerant is evaporated. The cooled heat transfer medium may then be used to satisfy a cooling load, and the evaporated refrigerant is drawn from evaporator 24 in line 58 leading back to first stage 16 of compressor 18.

As described above, first stage 16 and primary expansion means 22 separate cooling circuit 12 into high pressure side 60 and low pressure side 62, and booster inlet line 64 is provided for transmitting refrigerant vapor from the high pressure side of the cooling circuit to second stage 26 of compressor 18. In the embodiment depicted in Figure 1, inlet line 64 is connected to condenser 20 and transmits a portion of the refrigerant vapor passing through the condenser to second stage 26 of compressor 18. Alternately, line 64 could be directly connected to discharge line 52. Second stage 26 of compressor 18 further compresses the vapor transmitted thereto, further raising the temperature and pressure of the vapor. This further compressed vapor is discharged into line 66, leading to heat reclaiming condenser 30.

The refrigerant vapor enters and passes through heat reclaiming condenser 30 in heat transfer relation with a heat transfer fluid such as water passing through heat exchange coil 70 disposed within the heat reclaiming condenser. Heat is transferred from the refrigerant vapor to the fluid passing through coil 70, heating the fluid and condensing the refrigerant. The heated heat transfer fluid may then be employed to satisfy a heating load. Refrigerant condensed in heat reclaiming condenser 30 passes therefrom back to cooling circuit 12 via return means including auxiliary expansion means 32 and refrigerant lines 72 and 74.

More particularly, condensed refrigerant from heat reclaiming condenser 30 flows through orifice 32 via line 72, reducing the pressure and temperature of the refrigerant. Refrigerant line 74 transmits refrigerant from orifice 32 back to cooling circuit 12, specifically primary expansion device 22 thereof, for further use in the cooling circuit.

Guide vanes 34 may be controlled in response to any one or more of a number of factors indicative of changes in the load on cooling circuit 12 to vary the capacity thereof. For example, guide vanes 34 may be controlled in response to the temperature of the fluid leaving heat exchanger 56 of evaporator 24. As the cooling load increases or decreases, guide vanes 34 move between their minimum and maximum flow positions to increase or decrease, respectively, the refrigerant flow rate through cooling circuit 12. Similarly, valve 38 may be governed in response to any one or more of a number of factors indicating changes in the load on heating circuit 14 to vary the capacity thereof. For example, valve 38 may be controlled in response to the temperature of the fluid discharged from heat exchanger 70 of heat reclaiming condenser 30. Referring to Figure 2, when the heating load is increasing, normally open switch 76 in line 13 is closed, activating motor 40 to move valve 38 toward its maximum flow position to increase the flow rate through heating circuit 14. In contrast, when the heating load is decreasing, normally open switch 78 in line 15 is closed, activating motor 40 to move valve 38 toward its minimum flow position to reduce the flow rate through heating circuit 14. It should be noted that switches 76 and 78 may be mechanical devices, or these switches may be solid state electronic elements.

Thus, with the above-discussed control of valve 38, as the heating load on machine 10 decreases, the refrigerant flow rate through heating circuit 14 also decreases. Moreover, as the flow rate through booster compressor 26 decreases, the temperature of the vapor discharged therefrom tends to increase. As discussed above, if the refrigerant flow rate through booster compressor 26 is very low, the temperature of the vapor discharged therefrom may approach a level where the refrigerant may chemically breakdown into components that may damage the structure of machine 10. In light of this, machine 10 is uniquely designed to terminate the heating action of heating circuit 14, thus reducing temperatures therein, when the temperature of the vapor discharged from booster compressor 26 exceeds a preset value.

In the preferred embodiment illustrated in Figures 1 and 2, the above-mentioned heat terminating means includes thermostatic switch Th.S. and vent line 42. Thermostatic switch Th.S. is positioned in heat transfer relation with refrigerant vapor discharged from second stage 26 of compressor 18, for example the thermostatic switch may be secured to line 66. Thermostatic switch Th.S. is electrically located in line 9 of Figure 2, in series with relay coil K1 and, as previously mentioned, the thermostatic switch is normally closed. When the temperature of the vapor discharged from booster compressor means 26 exceeds the preset value, thermostatic switch Th.S. opens. When this occurs, referring to Figure 2, relay coil K1 is de-energized, opening contacts CR1 in line 4 and closing contacts CR1 in line 10 which are associated with Timer Relay KT1 in line 11. Timer Relay KT1 is a delay off, solid state timer that is electronically locked into an energized state when contacts CR1 in line 10 close, and the timer relay remains energized so long as contacts CR1 in line 10 remain closed and for a predetermined length of time after these contacts open. When timer relay KT1 in line 11 is activated, contacts CRT1 in line 12 open, deactivating relay coil K2. This, in turn, opens contacts CR2 in line 13 and closes contacts CR2 in lines 8 and 14. With contacts CR2 in line 13 open, motor 40 cannot be activated by the closing of switch 76 to open valve 38. In fact, with contacts CR2 in line 14 closed, switch 78 is bypassed and motor 40 is energized to move valve 38 towards its minimum flow position, decreasing the refrigerant flow rate through heating circuit 14. At the same time, when contacts CR2 in line 8 close, vent solenoid 46 is activated.

Referring back to Figure 1, activation of solenoid 46 opens vent line valve 44, allowing fluid flow through vent line 42. Heating circuit 14 is thus brought into communication with low pressure side 62 of cooling circuit 12. Specifically, a first end of vent line 42 is connected to line 72 and a second end of the vent line is connected to evaporator 24. Alternately, as will be apparent to those skilled in the art, the first end of vent line 42 could be connected to heat reclaiming condenser 30 or to discharge line 66, and the second end of the vent line could be connected to inlet line 58. Since the pressure in evaporator 24 is less than the pressure in heat reclaiming condenser 30 and discharge line 66 leading thereto, bringing heating circuit 14 into communication with the evaporator as described above lowers the refrigerant pressure in condenser 30 and line 66. This reduces the size of the pressure increase which booster compressor 26 must produce in the refrigerant passing therethrough, reducing the temperature increase which occurs as the refrigerant is compressed by the booster compressor. In this manner, the temperature of vapor discharged from booster compressor 26 is reduced, preventing the vapor from reaching temperatures that may cause the refrigerant to breakdown into potentially damaging components.

When the temperature of the vapor discharged from booster compressor 26 falls below the preset value, thermostatic switch Th.S. closes, re-energizing coil K1 and, thus, opening contacts CR1 in line 10 of Figure 2. Timer relay KT1 in line 11, however, remains energized until it runs for a preset length of time. This time delay enables the heating load which will be placed on circuit 14 when the circuit is reactivated to increase, insuring at least moderate vapor flow through the heating circuit when heating is reactivated. When timer KT1 automatically deactivates, contacts CRT1 in line 12 close, and coil K2 is energized. Vent line valve 44 is thus closed via action of solenoid 46 and contacts CR2 in line 8, and control of motor 40 is returned to switches 76 and 78 due to the closing of contacts CR2 in line 13 and the opening of contacts CR2 in line 14.

As mentioned above, the most desired, complete response of machine 10 to the vapor temperature in heating circuit 14 approaching excessive levels depends upon operating conditions of cooling circuit 12. More particularly, if the load on cooling circuit 12 is relatively high when action of heating circuit 14 is terminated because vapor temperatures therein are approaching excessive values, then preferably operation of the cooling circuit is continued unaffected by the action of the heating circuit. In contrast, if the load on cooling circuit 12 is relatively low as action of heating circuit 14 is terminated, then preferably operation of cooling circuit 12 is simulataneously terminated. It is desirable to terminate action of cooling circuit 12 under these latter conditions because otherwise all of the heat rejected by the refrigerant passing through the cooling circuit would be rejected via primary condenser 20, and it is preferred to temporarily terminate action of the cooling circuit until a later time when this heat can be recovered via heat reclaiming condenser 30.

In view of the above, sensing means is provided for sensing the cooling load or demand on machine 10. In the preferred embodiment illustrated in the drawings, the sensing means includes guide vane switch G.V.S. for sensing the position of guide vanes 34. Guide vane switch G.V.S. is open when the load on cooling circuit 12 is below a predetermined value, closes when guide vanes 34 reach a position indicating that the load on circuit 12 equals the predetermined value, and remains closed as long as the load on the cooling circuit is at or above the predetermined value. Referring to Figure 2, guide vane switch G.V.S. is electrically located in line 3 thereof. If guide vane switch G.V.S. is closed when thermostatic switch Th.S. opens, cooling circuit 12 continues to operate because, despite the opening of contacts CR1 in line 4, current is still conducted through relay coil K4 via guide vane switch G.V.S. in line 3. Since coil K4 remains energized, contacts CR4 in line 16 remain closed and compressor motor 50 remains connected to the source of electrical energy. Thus, machine 10 changes from a "heating and cooling" mode of operation to a "cooling only" mode of operation.

However, if guide vane switch G.V.S. is open when thermostatic switch Th.S. opens, the operation of machine 10, including the action of cooling circuit 12, is temporarily terminated. More particularly, as contacts CR1 in line 4 open in response to the opening of thermostatic switch Th.S. in line 9, if, at the same time, guide vane switch G.V.S. is open, then relay coil K4 in line 3 is disconnected from the electrical energy source and, hence, de-energized. When this happens, contacts CR4 in line 5 close and contacts CR4 in lines 3 and 16 open. The opening of contacts CR4 in line 16 disconnects compressor motor 50 from the source of electrical energy. Compressor 18 is deactivated and operation of machine 10 is terminated. Simultaneously, the closing of contacts CR4 in line 5 energizes Program Timer PT. Program Timer PT continues with its control sequence, and opens switch PT-3 to reset this switch for later restarting the compressor motor. Then switch PT-4 closes to maintain relay coils K3 and KT2 energized despite the possible opening of safety switch Saf.S. Next, switch PT-2 moves to the position shown in full line in Figure 2, deactivating oil pump o.p. and energizing relay timer KT3 via line 7 and closed contacts CRT2 therein. When timer KT3 is energized, contacts CRT3 in line 5 open, deactivating Program Timer PT.

Timer KT3 maintains compressor motor 50 and refrigeration machine 10 inactive for a predetermined length of time to prevent motor 50 and machine 10 from cycling on and off at an undesirably high frequency. Delaying the restart of machine 10 also increases the heating and cooling loads placed thereon when the machine is restarted. In this manner, machine 10 and specifically motor 50 will operate at a higher, more efficient capacity when restarted. When timer KT3 deactivates, contacts CRT3 in line 5 close, energizing Program Timer PT, and the program timer continues with its control sequence. Specifically, Program Timer PT moves switch PT-1 to the position shown in full line in Figure 2. This is the last step in the control sequence of Program Timer PT, and when it is completed, the Program Timer starts to repeat its control sequence. Particularly, switches PT-1 and PT-2 are moved back to the positions shown in broken lines in Figure 2. It should be noted that timer relay KT3 in line 7 is an "interval timer" and, once it deactivates, must be disconnected from the source of electrical energy before it can be reactivated. Thus, timer KT3 does not immediately restart after automatically deactivating despite the fact that at the time the timer deactivates, switch PT-2 is in the position shown in full line and the timer is connected to the electrical energy source. Next, switch PT-4 moves to the open position to insure that compressor motor 50 is not restarted unless safety switch Saf.S. is closed, and then switch PT-3 is closed. Preferably, the dwell time for timer KT3 is greater than the dwell time for timer KT1 in line 11. Hence, when switch PT-3 is closed as a consequence of timer KT3 deactivating, contacts CR1 in line 4 are closed, and the closing of switch PT-3 starts compressor motor 50 as explained above.

As will be apparent to those skilled in the art, valves 38 and 44 may be positioned by means other than electric motor 40 and electric solenoid 46 respectively. For example, hydraulic or pneumatic devices may be employed to position valves 38 and 44. Further, the temperature of vapor discharged from booster compressor 26 may be sensed by means other than a thermostatic switch, for example a thermo-sensitive bulb may be used. Additionally, it should be noted that the heating action of circuit 14 may be terminated in a number of ways other than as specifically described herein. For example, in a machine employing separate drive means to drive primary and booster compressors 16 and 26, the heating action of circuit 14 may be terminated by deactivating the booster compressor drive means.

Machine 10, as described above, effectively terminates heating action of heating circuit 14 when the temperature of vapor therein approaches an undesirable value, preventing this vapor temperature from actually reaching undesirable values. As mentioned earlier, it is often desirable to terminate heating action of circuit 14 for other reasons; for example, if the load on the heating circuit falls below a certain level. With the preferred embodiment of machine 10 illustrated in the drawings, since second stage 26 of compressor 18 is directly coupled to first stage 16 thereof, second stage 26 operates whenever first stage 16 operates irrespective of whether heating action of circuit 14 is terminated. If refrigerant flow through second stage 26 were to be eliminated when heating action of heating circuit 14 has been extinguished, undesirably high temperatures might be reached in the heating circuit and in second stage 26 of compressor 18. Thus, it is necessary to maintain at least a minimum flow of refrigerant through second compressor stage 26 and heating circuit 14 regardless of the heating load thereon.

If this minimum flow of refrigerant vapor through second compressor stage 26 were furnished at the discharge pressure from primary or first compressor stage 16, a substantial weight flow of vapor would be required to maintain the temperatures within the second compressor pressure stage 26 and heating circuit 14 below predetermined maximum levels. Second stage 26 would use a substantial amount of power in further compressing the vapor passing therethrough while producing no useful work. However, if the minimum flow of vapor were supplied to second stage 26 from a relatively low pressure source, the required weight flow of vapor could be reduced producing a concomitant reduction in the wasted power consumption. Further reductions in the power consumption of second stage 26 may be achieved when the heating action of circuit 14 is terminated by decreasing the pressure at the discharge side of this stage, minimizing the lift requirements thereof. Turning now to Figure 3, there is illustrated a modified refrigeration machine 10 which, in accordance with a second aspect of the present invention, is uniquely designed to terminate heating action of circuit 14 when the heating load thereon falls below a predetermined level and, simultaneously, lower the pressure differential across circuit 14 and supply vapor to booster compressor 26 thereof at a much lower than normal supply pressure.

Referring to the embodiment depicted in Figure 3, it should be noted that like reference numbers refer to like or corresponding elements of the embodiment disclosed in Figure 1. The embodiment shown in Figure 3 includes several elements not described above, specifically temperature sensor 76, signal generating line 78, switch 80, and electrical lines L-7 and L-8. Moreover, with the embodiment of Figure 3, valves 38 and 44 are controlled by positioning means 82 and 84 respectively, which preferably are electrically actuated solenoids connected to lines L-7 and L-8.

Temperature sensor 76 senses the temperature of the heat transfer fluid leaving heat reclaiming condenser 30, and when the sensed temperature falls below a predetermined value, indicating that the load on heating circuit 14 has fallen below a preset level, the sensor generates a signal. This signal is transmitted to switch 80 via line 78, closing the switch. This, in turn, actuates positioning means 82 and 84 to close substantially valve 38 and to open valve 44 respectively. When valve 38 is placed in its substantially closed position, it will permit a minimum flow of refrigerant through conduit 64 to the suction side of booster compressor 26. With valve 44 open, a by-pass flow path is established through line 42 about restriction means 22 and 32. Heat reclaiming condenser 30 is thus directly placed in communication with condenser 24 whereby the pressure within condenser 30 is lowered to substantially that of condenser 24. The pressure which booster compressor 26 must exceed to generate flow is thereby substantially reduced. Further, through the substantial closing of valve 38, the pressure of the vapor delivered through line 64 to the suction side of booster compressor 26 is substantially reduced, thereby minimizing the required weight flow of refrigerant for maintaining the temperature of the booster compressor below the preferred maximum level.

By placing the discharge side of the booster compressor 26 at substantially the pressure of condenser 30 and significantly reducing the pressure of the refrigerant vapor flowing through the suction side of the booster compressor, the lift requirements thereof are minimized while the weight flow of the refrigerant required to maintain the temperature of the booster compressor below the desired operating point is reduced, substantially decreasing the consumption of wasted power when the heating load on the refrigeration system has been terminated. In effect, the pressure differential across booster compressor 26 has been substantially equalized, with the pressure being reduced to approximately the lowest level within the refrigeration machine 10.

As an alternate to permitting a minimum flow of refrigerant through valve 38 as described above, valve 38 may be entirely closed upon the opening of valve 44. In this embodiment, a line 86 having a check valve 88 will communicate line 58 with line 64 downstream of valve 38. When valve 38 entirely closes, the pressure in the line downstream thereof will be substantially reduced thereby causing check valve 88 to open to permit refrigerant flow from line 58 to the inlet side of booster compressor 26. With the opening of check valve 88, booster compressor 26 will receive the necessary refrigerant flow for maintaining the booster compressor at a safe operating temperature. The flow of refrigerant through line 86 at substantially the suction pressure of first compressor stage 16 will provide the requisite low pressure refrigerant vapor to the inlet of booster compressor 26. Further, as the temperature of the vapor delivered through conduit 86 is at generally the lowest level within refrigeration machine 10, the operating temperature of booster compressor 26 will be significantly reduced.

While it is apparent that the invention herein disclosed is well calculated to fulfill the objects above stated, it will be appreciated that numerous modifications and embodiments may be devised by those skilled in the art, and it is intended that the appended claims cover all such modifications and embodiments as fall within the true spirit and scope of the present invention.

Claims

1. Apparatus (10) for satisfying heating and cooling demands comprising a cooling circuit (12) for satisfying the cooling demand and including a high pressure side (60) and a low pressure side (62); a heating circuit (14) for satisfying the heating demand and including a booster compressor (26) for drawing and compressing refrigerant vapor from the high pressure side (62) of the cooling circuit (12), and return means (72, 74, 32) for returning refrigerant from the heating circuit (14) to the cooling circuit (12); characterized by a sensor (Th.S.) for sensing the temperature of vapor discharged from the booster compressor (26); and means (38, 40, 42, 44, 46) responsive to the sensor (Th.S.) for terminating the heating action of the heating circuit (14) when the temperature of the vapor discharged from the booster compressor (26) exceeds a preset temperature.

2. The apparatus (10) as defined by claim 1 further characterized by the terminating means includes means (38, 40) for reducing the vapor flow rate through the heating circuit (14); and means (42, 44, 46) for venting vapor in the heating circuit (14) to a low pressure region (62) to lower the pressure of vapor in the heating circuit (14).

3. The apparatus (10) as defined by claim 2 further characterized by the reducing means includes a valve (38) for regulating the flow of vapor through the booster compressor; and positioning means (40) connected to the valve (38) and the sensor (Th.S.) for positioning the valve (38) to decrease the vapor flow rate through the booster compressor (26) when the temperature of the vapor discharged therefrom exceeds the preset temperature.

4. The apparatus (10) as defined by claim 3 further characterized by the valve (38) includes a modulating valve; the positioning means (40) includes a reversible electric motor for modulating the valve (38) between minimum and maximum flow positions; and the

temperature sensor (Th.S.) includes a thermostatic switch for connecting the electric motor (40) to a source of electrical energy to move the valve (38) toward the minimum flow position when the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature.

5. The apparatus (10) as defined by claim 1 further characterized by means (G.V.S.) for sensing the demand on the cooling circuit (12); and means (CR4) for terminating the cooling action of the cooling circuit (12) when both the cooling demand is below a predetermined load and the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature.

6. The apparatus (10) as defined by claim 5 further characterized by the heating action terminating means includes means (42, 44, 46) for venting vapor in the heating circuit (14) to a low pressure region (62) to lower the pressure of vapor in the heating circuit (14); and the cooling action terminating means includes means (CR4) for deactivating a drive means (50) for a compressor (16) of the cooling circuit (12).

7. The apparatus (10) as defined by claim 6 further characterized by the compressor drive means (50) includes an electric motor; the temperature sensor (Th.S.) includes a thermostatic switch; the cooling demand sensor (G.V.S.) includes a limit switch for sensing the position of a guide vane (34) of the compressor (16) of the cooling circuit (12); and the deactivating means (CR4) includes electrical contact means electrically connected to the thermostatic switch (Th.S.), the limit switch(G.V.S.), and the electric motor (50) for disconnecting the motor (50) from an electrical energy source (L-1, L-2) when both the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature and the demand on the cooling circuit (12) is below the predetermined load.

8. The apparatus (10) as defined by claims 2, 3, 4, 6, or 7 further characterized by the venting means includes a vent line (42) for transmitting refrigerant from the heating circuit (14) to the low pressure side (62) of the cooling circuit (12); a vent line valve (44) for regulating the flow of refrigerant through the vent line (42); and means (46) for opening the vent line valve (44) when the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature.

9. The apparatus (10) as defined by claim 8 further characterized by the opening means (46) includes a solenoid.

10. A control for a booster type heat reclaiming refrigeration machine (10) having a cooling circuit (12) for satisfying a cooling demand, a heating circuit (14) for satisfying a heating demand, a vent line (42) for venting refrigerant from the heating circuit (14) to a low pressure area (62), a vent line valve (44) for regulating the flow of refrigerant through the vent line (42), and means (46) for opening the vent line valve (44), the cooling circuit (12) having a primary compressor (16) for drawing vapor from a low pressure side (62) of the cooling circuit (12), compressing the vapor, and discharging the vapor into a high pressure side (60) of the cooling circuit (12), and the heating circuit (14) having a booster compressor (26) for drawing and further compressing vapor from the high pressure side (60) of the cooling circuit (12), a booster valve (38) for regulating the flow of refrigerant through the booster compressor (26), and positioning means (40) for positioning the booster valve (26), the control characterized by a sensor (Th.S.) for sensing the temperature of the vapor discharged from the booster compressor (26); and means (K1, K2, CR1, CR2) for connecting the positioning means (40) and the opening means (46) to the sensor (Th.S.) for operating the positioning means (40) and the opening means (46) to move the booster valve (38) to decrease the vapor flow rate through the booster compressor (26) and to open the vent line

valve (44) and allow refrigerant flow through the vent line (42) when the temperature of the vapor discharged from the booster compressor (26) rises above a preset temperature.

11. The control as defined by claim 10 for use with a refrigeration machine (10) having an electric motor (50) for positioning the booster valve (38) and a solenoid (46) for opening the vent line valve (44), further characterized by the sensor (Th.S.) includes a thermostatic switch in heat transfer relation with vapor discharged from the booster compressor (26); and the connecting means includes electrical contact means (CR2) associated with the thermostatic switch (Th.S.) for connecting the electric motor (40) and the solenoid (46) to an electrical energy source (L-1, L-2, L-3, L-4) when the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature to move the booster valve (38) to decrease the vapor flow rate through the booster compressor (26) and to open the vent line valve (44).

12. A control for a booster type heat reclaiming refrigeration machine (10) having a cooling circuit (12) for satisfying a cooling demand and a heating circuit (14) for satisfying a heating demand, the cooling circuit (12) having a primary compressor (16) for drawing vapor from a low pressure side (62) of the cooling circuit (12); compressing the vapor, and discharging the vapor into a high pressure side (60) of the cooling circuit (12); the heating circuit (14) having a booster compressor (26) for drawing and further compressing vapor from the high pressure side (60) of the cooling circuit (12), a booster valve (38) for regulating the flow of refrigerant through the booster compressor (26), and positioning means (40) for positioning the booster valve (38); the refrigeration machine (10) further having drive means (50) for driving the primary compressor (16), a vent line (42) for venting refrigerant from the heating circuit (14) to a low pressure area (62), a vent line valve (44) for regulating the flow of

refrigerant through the vent line (42), and means (46) for opening the vent line valve (44), the control characterized by a temperature sensor (Th.S.) for sensing the temperature of vapor discharged from the booster compressor (26); a cooling load sensor (G.V.S.) for sensing the demand on the cooling circuit (12); valve regulating means (K1, K2, CR1, CR2) for connecting the positioning means (40) and the opening means (46), to the temperature sensor (Th.S.) to activate the positioning means (40) and the opening means (46) to, respectively, move the booster valve (38) to decrease the vapor flow through the booster compressor (26) and open the vent line valve (44) when the temperature of vapor discharged from the booster compressor (26) exceeds a preset temperature; and drive regulating means (K1, K4, CR1, CR4) for connecting the temperature sensor (Th.S.) and the cooling load sensor (G.V.S.) to the primary compressor drive means (50) to deactivate the drive means (50) when both the temperature of the vapor discharged from the booster compressor (26) exceeds the preset temperature and the cooling demand is below a predetermined load.

13. The control as defined by claim 12 for use with a refrigeration machine (10) having a first electric motor (40) for positioning the booster valve (38), a second electric motor (50) for driving the primary (16) and booster compressors (26); means for connecting the first (40) and second (50) electric motors to a source of electrical energy (L-3, L-4, L-5, L-6), and a solenoid (46) for opening the vent line valve (44), further characterized by the temperature sensor (Th.S.) includes a thermostatic switch; the cooling load sensor (G.V.S.) includes a limit switch for sensing the position of an inlet guide vane (34) of the primary compressor (16); the valve regulating means includes first electrical contact means (CR2) associated with the thermostatic switch (Th.S.) for connecting the solenoid (46) and the first electric motor (40) to the source of electrical energy (L-3, L-4) when the temperature of the vapor discharged from the booster

compressor (26) exceeds the preset temperature; the drive regulating means includes second electrical contact means (CR4) associated with the thermostatic switch (Th.S.) and the limit switch (G.V.S.) for disconnecting the second electric motor (50) from the electrical energy source (L-5, L-6) when both the temperature of vapor discharged from the booster compressor (26) exceeds the preset temperature and the demand on the cooling circuit (12) is below the predetermined load.

14. The control as defined by claim 13 further characterized by first electric timer means (KT1) for maintaining the first electric motor (40) and the solenoid (46) connected to the electrical energy source (L-1, L-2, L-3, L-4) for a first preset length of time; and second electric timer means (KT2) for maintaining the second electric motor (50) disconnected from the electrical energy source (L-5, L-6) for a second preset length of time.

15. A method of controlling the operation of a booster type heat reclaiming refrigeration machine (10) including a cooling circuit (12) having a low pressure side (62) and a high pressure side (60) for satisfying a cooling load, and a heating circuit (14) for satisfying a heating load, the method characterized by the steps of passing refrigerant vapor from the high pressure side (60) of the cooling circuit (12) through the heating circuit (14); compressing refrigerant vapor passing through the heating circuit (14); transferring heat from the refrigerant passing through the heating circuit (14) to a first heat transfer fluid for satisfying the heating load and to condense the refrigerant; and terminating the transferring step when the temperature of the refrigerant passing through the heating circuit (14) exceeds a preset temperature.

16. The method as defined by claim 15 further characterized by the terminating step includes the steps of reducing the vapor flow

rate through the heating circuit (14); and venting vapor from the heating circuit (14) to a low pressure region (62) to lower the pressure in the heating circuit (14).

17. The method as defined by claim 16 further characterized by the steps of increasing the vapor flow rate through the heating circuit (14) when the temperature of the refrigerant passing therethrough falls below the preset temperature; and delaying the increasing step for a predetermined length of time.

18. The method as defined by claim 15 further characterized by the steps of compressing refrigerant vapor passing through the cooling circuit (12); and terminating the steps of compressing refrigerant vapor passing through the heating (14) and cooling (12) circuits when both the temperature of the refrigerant passing through the heating circuit (14) exceeds the preset temperature and the load on the cooling circuit (12) is below a predetermined load.

19. The method as defined by claim 18 further characterized by the step of restarting the steps of compressing refrigerant vapor passing through the heating (14) and cooling (12) circuits a predetermined length of time after the compressing steps are terminated.

20. A control for a booster type heat reclaiming refrigeration machine (10) having a cooling circuit (12) for satisfying a cooling demand, and a heating circuit (14) for satisfying a heating demand, the cooling circuit (12) having a low pressure side (62) and a high pressure side (60), the control characterized by means (64) for maintaining a continuous flow of refrigerant through the heating circuit (14) regardless of changes in the heating load thereon; and refrigerant flow control means (42, 44, 84) responsive to the changes in the heating load on the heating circuit (14) for substantially equalizing the pressure within the

heating circuit (14) when the load thereon decreases below a predetermined level.

21. A control in accordance with claim 20 further characterized by the refrigerant flow control means includes a first conduit (42) communicating refrigerant condensing means (30) of the heat reclaiming circuit with refrigerant evaporator means (24) of the refrigeration machine (10); a first normally closed valve (44) interposed in the first conduit (42) for controlling refrigerant flow from the condensing means (30) to the evaporator means (24); and load sensing means (76) for opening the first normally closed valve (44) when the load on the heating circuit (14) decreases below the predetermined level for enabling refrigerant to flow from the condensing means (30) to the evaporator means (24) for substantially equalizing the pressure therebetween.

22. A control in accordance with claims 20 or 21 further characterized by the flow maintaining means includes means (34, 86, 88) for delivering refrigerant from the low pressure side (62) of the cooling circuit (12) to the heating circuit (14) the suction pressure of the low pressure stage to the inlet when the heating load thereon falls below the predetermined level.

23. A control in accordance with claim 22 further characterized by the refrigerant delivering means includes a second conduit (86) connecting the low pressure side (62) of the cooling circuit (12) with the heating circuit (14); a normally closed second valve (88) interposed in the second conduit (86) for controlling flow of refrigerant therethrough; and means (38) for opening the normally closed second valve (88) upon the opening of the first normally closed valve (44).

24. A control for a booster type heat reclaiming refrigeration machine (10) having a cooling circuit (12) for satisfying a cooling demand, and a heating circuit for satisfying a heating

demand, the cooling circuit (14) having a low pressure side (62) and a high pressure side (60), the control characterized by means (64) for maintaining a continuous flow of refrigerant through the heating circuit (14) regardless of changes in the heating load thereon; refrigerant flow control means (86, 88, 38) responsive to changes in the heating load on the heating circuit (14) including pressure reducing means (86, 88, 38) for reducing the pressure of refrigerant delivered to the heating circuit (14); pressure equalizing means (42, 44, 84) for substantially equalizing the pressure between the heating circuit (14) and the low pressure side (62) of the cooling circuit (12); and actuating means (76, 78, 80) for simultaneously activiating the pressure reducing means (86, 88, 38) and the pressure equalizing means (42, 44, 84) when the load on the heating circuit (14) decreases below a predetermined level.

25. A control in accordance with claim 24 further characterized by the pressure reducing means includes means (86, 88, 38) for delivering refrigerant from the low pressure side (62) of the cooling circuit (12) to the heating circuit (14) when the heating load thereon falls below the predetermined level.

26. A control in accordance with claim 25 further characterized by the refrigerant delivering means includes a conduit (86) connecting the low pressure side (62) of the cooling circuit (12) with the heating circuit (14); a normally closed valve (88) interposed in the conduit (86) for controlling flow of refrigerant therethrough; and means (38) for opening the normally closed valve (88) upon activation of the pressure equalizing means (42, 44, 84).

27. A method of controlling operation of a refrigeration machine (10) of the type utilizing relatively low pressure refrigerant discharged from a low pressure stage (16) of a multi-stage centrifugal compressor (18) to satisfy a cooling load and

relatively high pressure refrigerant discharged from a high pressure stage (26) of the compressor (16) to satisfy a heating load, the method characterized by the steps of monitoring the load on the high pressure stage (26); maintaining a continuous flow of refrigerant through the high pressure stage (26) regardless of changes in the load thereon; substantially equalizing the pressure between the inlet and discharge sides of the high pressure stage (26) when the load thereon falls below a predetermined level; and simultaneously decreasing the pressure at the inlet side of the high pressure stage (26) for reducing the weight flow of refrigerant therethrough.

28. A method in accordance with claim 27 further characterized by the equalizing step includes placing the discharge side of the high pressure stage (26) in communication with the inlet side of the low pressure stage (16) for substantially decreasing the pressure of the discharge side of the high pressure stage (26).

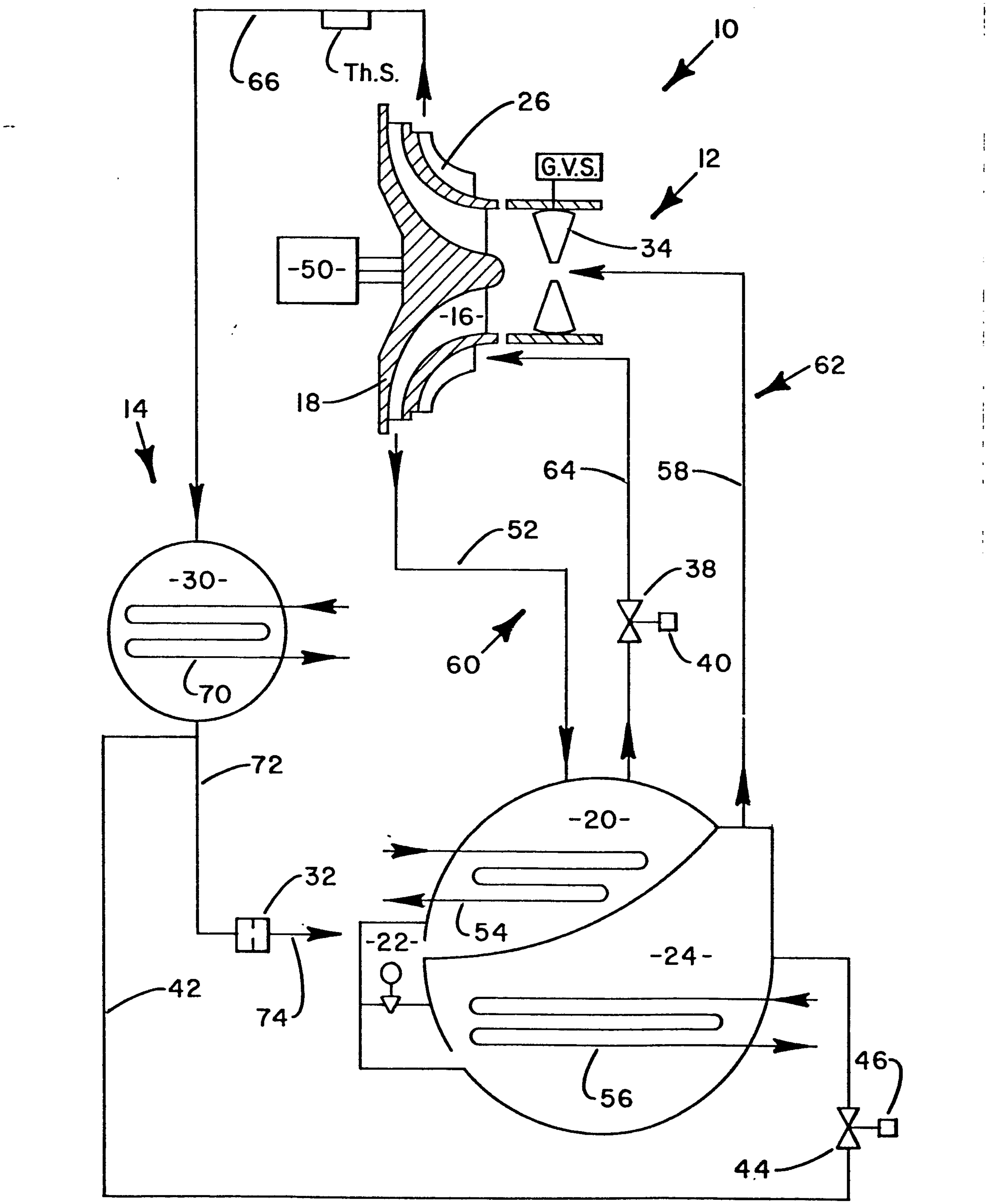
29. A method in accordance with claim 28 further characterized by the step of communicating the inlet sides of the low pressure (16) and high pressure (26) stages for delivering refrigerant gas at the pressure of the inlet side of the low pressure stage (16) to the inlet side of the high pressure stage (26) when the load on the high pressure stage (26) falls below the predetermined level.

30. A control for a multi-stage centrifugal compression refrigeration machine (10) having a first condensor (20) for receiving relatively low pressure refrigerant discharged from a low pressure stage (16) of the centrifugal compression refrigeration unit (10), a second condensor (30) for receiving relatively high pressure refrigerant discharged from a high pressure stage (26) of the centrifugal compression refrigeration unit (10) to satisfy a heating load, an evaporator (24) for receiving condensed refrigerant from the first (20) and second (30) condensors to satisfy a cooling load, and a first conduit

including expansion means (22) for defining a first refrigerant flow path from the first (20) and second (30) condensors to the evaporator, the control characterized by a first by-pass conduit (42) including a first normally closed valve (44) for defining a second refrigerant flow path from the second condensor (30) to the evaporator (24); and actuating means (76, 78, 80, 84) responsive to the heating load on the second condensor (30) for opening the normally closed valve (44) when the heating load decreases below a predetermined level for enabling refrigerant to flow from the second condenser (30) to the evaporator through the by-pass conduit (42).

31. A control in accordance with claim 30 further characterized by means (86, 88, 34) for delivering relatively low pressure refrigerant to the inlet side of the high pressure stage (26) of the refrigeration unit (10) when the heating load falls below the predetermined level.

32. A control in accordance with claim 31 further characterized by the delivering means includes a second conduit (86) connecting the suction side of the low pressure stage (16) with the suction side of the high pressure stage (26); a normally closed second valve (88) interposed in the second conduit (86) for controlling flow of refrigerant therethrough; and means (38) for opening the second normally closed valve (88) upon the opening of the first normally closed valve (44) for enabling refrigerant gas at the suction pressure of the low pressure stage (16) to flow to the suction side of the high pressure stage (26).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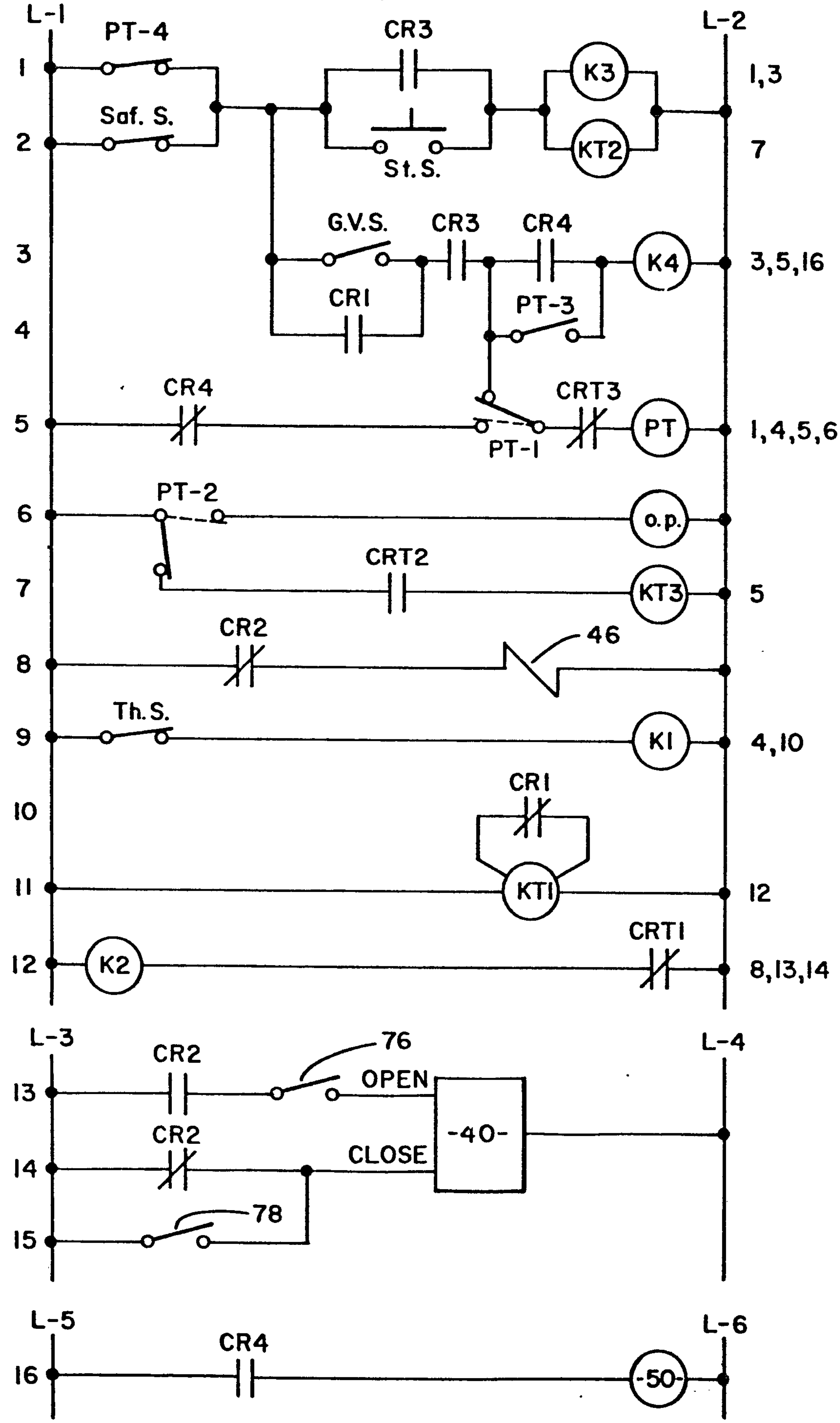
Th.S.
G.V.S.

FIG. 1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K4
3,5,16
CR1
PT-3
CR4
CRT3
PT
PT-1
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PT-2
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CRT2
KT3
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CR2
46
Th.S.
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4,10
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12
K2
CRT1
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L-3
L-4
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L-5
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FIG. 2

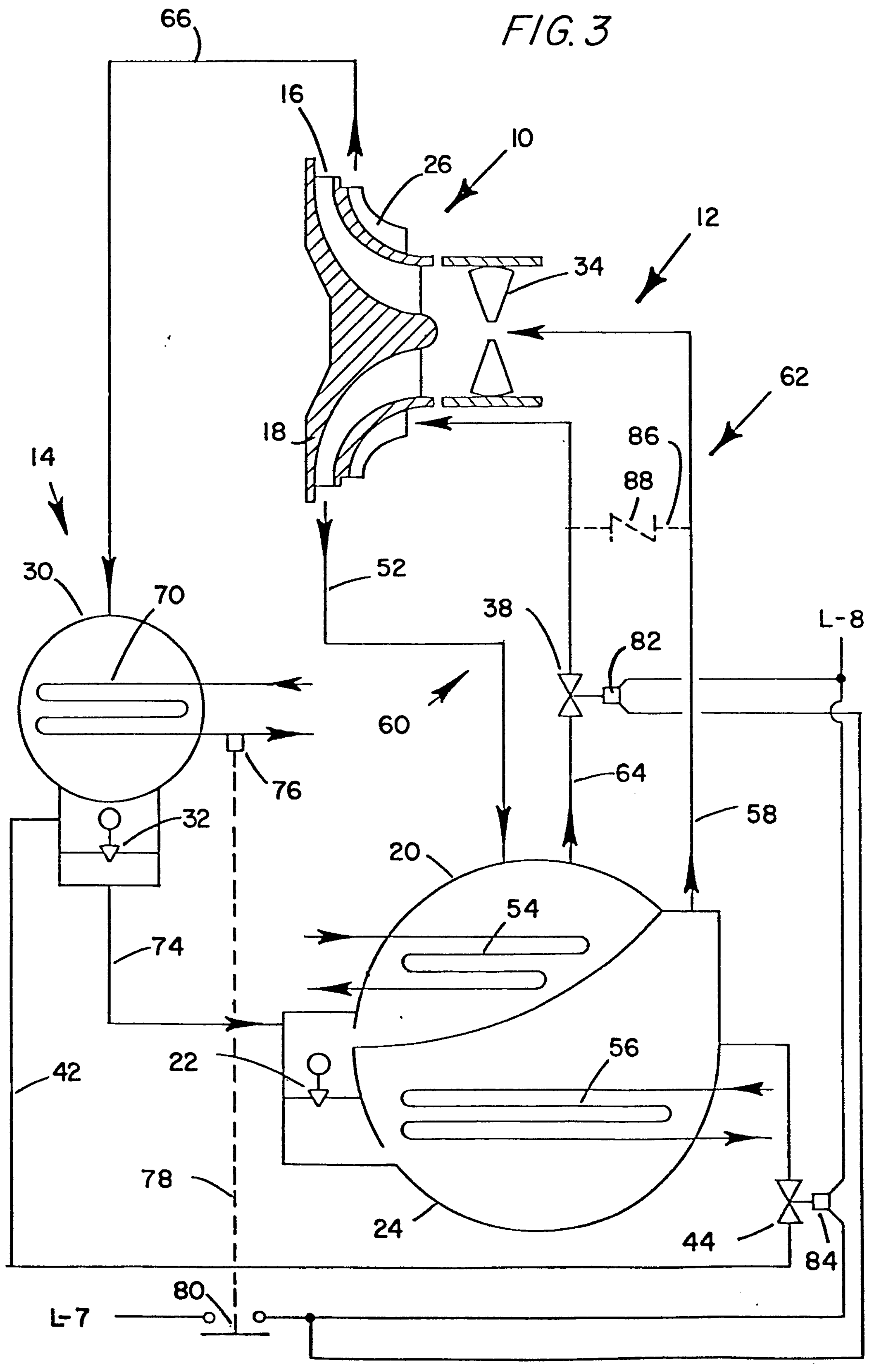
FIG. 3
66
16
10
26
12
34
62
18
86
88
14
30
70
52
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82
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64
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74
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80
L-7

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European Patent Office

EUROPEAN SEARCH REPORT

Application number
EP 80 10 6074

Category	DOCUMENTS CONSIDERED TO BE RELEVANT Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.³)
	US - A - 3 859 815 (K. KASAHARA) * Column 1, line 51 - column 2, line 55; figure 1 * --	1	F 25 B 49/00 29/00 G 05 D 23/19
	US - A - 3 665 724 (C. ANDERSON) * Column 2, line 3 - column 3, line 67; figure * --	2,5, 12,15, 20	
	US - A - 3 635 041 (J. ENDRESS) * Column 1, lines 41-63; column 1, line 71 - column 3, line 56; figure * --	2,5, 12	TECHNICAL FIELDS SEARCHED (Int. Cl.³) F 25 B G 05 B
	GB - A - 2 017 345 (CARRIER) * Page 2, lines 23-93; figure 2 * --	4	
	US - A - 3 522 711 (W. SHAUGHNESSY) * Column 8, lines 21-36; figure * --	5	
	US - A - 3 700 914 (G. GRANIERI) * Abstract; figure 1 * --	14	CATEGORY OF CITED DOCUMENTS X: particularly relevant A: technological background O: non-written disclosure P: intermediate document T: theory or principle underlying the invention E: conflicting application D: document cited in the application L: citation for other reasons
A	GB - A - 2 003 264 (CARRIER) * Page 2, line 60 - page 3, line 21; page 5, lines 21-76; figures 1,2,5 * --	1	
A	US - A - 2 888 809 (S. RACHFAL) ./.	1	&: member of the same patent family, corresponding document

X The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
The Hague	28-01-1981	V. HELOT

EPO Form 1503.1 06.78

European Patent Office

EUROPEAN SEARCH REPORT

Application number

EP 80 10 6074

-2-

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim
	* Column 2, line 13 - column 3, line 17; figure *	
	--	
A	US - A - 3 011 322 (E. TANZBERGER)	1
	* Column 1, line 61 - column 3, line 29; figures 1-3 *	

CLASSIFICATION OF THE APPLICATION (Int. Cl. 3)

TECHNICAL FIELDS SEARCHED (Int. Cl. 3)