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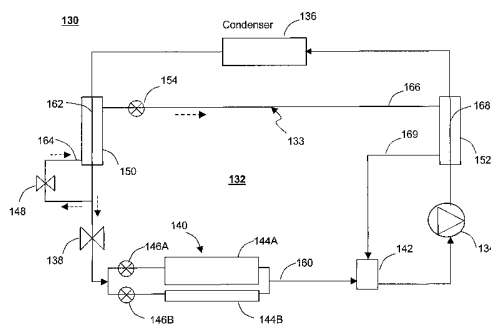
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(54) Title: MULTIZONE AIR-CONDITIONING SYSTEM WITH A SINGLE FREQUENCY COMPRESSOR



(57) **Abstract:** A variable capacity refrigeration system employing a constant speed compressor that operates continuously irrespec-
tive of heat load, and a refrigerant bypass path including a secondary expansion device, two heat exchangers, a pressure differential
accommodation device, and a flow control device to divert permit a portion of the refrigerant exiting the condenser to the bypass path.
One heat exchanger is connected so that the refrigerant exiting the condenser passes through a first flow path. At the exit of the first
flow path, a portion of the refrigerant is diverted through a secondary expansion device to the bypass path which then passes through
a second flow path of the first heat exchanger, whereby the first heat exchanger provides subcooling of the refrigerant exiting the
condenser. Refrigerant exiting the compressor passes through a first flow path of the second heat exchanger. The refrigerant exiting
the second flow path of the first heat exchanger passes through a second flow path of the second heat exchanger, and then returns
to the compressor inlet. The second heat exchanger therefore provides desuperheating of the refrigerant in the primary refrigerant loop.
This construction permits use in a multizone system where as much as 70% of the refrigerant mass flow can be diverted to the bypass
path to reduce cooling capacity while still permitting full vaporization of the diverted refrigerant before return to the compressor. An
accumulator may also be provided in the bypass path.

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MULTIZONE AIR-CONDITIONING SYSTEM
WITH A SINGLE FREQUENCY COMPRESSOR

CROSS-REFERENCE TO PRIOR APPLICATION

This application claims priority to U.S. Provisional Application Serial No. 60/426,073, filed November 11, 2002.

FIELD OF THE INVENTION

The present invention relates generally to multizone air-conditioning systems and, more specifically, to such an air-conditioning systems utilizing single frequency compressors for variable thermal loads.

BACKGROUND OF THE INVENTION

Fig. 1 is a block diagram of a conventional refrigeration system, generally denoted at 10. The system includes a compressor 12, a condenser 14, an expansion device 16 and an evaporator 18. The various components are connected together via copper tubing such as indicated at 20 to form a closed loop system through which a refrigerant such as R 12, R 22, R 134a, R 407c, R 410a, ammonia, carbon dioxide or natural gas is cycled.

The main steps in the refrigeration cycle are compression of the refrigerant by compressor 12, heat extraction from the refrigerant to the environment by condenser 14, throttling of the refrigerant in the expansion device 16, and heat absorption by the refrigerant from the space being cooled in evaporator 18. This process, sometimes referred to as a vapor compression refrigeration cycle, is used in air conditioning systems which

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cool and dehumidify air in a living space, or vehicle (e.g., automobile, airplane, train, etc.), in refrigerators and in heat pumps.

Fig. 2 shows the temperature entropy curve for the vapor compression refrigeration cycle illustrated in Fig. 1. The refrigerant exits evaporator 18 as a superheated vapor at evaporator pressure (Point 1), and is compressed by compressor 12 to a very high pressure. The temperature of the refrigerant also increases during compression, and it leaves the compressor as superheated vapor at condenser pressure (Point 2).

A typical condenser comprises a single conduit formed into a serpentine shape with a plurality of rows of the conduit lying in a spaced parallel relationship. Metal fins or other structures which provide high heat conductivity are usually attached to the serpentine conduit to maximize the transfer of heat between the refrigerant passing through the condenser and the ambient air. As the superheated refrigerant gives up heat in the upstream portion of the condenser, the superheated vapor becomes a saturated vapor (Point 2a), and after losing further heat as it travels through the remainder of condenser 14, the refrigerant exits as subcooled liquid (Point 3).

As the subcooled liquid refrigerant passes through expansion device 16, its pressure is reduced, and it becomes a liquid vapor mixture comprised of approximately 20% vapor and 80% liquid. Also, its temperature drops below the temperature of the ambient air as it goes through the expansion device (Point 4 in Fig. 2).

Evaporator 18 physically resembles the serpentine conduit of the condenser. Air to be cooled is exposed to the surface of the evaporator where heat is transferred to the refrigerant. As the refrigerant absorbs heat in evaporator 18, it becomes a superheated

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vapor at the suction pressure of the compressor and reenters the compressor thereby completing the cycle (Point 1 in Fig. 2).

One of the challenges in the design and operation of an air conditioning or refrigeration system is the variation of thermal load over time. There are two types of variations. First, the variations in the thermal load result as the outdoor weather conditions change. One method to cope with such a variation in thermal load is to simply turn ON and OFF the system according to the need for cooling. The system becomes tremendously inefficient if it is repeatedly turned ON and OFF because there is a significant energy loss associated with the start up of a compressor. Second, the thermal load is significantly lower at night as the need to cool the whole house disappears. When the cooling need for bedrooms is substantially smaller than the whole house cooling need, the reduction in the thermal load is often more than 60%. In other words, the cooling capacity should decrease by 60-70% for efficient operation.

One way to avoid the frequent ON OFF operations, is to use an inverter compressor, as illustrated in Fig. 3. The system, generally denoted at 10A, is the same as illustrated in Fig. 1, except for compressor 12A which is essentially a variable speed compressor. Instead of cycling compressor 12A on and off, the frequency is varied depending on the required thermal load. However, the inverter compressor is not a solution for the second problem mentioned above as it cannot handle the 60-70% reduction in the thermal load.

Figures 4 and 5 show typical performance curves of heat absorption at the evaporator and EER (energy efficiency ratio) versus frequency. (As known by those skilled in this art, EER is defined as the ratio of the cooling capacity to compressor work.)

Figure 4 demonstrates the benefit of the inverter type compressor, which provides 17% more cooling capacity when the frequency increases from the base frequency of 60 Hz to 80 Hz. Furthermore, the cooling capacity decreases by 40% when the frequency decreases from 60 to 30 Hz, an excellent performance relative to thermal load variation.

However, the additional cooling capacity of 17% at 80 Hz has its price: there is a severe penalty in the form of a reduced efficiency. As depicted in Fig. 5, there is 18% drop in the EER when the frequency increased from 60 to 80 Hz. Furthermore, the cost of an inverter compressor often represents one third of the total cost of an air conditioning or refrigeration system, which is almost prohibitively expensive for many applications such as room air-conditioners.

Also, the inverter compressor cannot be run at a low enough frequency to achieve a 60-70% reduction in cooling capacity. Thus, a need clearly exists for a way to handle the variations in the thermal load over time without the cost and other disadvantages of the inverter-type compressor.

Fig. 6 shows an air-conditioning system 10B which uses a variable-gap scroll compressor 12B. As will be recognized by those skilled in the art, compressor 12B uses two identical, concentric scrolls, one inserted within the other. One scroll is stationary whereas the other orbits around it. This scroll movement pushes gas through successively smaller spaces formed by the scroll's rotation, until the gas obtains maximum pressure at the center of a compression chamber. Then the compressed gas exits through a discharge port in the fixed scroll.

To reduce the cooling capacity of air-conditioning system 10B, the stationary scroll is moved slightly away from the orbiting scroll to accommodate the reduced

compressor load corresponding to the reduced cooling load. Again, however, cost is a limiting factor: the cost of a variable-gap scroll compressor is significantly more (approximately four times) than a single-frequency compressor. Hence, the variable-gap scroll compressor is not a satisfactory way to vary the cooling capacity of an air-conditioning system.

Alternatively, a high efficiency air conditioner with two compressors of different capacities has been suggested by Hwang, Choe, Kim, and Chung, in *The Development of High Efficiency Air Conditioner with Two Compressors of Different Capacities*, Purdue Compressor Engineering and Refrigeration and Air Conditioning Conferences at Purdue University, West Lafayette, IN 2002. Such a two-compressor system is illustrated at 10C in Fig. 7 herein.

System 10C differs from systems 10, 10A, and 10B in that it employs two compressors 12C-1 and 12C-2, and two check valves 13C-1 and 13C-2. Compressor 12C-1, has a high capacity, and compressor 12C-2 has a low capacity. Both compressors are run for a full cooling load. When the cooling load substantially decreases at night, large compressor 12C-1 is turned off, and only the small one 12C-2 is operated.

This approach is known to have a problem associated with the mal-distribution of oil. In order to prevent the mal-distribution of oil between the two compressors, a common accumulator 15 is needed. Another disadvantage of system 10C is that use of two compressors plus two check valves substantially increases the cost of the air-conditioning system (at least 2 times) compared to a single-compressor system.

Commonly owned U.S. Patent 6,662,576, dated December 16, 2003, for "Refrigeration System With De-Superheating Bypass," the disclosure of which is hereby

incorporated by reference in its entirety herein, discloses a zoned air conditioning system, which is shown in Fig. 8, and generally denoted at 90.

Refrigeration system 90 includes a primary refrigeration loop 91 and a refrigerant bypass path 92. Primary loop 91 includes a compressor 12, a condenser 14, a primary expansion device 16, and a multi-zone evaporator sub-system 96, all of which may be of any conventional or desired type as known to those skilled in the art. Evaporator sub-system 96 includes a plurality of parallel-connected evaporator units, two of which are shown at 98A and 98B, located as required in the space being cooled, and respective associated flow control valves 100A and 100B, by which the evaporator units are connected to main expansion device 16. These, too, may be of any conventional or desired type.

Bypass path 92 is comprised of a secondary expansion device 94 connected to the outlet of compressor 14 by an adjustable flow control valve 95, and a heat exchanger 97 having a first flow path connecting the outlet of compressor 12 to the inlet of condenser 14, and a second flow path connected to the outlet of secondary expansion device 94. The common return for primary loop 91 and bypass path 92 to compressor 12 is provided by a pressure differential accommodating device (PDAD) 38, the latter being employed because the refrigerant pressure in bypass line 92 at the outlet of heat exchanger 97 is greater than the pressure at the common outlet 104 of evaporator units 98a and 98b.

PDAD 38 can be either a vacuum-generating device such as a vortex generator as shown in U.S. Patent 6,250,086, the disclosure of which is hereby incorporated in its entirety herein by reference, or any other desired or suitable device which relies on geometry and fluid dynamics to create a vacuum such as a venturi tube which permits

pressure equalization and mixing of the vapors before return to the compressor inlet.

Alternatively, a pressure reducing device such as a capillary tube, a restricted orifice, a valve, or a porous plug may be employed. In any case, it will be appreciated that whatever type of PDAD is employed, it will function to reduce the pressure of the refrigerant stream exiting the bypass path to match the evaporator outlet pressure.

In operation, when cooling in both zones is required, valves 100a and 100b are opened, and refrigerant flows through both evaporator units 98a and 98b. Valve 95 is adjusted to divert between 10 and 15 percent of the refrigerant from condenser 14 into bypass path 92 to achieve maximum cooling and efficiency.

As an additional feature of system 90, however, if cooling is required, e.g., only in the zone served by evaporator unit 98a, valve 100a is opened, valve 100b is closed, and valve 95 is adjusted to divert the refrigerant which would otherwise flow through evaporator 98b into bypass path 92, along with the refrigerant required for de-superheating. To vary the bypass mass flow rate, valve 95 in bypass line 92 is made continuously adjustable or adjustable in steps, to provide a desired number of different flow rates. For example, 10% diversion could be provided for maximum performance, with 20%, 30%, and 40% diversion for reduced cooling capacity.

PCT application No. PCT/US03/36424, filed November 12, WO 2004/044503, for "Refrigeration System With Bypass Subcooling and Component Size Optimization", the disclosure of which is hereby incorporated in its entirety herein, discloses another zoned air conditioning system, which is shown herein in Fig. 9, and generally denoted at 110. This system is similar to system 90 shown in Fig. 8. except that bypass path 92A is differently configured.

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In particular, an adjustable control valve 95A is located on the downstream side of a first flow path through heat exchanger 97A, and secondary expansion device 94A connects the outlet of condenser 24 directly to the inlet of the first flow path of heat exchanger 97A. The second flow path for heat exchanger 97A is located between the outlet of condenser 24 and the inlet of expansion device 16, rather than between the compressor and the condenser.

In the operation of multizone air-conditioning systems such as shown in Figs. 8 and 9, for residential cooling, the maximum cooling capacity is used to cool the whole house during daytime. At night, the cooling capacity significantly decreases as the whole house space does not require cooling, but only bed rooms need to be cooled at much reduced rates. Thus, as previously mentioned, this can reduce the cooling load by 60-70%¹ from the maximum cooling load, and the bypass rate should be 60-70% based on mass flow rate.

The bypass methods described above cannot handle such high bypass rates because there will not be enough heat available to vaporize all the liquid refrigerant which would have to flow in the bypass path. It is important that the refrigerant entering the compressor from the bypass path be fully vaporized. If there are any liquid droplets left at the end of the bypass path, the compressor will be damaged. While these systems have been the best known ways to provide variable cooling capacity in multizone systems, they simply cannot handle such a huge quantity of liquid refrigerant. Accordingly there is still a need to have a method and apparatus whereby a large quantity of refrigerant (i.e., 60-70% of the

total mass flow) can be bypassed without allowing any liquid refrigerant to reach the compressor.

SUMMARY OF THE INVENTION

It is accordingly among the objects of this invention to provide an improved multizone air conditioning system having variable cooling capacity.,

It is another object of the invention to provide a multizone, variable capacity air conditioning system employing a single speed compressor which is run continuously irrespective of the required cooling capacity.

It is a further object of the invention to provide an air-conditioning system using a conventional single-speed compressor which provides the variable cooling capacity of systems using inverter compressors, variable-gap scroll compressors, or multiple compressors without the disadvantages of such systems.

A more general object of the invention is to provide an air-conditioning system using a conventional single-speed compressor which can be operated continuously irrespective of cooling load.

Another general object of the invention is to provide a variable cooling capacity refrigeration system which does not rely on costly inverter compressors or variable-gap scroll compressors, or on multiple compressors.

A further object of the invention is to provide a variable capacity air conditioning system including a refrigerant bypass path having the capability to selectively provide subcooling and desuperheating.

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It is also an object of the invention to provide a method of operating an air-conditioning system having a single-speed compressor which provides variable cooling capacity without the disadvantages of known variable cooling capacity systems; and

Yet a further object of the invention is to provide a method of operating a multizone air-conditioning system in which a single speed compressor can be run continuously but which provides variable cooling capacity.

The above-stated objects are achieved according to the present invention in an air conditioning system having a primary refrigeration path and a bypass path wherein the bypass path includes two heat exchangers, one for subcooling and the other for desuperheating. The principles of the invention are applicable to both single zone systems such as room air conditioners, or multizone systems suitable for cooling larger spaces.

According to a first aspect of the invention, there is provided a variable capacity refrigeration system having condenser means, expansion means, evaporator means and a refrigerant compressor means that operates continuously at a fixed speed when the system is energized, irrespective of the cooling load, a refrigerant bypass path that includes secondary expansion means, first and second heat exchanger means, and flow control means. When the cooling load is below a predetermined high cooling load threshold, the flow control means permits a portion of the refrigerant exiting from the condenser means to flow through the bypass path to the first and second heat exchanger means, whereby the two heat exchanger means operate as additional evaporator means. When the cooling load is not below the high cooling load threshold, the flow control means prevents refrigerant exiting the condenser means from flowing through the bypass path to the compressor means.

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According to a second aspect of the invention, there is provided a variable capacity refrigeration system having a condenser, an expansion device, an evaporator and a compressor that operates continuously at a fixed speed when the system is energized, irrespective of the cooling load, and a refrigerant bypass path that includes a secondary expansion device, first and second heat exchangers, and a flow control device. When the cooling load is above a predetermined high cooling load threshold, the flow control device is operative to shut off the flow of refrigerant to the bypass path, to thereby provide high cooling capacity.

Optionally, a minimum mass flow of refrigerant may be diverted through the bypass path at all times to provide additional improvements in operating efficiency. The bypass may then be progressively be increased from the minimum level when the cooling load drops below a selected high cooling load threshold. When operated in this manner, as a preferred example, between 5% and 15% of the mass flow of refrigerant may be diverted to the bypass path even under maximum thermal load. When the cooling load is not above the selected threshold, the cooling capacity is reduced by diverting more refrigerant to the bypass path.

According to a third aspect of the invention, there is provided a multizone system having a primary refrigerant loop which may be of any conventional or desired design including spaced multiple parallel-connected evaporator units selectably connectable to the outlet of an expansion valve, and a refrigerant bypass path which provides reduced cooling capacity as well as subcooling and desuperheating for improved overall operating efficiency and to assure that the refrigerant diverted through the bypass path is fully

vaporized before returned to the compressor. This protects the compressor from damage due to inflow of liquid refrigerant.

In a preferred embodiment, the reduced cooling capacity is obtained by diverting a portion of the refrigerant exiting from the condenser. The condenser outlet is connected to the inlet of the high heat flow path of a first heat exchanger. At the outlet of this flow path, a control valve diverts the desired quantity of refrigerant into the bypass path, where it passes through a secondary expansion device thereby lowering its pressure and temperature. The outlet of the secondary expansion device is connected to the inlet of the low heat flow path of the first heat exchanger. The resulting heat transfer provides additional subcooling for the primary refrigerant loop.

The high heat flow path of a second heat exchanger is connected between the outlet of the compressor and the inlet of the condenser, and the refrigerant exiting the low heat flow path of the first heat exchanger flows through the low heat flow path of the second heat exchanger. The resulting heat transfer in the second heat exchanger provides desuperheating for the primary refrigerant loop. The refrigerant exiting the low heat flow path of the second heat exchanger is then returned to the compressor inlet, along the refrigerant exiting the evaporator system. The aggregate heat transfer to the refrigerant in the bypass path is sufficient to provide the desired complete vaporization.

Also, when the heat exchanger is operating to reduce the cooling capacity, refrigerant pressure in the heat exchanger may be maintained at a higher level than the pressure in the primary evaporator as the expansion of the liquid refrigerant is not fully completed at the secondary expansion device. In that case, a pressure differential accommodating device (PDAD) may be used to reduce the pressure of the refrigerant

exiting the bypass path. The PDAD may be a vacuum generator such as a vortex generator or a venturi tube, or a flow restrictor such as a capillary tube. If the system is run with no pressure differential between the primary evaporator and the heat exchanger, a PDAD does not have to be used.

In a variation of the above, according to a second preferred embodiment, an accumulator is provided at the upstream end of the bypass path. In its simplest form, the accumulator may be an enlarged diameter portion of the connecting conduit which forms the bypass path.

The refrigerant circulated in the system may consist of a single component, or alternatively, may be a mixed-refrigerant comprising a plurality of components selected to provide a desired combination of thermal and flammability characteristics.

Other features and advantages of the invention will become apparent from the following description of the invention which refers to the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 shows a block diagram of a conventional air-conditioning system.

Figure 2 shows a temperature entropy curve for the conventional air-conditioning system of Figure 1.

Figure 3 shows a block diagram of an air-conditioning system with a conventional inverter compressor.

Figure 4 shows a typical performance curve of a heat absorption vs. frequency for a conventional inverter compressor.

Figure 5 shows a typical performance curve of an EER vs. frequency for a conventional inverter compressor.

Figure 6 shows a block diagram of an air-conditioning system with a variable-gap scroll compressor.

Figure 7 shows a block diagram of an air-conditioning system with two compressors and two check valves using a common accumulator.

Figure 8 is a block diagram showing an application of the bypass to a zoned cooling system.

Figure 9 is a block diagram showing another application of the bypass to a zoned cooling system.

Figure 10 is a block diagram showing the application of the present invention using two heat exchangers in the bypass path to a zoned cooling system.

Figure 11 is a block diagram showing the application of the present invention using a part of the bypass path as in-line accumulator for a multizone cooling system.

Throughout the drawings, like parts are given the same reference numerals.

DETAILED DESCRIPTION OF EMBODIMENTS OF THE INVENTION

Fig. 10 shows a first embodiment of the present invention, generally denoted at 130, which uses two heat exchangers in the bypass path. Like system 90 shown in Fig. 8, system 130 includes a primary refrigeration loop 132 and a refrigerant bypass path 133. Primary loop 132 includes a compressor 134, a condenser 136, a primary expansion device 138, and a multi-zone evaporator sub-system 140, all of which may be of any conventional or desired type as known to those skilled in the art. The refrigerant circulated in the

system may consist of a single component, or alternatively, may be a mixed-refrigerant comprising a plurality of components selected to provide a desired combination of thermal and flammability characteristics.

Evaporator sub-system 140 includes a plurality of parallel-connected evaporator units, two of which are shown at 144A and 144B, located as required in the space being cooled, and respective associated flow control valves 146A and 146B, by which the evaporator units are connected to primary expansion device 138. Again, these elements can be of any conventional or desired type.

Bypass path 133 includes of a secondary expansion valve 148, first and second heat exchangers 150 and 152, and a flow control valve 154. Heat exchangers 150 and 152 are shown schematically as shell-and-tube units, but it should be understood that any conventional or desired design suitable for the purposes described may be employed.

The tube flow path 162 of heat exchanger 150 is employed as the high temperature conduit. This is coupled between the outlet of condenser 136 and the inlet of primary expansion device 138. The shell flow path 164 serves as the low temperature conduit, and is coupled through secondary expansion device 148 to the outlet of tube flow path 162 whereby a portion of the refrigerant exiting condenser 136 is diverted to bypass path 133 after passing through the heat exchanger.

The outlet end of shell flow path 164 is connected through flow control valve 154 to the inlet end 166 of the shell flow path of second heat exchanger 152, which serves as the low temperature conduit. The outlet end 169 of the shell flow path of heat exchanger 152 is connected to a high pressure inlet of PDAD 142. The tube flow path 168 of heat

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exchanger 152 serves as the high temperature conduit, and is connected between the outlet of compressor 134 to the inlet of condenser 136.

As the refrigerant pressure at the common outlet 160 of evaporator subsystem 140 is lower than the pressure in bypass path 133, outlet 160 is connected to a low pressure inlet of PDAD 142. The construction of PDAD 142 may be the same as described above in connection with PDAD 38 shown in Fig. 8. As will be understood, if there is no pressure differential, PDAD 142 is not required.

The construction illustrated in Fig. 10 allows adjustment of the cooling capacity to match a varying thermal load by controlling the circulating mass flow rate of the refrigerant through evaporator subsystem 140. Obviously, one way to reduce the cooling capacity is to close main expansion device 138 as much as possible. However, in order to close the main expansion valve 138, it is necessary to provide sufficient subcooling of the refrigerant exiting condenser 136. If sufficient subcooling is not provided, a liquid-vapor mixture may enter main expansion valve 138, and the system will cease to provide cooling.

With the present invention, however, the subcooling from heat exchanger 150 is sufficient to completely liquefy the refrigerant entering the main expansion valve 138 and accordingly the system becomes stable even when the main expansion valve is closed as much as possible.

The necessary subcooling is provided in heat exchanger 150 by diverting a portion of the liquid refrigerant exiting the first heat exchanger to secondary expansion device 148. As the liquid refrigerant is throttled through secondary expansion device 148, it become a cold refrigerant mixture at a low pressure. The secondary expansion device 148

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is selected to maintain the pressure of the refrigerant in the bypass path at a value slightly greater than the evaporator pressure.

As previously mentioned, when the required cooling capacity is at its minimum, as much as 70% of the refrigerant mass flow rate should be diverted to bypass path 133. It is found, however, that when the mass flow of diverted refrigerant is over 50% in a multizone variable capacity system, a very large heat transfer capacity is required to convert the liquid refrigerant in the bypass path to superheated vapor as required for the safe operation of the compressor. Achieving this in conventional systems is very difficult or even impossible.

According to the present invention, this is achieved by use of the two heat exchangers. The refrigerant in the bypass path first throttles through the secondary expansion device 148, resulting in mixture (mostly made up of liquid refrigerant), which enters the first heat exchanger. In the first exchanger, the refrigerant in the bypass path absorbs heat and the liquid component is reduced. The first heat exchanger is positioned between the condenser exit and the inlet of the main expansion device and produces the subcooling of the liquid refrigerant flowing in the main loop. The benefits of the subcooling have been described in WO 2004/044503, mentioned above, and in PCT/US04/05721, filed February 25, 2004 for "Refrigeration System Having an Integrated Bypass Device, the disclosure of which is also hereby incorporated by reference herein in its entirety.

The refrigerant mixture exiting heat exchanger 150 passes through heat exchanger 152 and gains heat by removing heat from the hot discharge vapor exiting the compressor. Thus, the refrigerant mixture in bypass path 133 becomes superheated after passing the

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second heat exchanger. This assures that the refrigerant is fully vaporized, as required to protect the compressor, and reduces the temperature of the refrigerant discharged from compressor 134 before it reaches condenser 136.

Reducing the compressor discharge temperature has the added benefit improving the overall performance of the system. Also, since the temperature of the discharge vapor from compressor 134 is much greater than the refrigerant mixture in bypass path 133, there will be efficient heat transfer in heat exchanger 152, producing the superheated vapor at the end of the bypass path.

In the present invention, the refrigerant in the bypass path passes through two heat exchangers positioned in series, collecting enough heat energy so that the refrigerant in the bypass path becomes superheated vapor at the end of the bypass path, a condition which is required for the multizone variable capacity application of the bypass concept.

Operational benefits even for high heat loads are achieved by use of the bypass concept, as discussed in the various documents referenced herein. For example the refrigeration system of Fig. 10 may be run with no diversion of refrigerant to the bypass path for maximum heat loads, or alternatively, flow control valve 154 in bypass path 133 can be operated to divert about 5% to 15% of the mass flow of the refrigerant to the bypass path when maximum cooling capacity is required due to high thermal load, and to divert up to about 60% or even up to 70% of the refrigerant to the bypass path according to reductions in thermal load.

Fig. 11 shows another embodiment of the present invention, generally denoted at 130A. This embodiment is similar to that of Fig. 10, except that the upstream end of the

bypass path 133A is differently configured. In the interest of brevity, only the differences will be described.

As shown in Fig. 11, the upstream end of bypass path 133A is formed of a large diameter pipe, preferably about 3 times the diameter of the rest of the tubing in the system (e.g., 2.54 cm.) This is used as an in-line storage space (or accumulator) for refrigerant during the bypass operation. When the maximum cooling is required in the multizone system, the valve 194 in the bypass path is closed to prevent diversion of refrigerant to bypass path 133A whereby 100% of the refrigerant is circulated through evaporator subsystem 140 and condenser 136 in primary refrigerant loop 132.

As the cooling load decreases below some preset level, e.g., corresponding to a 50% reduction in required cooling capacity, valve 95A is opened to allow more than 50% of the refrigerant to enter bypass path 133A. As the refrigerant flows through the bypass path, however, a major fraction of the refrigerant in the bypass path remains stored in accumulator 190. The amount stored is preferably at least about 50%, and more preferably, up to about 70% of the diverted mass flow. The remaining portion of the diverted refrigerant passes through secondary expansion device 148 to heat exchangers 152 and 152 which function to provide subcooling and desuperheating, as described above.

The advantage of the configuration shown in Fig. 11 is the feed line of the secondary expansion device used as an accumulator, which stores a part of liquid refrigerant when the cooling need is reduced.

As a further preferred embodiment, system 130A can be operated in the manner described above for system 130 (Fig. 10). Thus, when heat load is at 100%, valve 95A

and 194 are operated to permit about 5-10% of the refrigerant to flow through the bypass path in order to take advantage of bypass technology. When heat load decreases below a predetermined threshold, e.g., 70-80% or the design maximum, the control valves are operated to increase the mass flow through the bypass path thereby decreasing the cooling capacity of the whole system.

In describing the invention, specific terminology has been employed for the sake of clarity. However, the invention is not intended to be limited to the specific descriptive terms, and it is to be understood that each specific term includes all technical equivalents that operate in a similar manner to accomplish a similar purpose.

Similarly, the embodiments described and illustrated are also intended to be exemplary, and various changes and modifications, and other embodiments within the scope of the invention will be apparent to those skilled in the art in light of the disclosure. The scope of the invention is therefore intended to be defined and limited only by the appended claims, and not by the description herein.

WHAT IS CLAIMED IS:

1. A variable capacity refrigeration system comprising:
a primary refrigerant loop and a refrigerant bypass path,
the primary refrigerant loop including:
compressor means operating continuously at a constant speed irrespective of the
heat load for compressing refrigerant in the primary refrigerant loop;
condensing means for removing heat from the compressed refrigerant;
primary expansion means for relieving the pressure on the refrigerant exiting the
condensing means; and
evaporator means for removing heat from a space being cooled;
the refrigerant bypass path including:
secondary expansion means;
first and second heat exchanger means thermally coupled to the primary refrigerant
path for removing heat from the refrigerant therein; and
flow control means operative according to heat load for diverting a portion of the
refrigerant in the primary refrigerant loop up stream of the primary
expansion means to flow through the heat exchangers to an inlet of the
compressor means,
the heat exchanger means being coupled to the primary refrigerant loop to provide
subcooling and desuperheating, with sufficient heat being absorbed by the
refrigerant in the bypass path to fully vaporize the diverted refrigerant,
irrespective of the mass flow thereof, before it is returned to the compressor
inlet.

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2. A refrigeration system according to claim 1, further including pressure differential accommodation means for connecting the bypass path and the outlet of the evaporator means to an inlet of the compressor means.

3. A refrigeration system according to claim 1, wherein the evaporating means includes:

a plurality of evaporating units at selected locations in a space being cooled;

means for selectably connecting the inlet of each evaporating units to the outlet of the primary expansion means; and

means for connecting the outlets of the evaporation units in common to an inlet of the compressor means.

4. A refrigeration system according to claim 3, wherein the connecting means comprises pressure differential accommodation means for connecting the bypass path and the outlet of the evaporator means to the inlet of the compressor means.

5. A refrigeration system according to claim 1, further including means in the bypass path for storing a portion of the refrigerant diverted to the bypass path.

6. A refrigeration system comprising:
a primary refrigerant loop including:

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a compressor that operates continuously at a constant speed, irrespective of the heat load;

a condenser;

a primary expansion device; and

an evaporator; and

a refrigerant bypass path which connects the primary loop to an inlet of the compressor,

the bypass path including:

a secondary expansion device;

first and second heat exchangers; and

a flow control device,

the flow control device being operative according to the heat load, to permit a

portion of the refrigerant exiting from the condenser to pass through the

secondary expansion device and to flow through the heat exchangers to the

inlet of the compressor,

the heat exchangers being thermally coupled to the primary refrigerant loop

whereby one heat exchanger provides subcooling and the other provides

desuperheating, and sufficient heat is absorbed by the refrigerant in the

bypass path to fully vaporize it irrespective of the mass flow thereof, before

it is returned to the compressor inlet.

7. A refrigeration system according to claim 6, further including pressure differential accommodation device which connects the bypass path and the outlet of the evaporator to the inlet of the compressor.

8. A refrigeration system according to claim 6, wherein the evaporator includes: a plurality of evaporating units at selected locations in a space being cooled, the outlets of the evaporation units being connected in common to the inlet of the compressor; and control valves operable to selectably connect the respective inlets of the evaporating units to the outlet of the primary expansion device.

9. A refrigeration system according to claim 8, wherein: the outlets of the evaporation units are connected in common to an inlet of a pressure differential accommodation device; the outlet of the bypass path is connected to another inlet of the pressure differential accommodation device; and the outlet of the pressure differential accommodation device is connected to the inlet of the compressor.

10. A refrigeration system according to claim 6, further including an accumulator which stores a portion of the refrigerant diverted to the bypass path.

11. A variable capacity refrigeration system according to claim 6, wherein: the bypass path includes, in series, the secondary expansion device, a first flow path of the first heat exchanger, and a first flow path of the second heat exchanger;

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a second flow path of the first heat exchanger is coupled between the outlet of the condenser and the inlet of the primary expansion device;
a second flow path of the second heat exchanger is coupled between the outlet of the compressor and the inlet of the condenser; and
the inlet to the bypass path is provided between the outlet of the first flow path of the first heat exchanger and the inlet of the primary expansion device.

12. A refrigeration system according to claim 11, wherein the flow control device is operative to increase the mass flow of refrigerant to the bypass path when the heat load decreases, and to decrease the mass flow of refrigerant to the bypass path when the heat load increases.

13. A refrigeration system according to claim 11, further including an accumulator which stores a portion of the refrigerant diverted to the bypass path.

14. A refrigeration system according to claim 13, wherein the accumulator is comprised of an enlarged diameter section of the bypass path conduit upstream of the secondary expansion device.

15. A refrigeration system according to claim 6, wherein the bypass path is connected to the outlet of the primary loop downstream of the first heat exchanger.

16. A refrigeration system according to claim 6, wherein:

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the evaporator is comprised of a plurality of parallel-connected evaporator units located in respective portions of the space being cooled by the system; and

the system further includes a plurality of control valves respectively connecting the primary expansion device to the evaporator units, the control valves being operable to idle respective evaporator elements by shutting off the flow of refrigerant thereto when cooling of a particular location is not required at given time; and

the flow control valve in the bypass path is operative to control the flow of refrigerant in the bypass path such that refrigerant mass flow which is not required in the primary refrigerant path when a particular evaporator element is idle flows to the bypass path.

17. A refrigeration system according to claim 16, wherein the flow control valve in the bypass path is operable to divert about 5% to about 15% of the refrigerant to the bypass path when maximum cooling capacity is required due to operation of all the evaporator elements, and to divert up to about 70% of the refrigerant to the bypass path according to reductions in thermal load due to deactivation of particular evaporator elements.

18. A refrigeration system according to claim 6, wherein the flow control valve in the bypass path is operable to divert about 5% to about 15% of the refrigerant to the bypass path when maximum cooling capacity is required due to high thermal load, and to divert up to about 70% of the refrigerant to the bypass path according to reductions in thermal load.

19. A refrigeration system according to claim 6, wherein the refrigerant circulated in the system consists of a single component.

20. A refrigeration system according to claim 6, wherein the refrigerant circulated in the system is a mixed-refrigerant comprising a plurality of components selected to provide a desired combination of thermal and flammability characteristics.

21. A refrigeration system according to claim 6, wherein:
the refrigerant exiting the condenser flows through one flow path of the first heat exchanger; and
a portion of the refrigerant exiting the first flow path of the first heat exchanger is coupled through the secondary expansion device to a second flow path of the first heat exchanger whereby heat is transferred from the refrigerant exiting the condenser to the refrigerant flowing in the bypass path and subcooling is provided.

22. A refrigeration system according to claim 21, wherein:
the refrigerant exiting the compressor flows through one flow path of the second heat exchanger; and
the outlet of the second flow path of the first heat exchanger is coupled to a second flow path of the second heat exchanger whereby heat is transferred from the refrigerant exiting the compressor to refrigerant flowing in the bypass path and desuperheating is provided.

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23. A refrigeration system according to claim 22, further including an accumulator located between the outlet of the first flow path of the first heat exchanger and the secondary expansion device.

24. A refrigeration system according to claim 6, wherein:
refrigerant diverted to the bypass path passes a first flow path of the first heat exchanger and absorbs heat from the refrigerant exiting the condenser flowing through a second flow path of the first heat exchanger;
the refrigerant exiting the first flow path of the first heat exchanger flows through a first flow path of the second heat exchanger and absorbs heat from the refrigerant exiting the compressor flowing through a second flow path of the second heat exchanger; and
the refrigerant exiting the first flow path of the second heat exchanger flows to the inlet of the compressor,
the quantity of heat having being transferred to the refrigerant in the bypass path from the refrigerant in the primary loop is sufficient to fully vaporize it before it reaches the compressor inlet.

25. A variable capacity refrigeration system according to claim 24, wherein the flow control device in the bypass path is operative to increase the mass flow of refrigerant diverted to the bypass path when the heat load decreases, and to decrease the mass flow of refrigerant to the bypass path when the heat load increases.

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26. A method of increasing the efficiency of a variable capacity refrigeration system comprised of a primary refrigerant loop including a compressor, a condenser, a primary expansion device, and an evaporator connected together to form a closed loop system with a refrigerant circulating therein, the method comprising the steps of: diverting a portion of the refrigerant exiting the condenser into a secondary refrigerant path which includes a secondary expansion device and two heat exchanger, one heat exchanger being thermally coupled to the primary refrigerant path between the condenser outlet and the primary expansion device inlet, and the other being thermally coupled to the primary refrigerant path between the outlet of the compressor and the inlet of the condenser; passing the diverted refrigerant through the heat exchangers to remove heat from refrigerant flowing in the primary refrigerant path; and returning the refrigerant exiting the bypass path and the refrigerant exiting the evaporator to an inlet of the compressor.

27. A method according to claim 26, wherein the refrigerant exiting the bypass path and the refrigerant exiting the evaporator are returned to the compressor inlet through a pressure differential accommodating device that mixes two vapors at different pressures.

28. A method according to claim 26, wherein the refrigerant is diverted to the bypass path at a location downstream of the heat exchanger.

29. A method according claim 26, further including the steps of:

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controlling the quantity of refrigerant outflow from the condenser which is diverted to the bypass path to adjust the cooling capacity of the system according to the thermal load; and
running the compressor substantially continuously at a constant speed irrespective of the heat load.

30. A method according claim 29, wherein between about 5% and about 15% of the liquid refrigerant outflow from the condenser is diverted to the bypass path under maximum heat load conditions, and up to 70% is diverted under minimum heat load conditions.

31. A method according claim 26, wherein:
the primary refrigeration path includes a plurality of evaporator units located in respective locations to be separately cooled; and the method further includes the steps of:
diverting a predetermined minimum quantity of refrigerant to the bypass path when maximum cooling capacity is required to cool all of the locations; and
diverting increasing quantities of refrigerant to the bypass path as thermal load decreases.

32. A method according claim 31, further includes the steps of:
idling particular evaporator units in locations which do not require cooling at a given time by blocking the flow of refrigerant thereto;
diverting the refrigerant normally delivered to a particular evaporator units to the bypass path what that evaporator is idle.

33. A method according to claim 26, wherein the refrigerant circulated in the system consists of a single component.

34. A method according to claim 26, wherein the refrigerant circulated in the system is a mixed-refrigerant comprising a plurality of components selected to provide a desired combination of thermal and flammability characteristics.

35. A method according to claim 26, further including the step of storing a portion of the refrigerant diverted to the bypass path in an accumulator.

36. A method according to claim 26, further including the step of subcooling the refrigerant exiting the condenser by:
passing it through a first flow path of the first heat exchanger; and
diverting a portion of the refrigerant exiting the first flow path of the first heat exchanger through the secondary expansion device to a second flow path of the first heat exchanger whereby heat is transferred from the refrigerant exiting the condenser to the refrigerant flowing in the bypass path.

37. A method according to claim 36, further including the step of desuperheating the
refrigerant exiting the compressor by:
passing it through a first flow path of the second heat exchanger; and

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passing the refrigerant exiting the second flow path of the first heat exchanger through a second flow path of the second heat exchanger whereby heat is transferred from the refrigerant exiting the compressor to refrigerant flowing in the bypass path.

38. A method according to claim 37, further including the step of storing the refrigerant diverted to the bypass path from the outlet of the first flow path of the first heat exchanger in an accumulator.

39. A method according to claim 37, wherein the quantity of heat having transferred to the refrigerant in the bypass path from the refrigerant in the primary loop is sufficient to fully vaporize it before it is delivered to the compressor inlet.

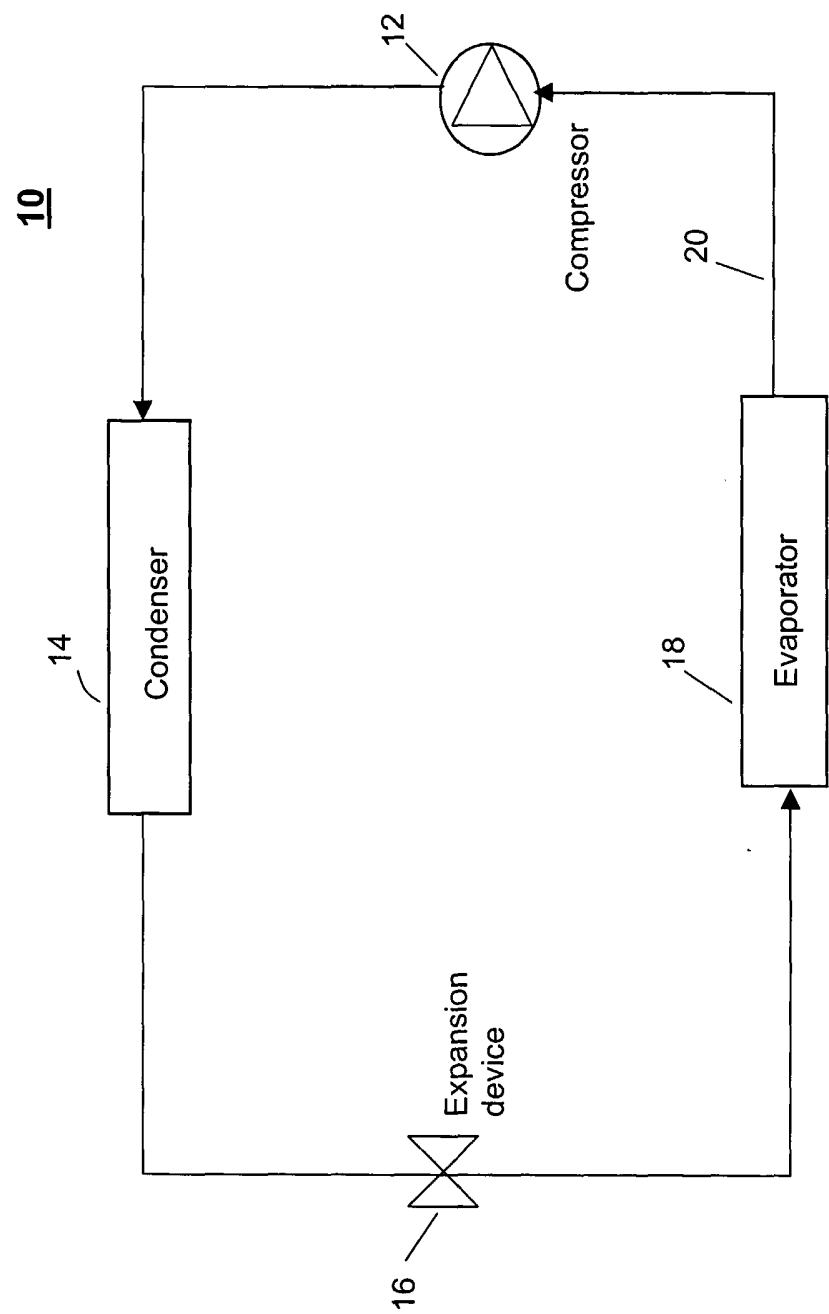


FIG. 1 PRIOR ART

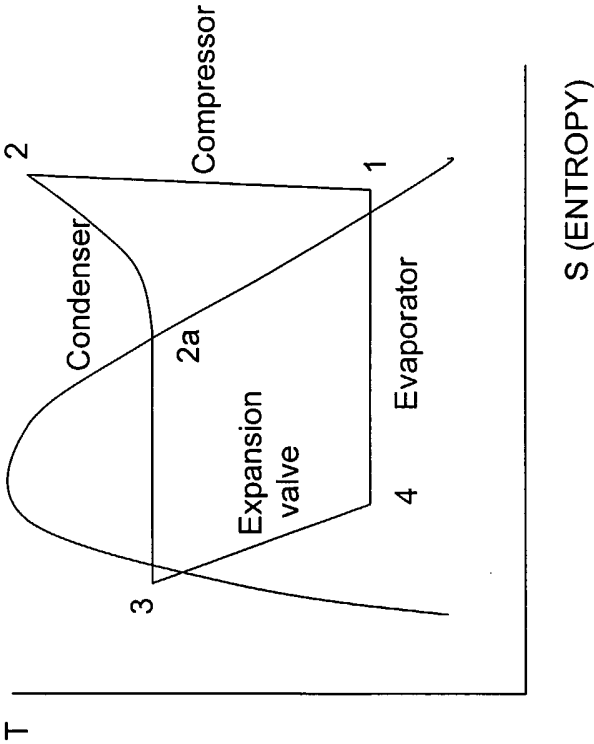


FIG. 2

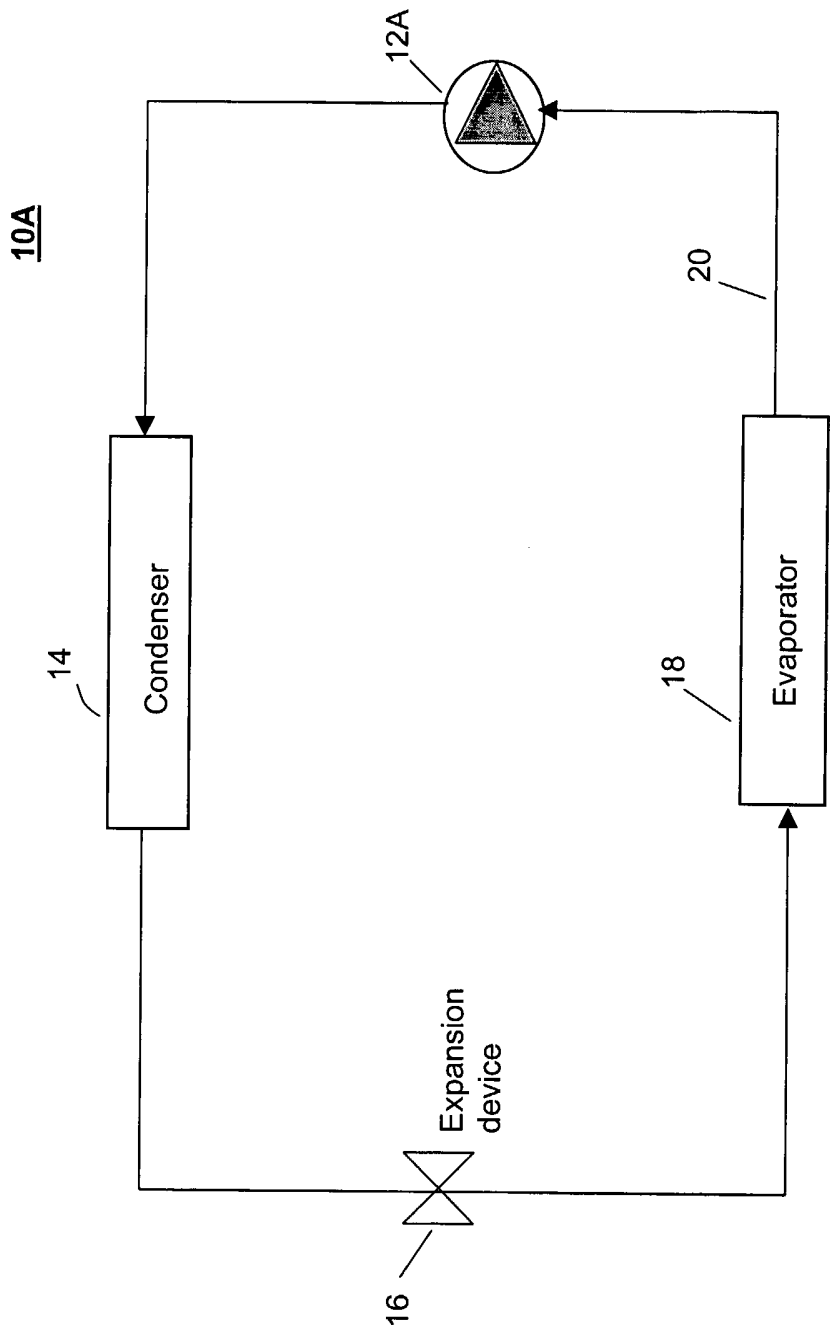


FIG. 3 PRIOR ART

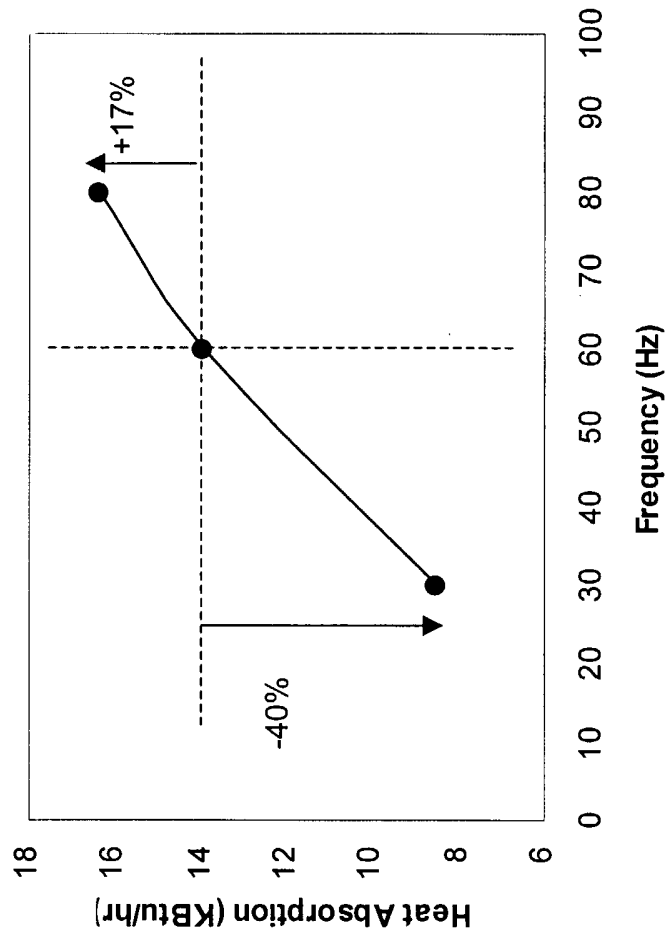


FIG. 4

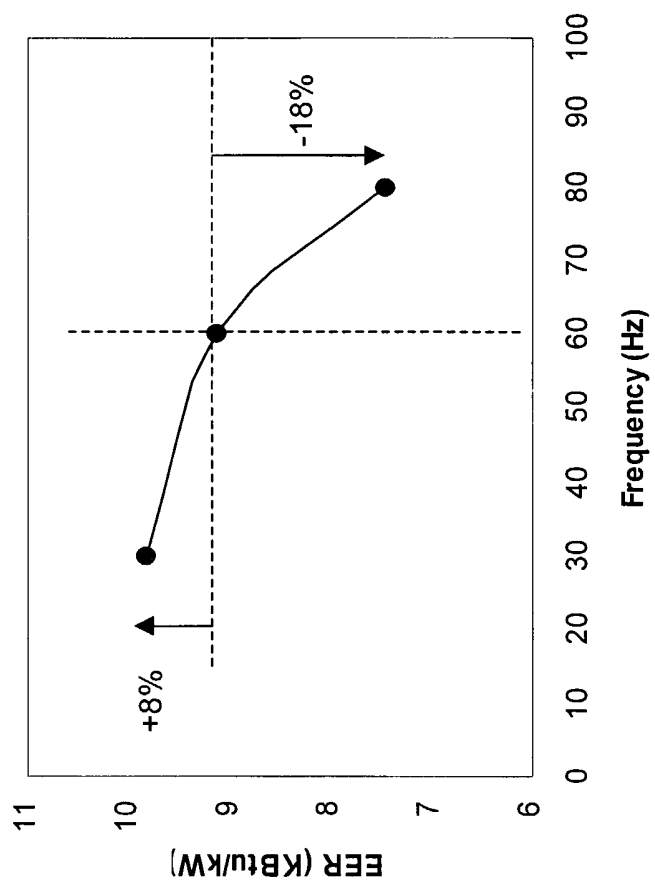


FIG. 5

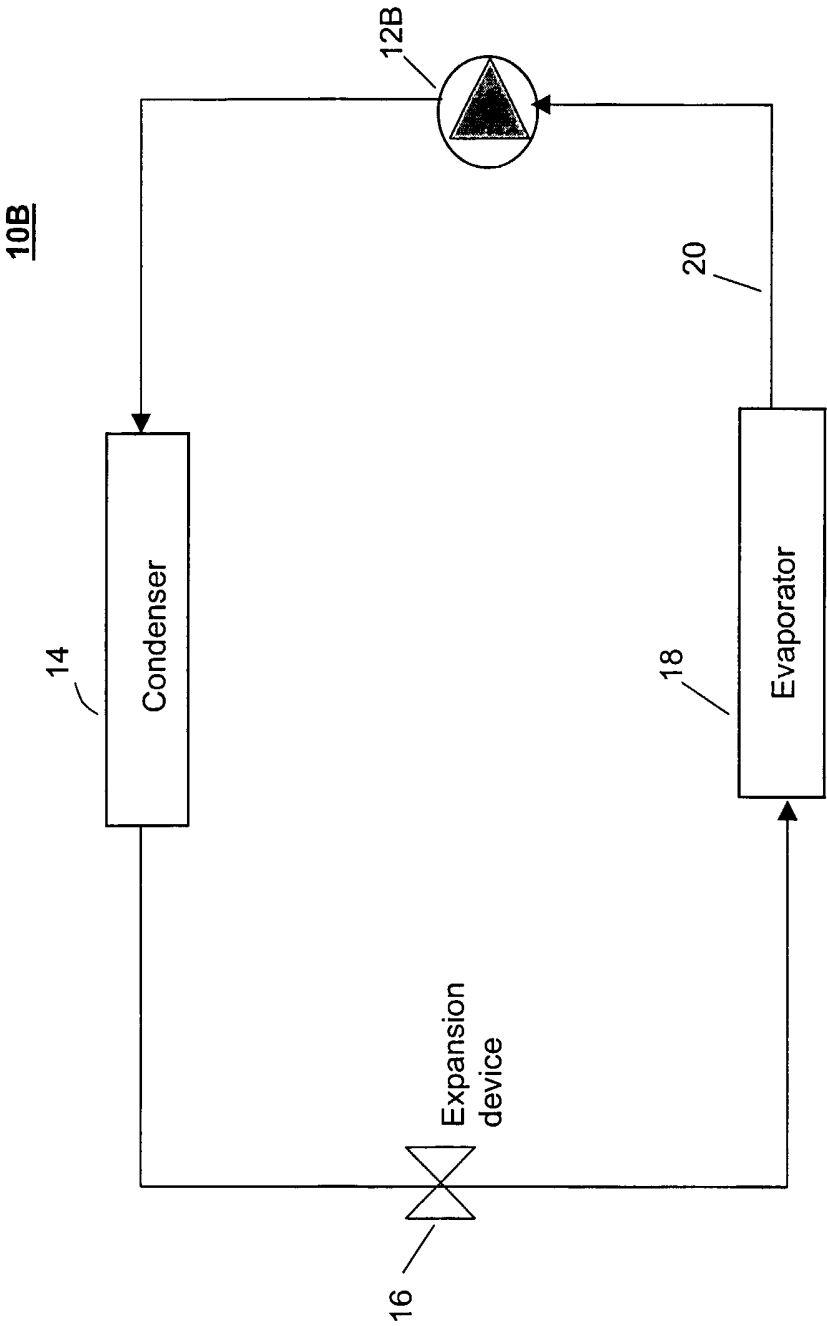


FIG. 6 PRIOR ART

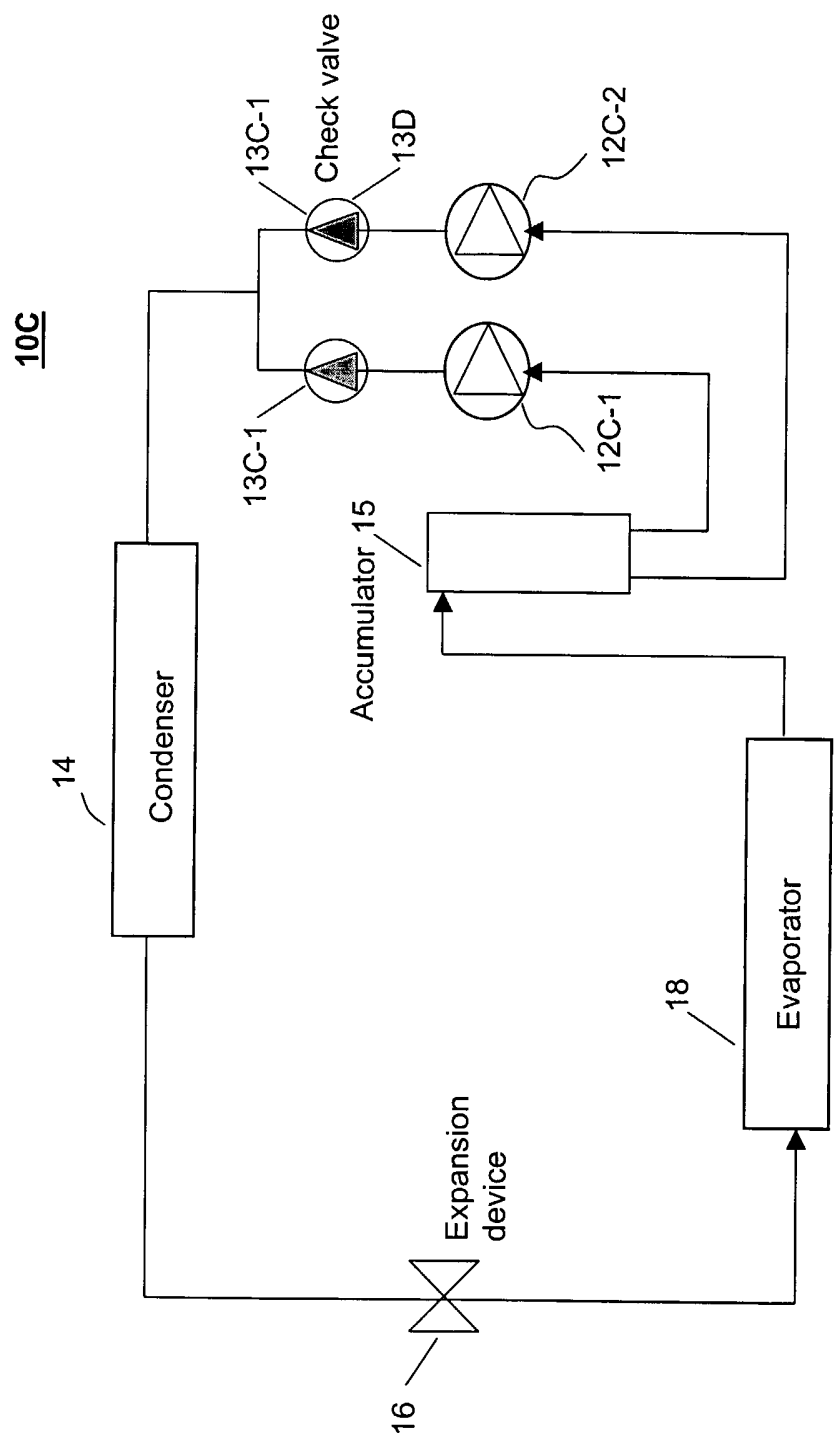


FIG. 7 PRIOR ART

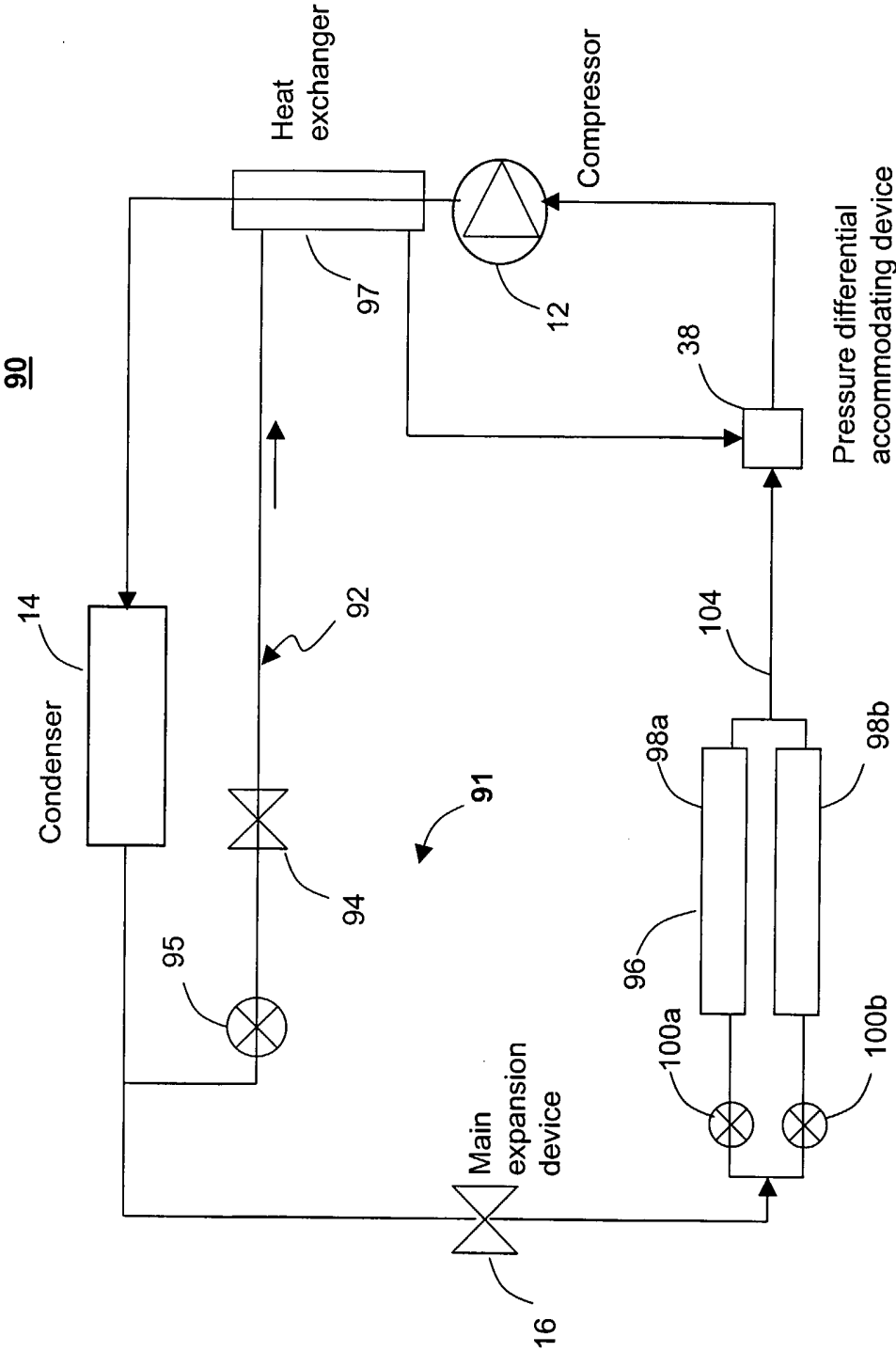


FIG. 8 PRIOR ART

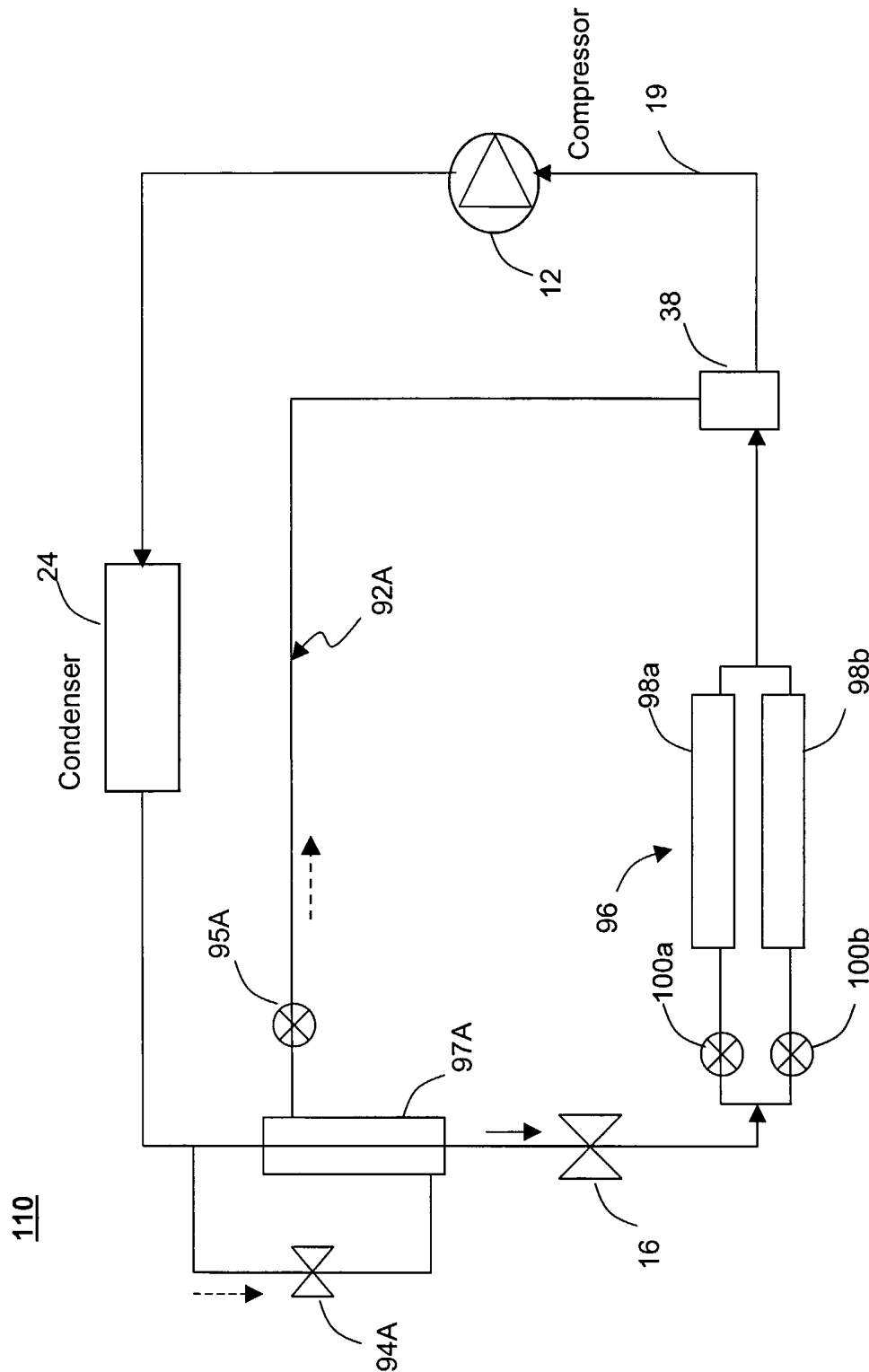


FIG. 9 PRIOR ART

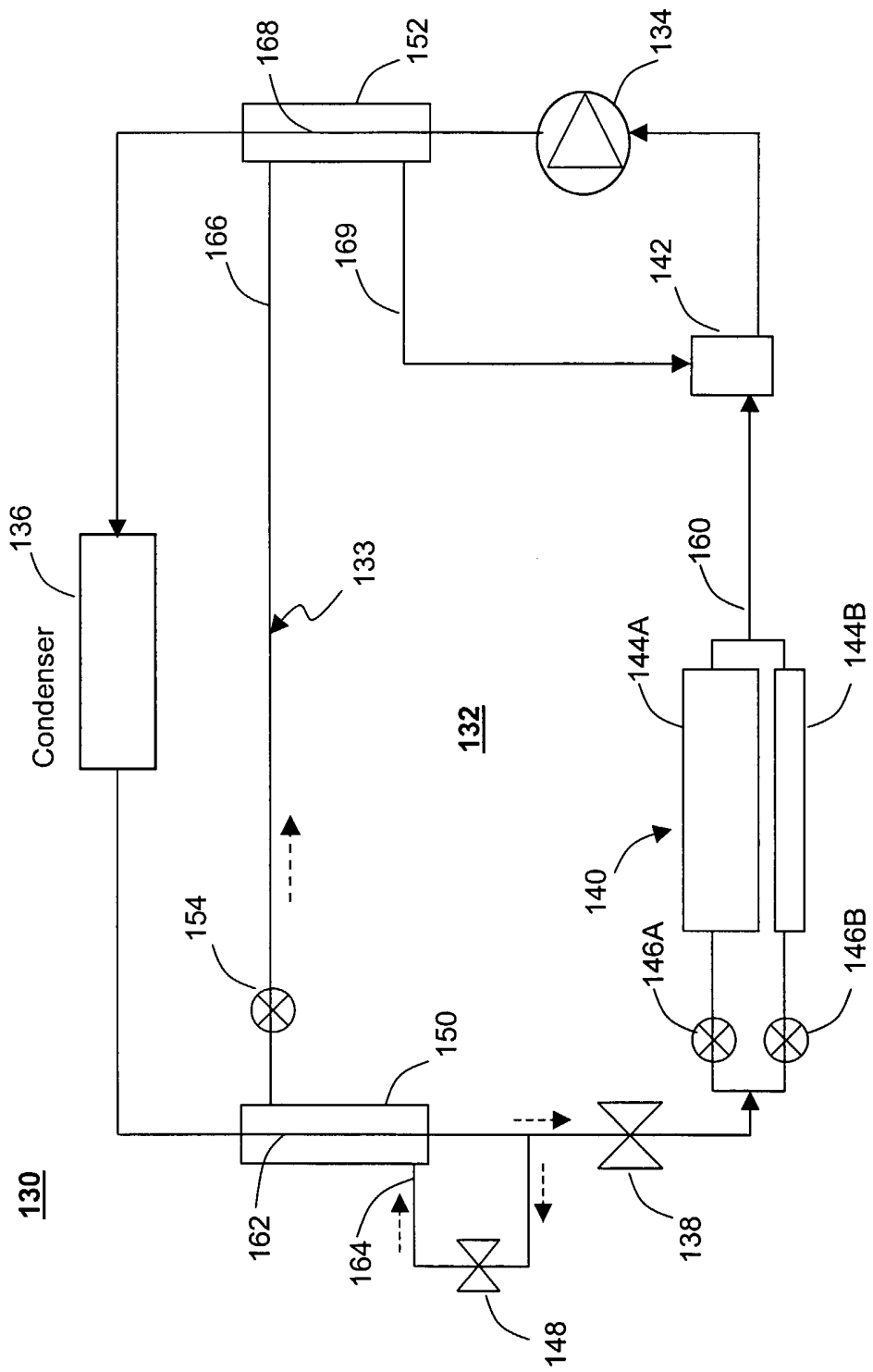


FIG. 10

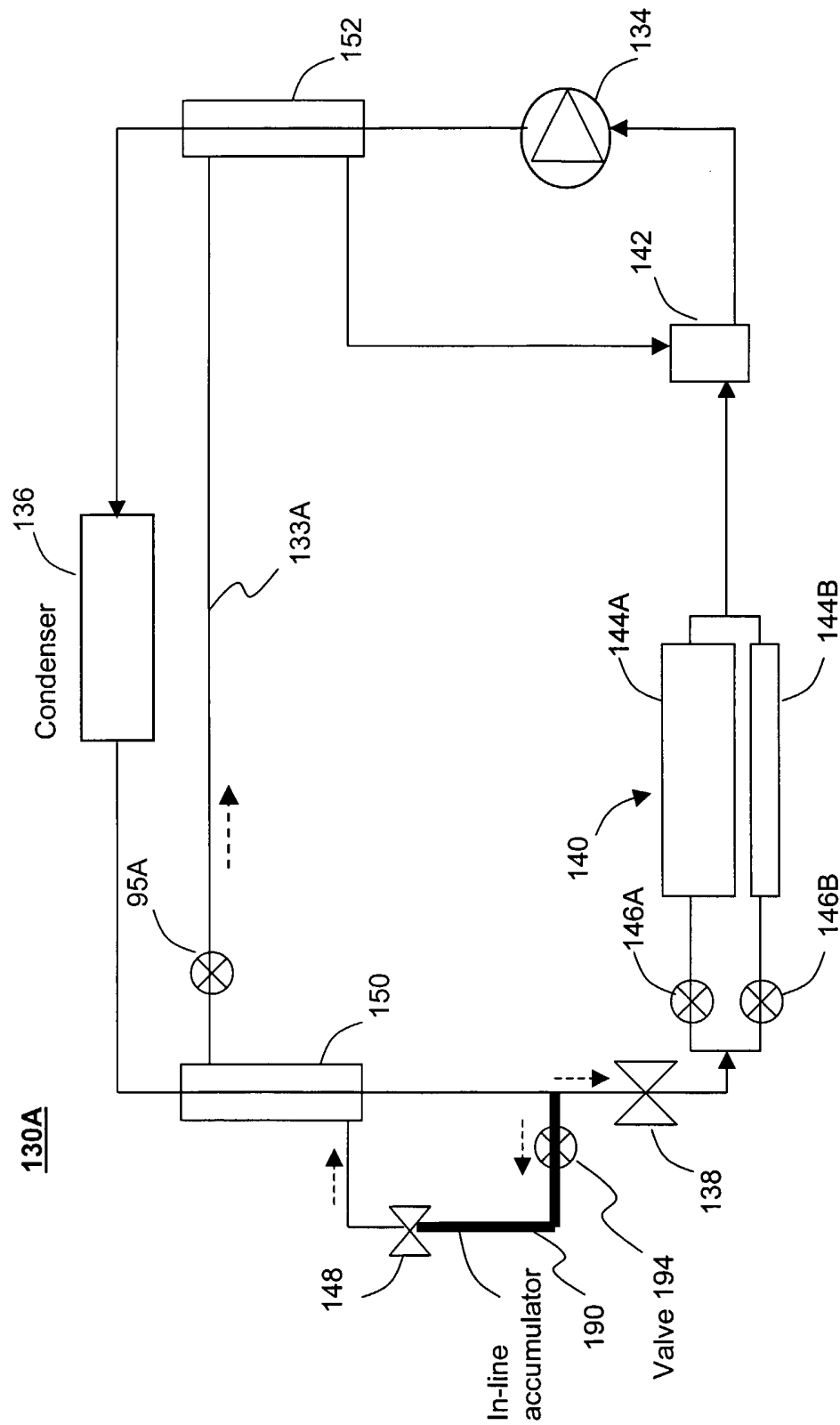


FIG. 11