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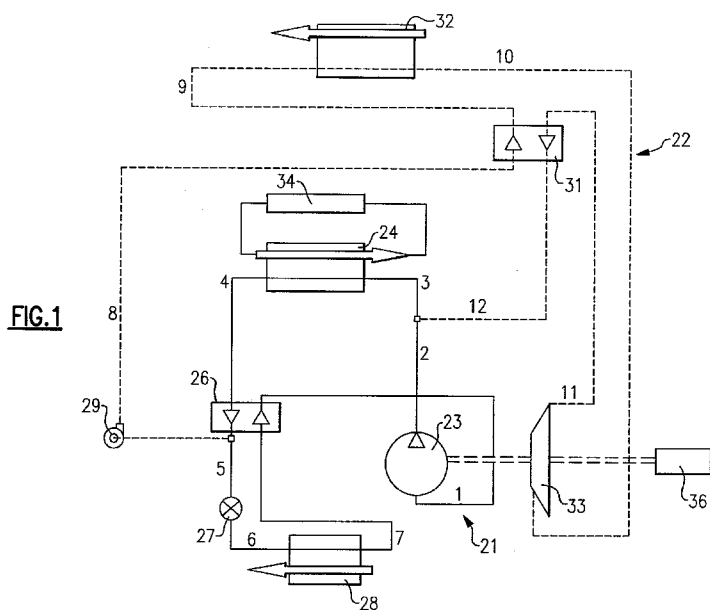
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(54) Title: TRANSCRITICAL THERMALLY ACTIVATED COOLING, HEATING AND REFRIGERATING SYSTEM



(57) Abstract: A combined vapor compression and vapor expansion system uses a common refrigerant which enables a supercritical high pressure portion and a subcritical low pressure portion of the vapor expansion circuit. Provision is made to combine the refrigerant flow from the vapor expander and from the compressor discharge. The outdoor heat exchanger is so sized and designed that the working fluid discharged therefrom is always in a liquid form so as to provide a liquid into the pump inlet. The pump and expander are so sized and designed that the high pressure portion of the vapor expansion circuit is always super-critical. A topping heat exchanger, liquid to suction heat exchanger, and various other design features are provided to further increase the thermodynamic efficiency of the system.

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## **Transcritical Thermally Activated Cooling, Heating and Refrigerating System**

### **Cross Reference to Related Application**

[0001] This disclosure relates to pending US Application 07/18958, assigned to the assignee of the present disclosure.

[0002] Reference is made to and this application claims priority from and the benefit of U.S. Provisional Application Serial No. 61/173,776, filed April 29, 2009, entitled "TRANSCRITICAL THERMALLY ACTIVATED COOLING, HEATING AND REFRIGERATING SYSTEM", which application is incorporated herein in their entirety by reference.

### **Technical Field**

[0003] This disclosure relates generally to vapor compression systems and, more particularly, to a combined vapor compression and vapor expansion system.

### **Background of the Disclosure**

[0004] It is known to combine a vapor compression system with a vapor expansion, i.e. Rankine cycle, system. See, for example, US Patent 6,962,056, assigned to the assignee of the present invention, and US Patent 5,761,921.

[0005] US Patent 5,761,921 generates power in the Rankine cycle which is then applied to drive the compressor of the vapor compression cycle, and the combined systems operate on three pressure levels, i.e. the boiler, condenser and evaporator pressure levels. A common refrigerant R-134 is used in both the vapor compression and the Rankine cycle systems. Such combined systems have generally not allowed use of transcritical refrigerants, since transcritical systems have generally not had a condenser (but only a gas cooler), and therefore no liquid refrigerant available downstream of the gas cooler for pumping through the Rankine circuit. The expander requires a high entering pressure, but the high inlet pressure elevates the boiling temperature and the leaving temperature of the heating fluid carrying the thermal power. The elevated leaving temperature reduces the extent of the waste heat utilization. For those reasons the systems do not sufficiently utilize available thermal energy and, therefore have a low level of thermodynamic

efficiency. Further, they do not provide an adequate performance when the available hot source is below 180°F.

**[0006]** US Patent Application 07/18958 provides for a combined flow of refrigerant from the two systems at the discharge of the compressor and the expander, respectively. Further, a suction accumulator is provided such that liquid refrigerant is always available to the pump in the Rankine cycle system such that transcritical operation is made possible. However, such use of a suction accumulator may be undesirable because of the need for a larger pump with greater power requirements. The pump power is defined by a product of pressure differential across the pump and the specific volume of the refrigerant stream at the pump inlet. Although the liquid in the suction accumulator has a low specific volume, the pump may be required to work against high pressure differentials. When the disadvantage of the pressure differential increase exceeds the advantage of the liquid specific volume reduction, feeding of the pump with liquid refrigerant from the condenser is considered to be an advantage over the use of a suction accumulator.

#### **Disclosure**

**[0007]** Briefly, in accordance with one aspect of the disclosure, a combined vapor compression circuit and vapor expansion circuit includes a common refrigerant which enables a supercritical high pressure portion and a sub-critical low pressure portion of the vapor expansion circuit, and combines the refrigerant from the expander discharge and the compressor discharge at the entrance to the outdoor heat exchanger. The outdoor heat exchanger is so sized and designed that the refrigerant discharge therefrom is always in a liquid form so that it can flow directly to the vapor expansion circuit pump. The pump and expander are so sized and designed that the high pressure portion of the vapor expansion circuit is always super-critical.

**[0008]** In accordance with another aspect of the disclosure, the outdoor heat exchanger includes a cooling tower to ensure that the refrigerant is converted to a liquid in the heat exchanger.

[0009] In accordance with another aspect of the disclosure, a liquid to suction heat exchanger is provided between the outdoor heat exchanger and the pump in order to increase subcooling and refrigerant density prior to the refrigerant liquid's passing to the pump.

[0010] In accordance with yet another aspect of the invention, a topping heat exchanger is provided downstream of the expander outlet for the purpose of regenerating enthalpy of the hot stream.

[0011] In accordance with yet another aspect of the invention, a power generation vapor expansion circuit is used as a stand alone system and generates electrical power, which may be used as an electrical power supply for different purposes, including driving a refrigeration system.

### **Brief Description of the Drawings**

[0012] For a further understanding of these and objects of the invention, reference will be made to the following detailed description of the invention which is to be read in connection with the accompanying drawing, where:

[0013] FIG. 1 is a schematic illustration of a thermally activated refrigerant system for cooling or heating only.

[0014] FIG. 2 is a schematic illustration of a temperature-entropy, (T-S) diagram of processes for the thermally activated refrigerant system for cooling or heating only.

[0015] FIGS. 3A-3C are schematic illustrations comparing glides in supercritical and subcritical applications, respectively.

[0016] FIG. 4 is a schematic illustration of a thermally activated vapor expansion system with multi-stage expansion.

[0017] FIG. 5 is a schematic illustration of a T-S diagram of processes for the thermally activated vapor expansion system with multi-stage expansion.

[0018] FIG. 6 is a schematic illustration of a thermally activated refrigerant system providing both air conditioning and refrigeration.

[0019] FIG. 7 is a schematic illustration of a thermally activated heat pump with two expansion devices.

[0020] FIG. 8 is a schematic illustration of a thermally activated heat pump with one bidirectional expansion device.

[0021] FIG. 9A and 9B are schematic illustrations of reversing and check valve arrangements, respectively.

[0022] FIG. 10 is a schematic illustration of a thermally activated heat pump with two different hot sources.

[0023] FIG. 11 is a schematic illustration of a thermally activated heat pump with multi-stage compression.

[0024] FIG. 12 is a schematic illustration of a thermally activated heat pump with a vapor-to-vapor ejector.

[0025] FIG. 13 is a schematic illustration of a thermally activated heat pump with a two-phase ejector.

[0026] FIG. 14 is a schematic illustration of a thermally activated heat pump with an economized cycle.

[0027] FIG. 15 is a schematic illustration of a thermally activated heat pump with a two-phase expander.

#### **Detailed Description of the Disclosure**

[0028] While the present disclosure has been particularly shown and described with reference to the preferred mode as illustrated in the drawing, it will be understood by one skilled in the art that various changes in detail may be effected therein without departing from the spirit and scope of the disclosure as defined by the claims.

[0029] In accordance with FIG. 1 a thermally activated refrigerant system incorporates a vapor compression circuit 21 shown as solid lines and a vapor expansion circuit 22 shown as dashed lines. The vapor compression circuit 21 includes a compressor 23, a condenser 24, a liquid-to-suction heat exchanger 26, an expansion device 27, and an evaporator 28. The vapor expansion circuit 22 consists of a pump 29, a topping heat exchanger 31, a heater 32, an expander 33, and the condenser 24. A refrigerant vapor stream at the outlet from the compressor and a vapor refrigerant stream at the outlet from the expander are connected at the condenser inlet to provide a combined flow through the condenser 24. A refrigerant

liquid stream at the condenser outlet, or at the outlet of the liquid to suction heat exchanger 26 as shown, splits into two streams: one feeds the pump, and another circulates through the components of the vapor compression circuit.

**[0030]** The thermally activated refrigeration system has three pressure levels: a heating pressure, a heat rejection pressure level, and evaporating pressure. The heating pressure is the pump discharge pressure, the heat rejection pressure is compressor or expander discharge, and the evaporating pressure is the compressor suction pressure. The heating and heat rejection pressures are high and low pressures of the vapor expansion circuit. The heat rejection and evaporating pressures are high and low pressures of the vapor compression circuit

**[0031]** One common working fluid is used for both the vapor compression and the vapor expansion circuits. The working fluid has the following feature: it provides super-critical operation for a high pressure portion of the vapor expansion circuit and a sub-critical operation for the low-pressure portion of the vapor expansion circuit. Thus, the working fluid in the vapor expansion circuit at the high pressure remains gaseous, but the working fluid in the condenser appears in the region to the left of the vapor dome and is liquefied. Examples of such working fluid are CO<sub>2</sub> or CO<sub>2</sub> based mixture, such as CO<sub>2</sub> and propane, or the like.

**[0032]** The heater 32 provides a thermal contact between a heating medium and the pumped refrigerant stream. Usually the heat source is a waste heat such as may be available from a fuel cell, a solar device, a micro-turbine, a reciprocating engine, or the like. Pressure in the heater is supercritical, that is, above the critical pressure of the refrigerant. This provides a favorable temperature glide compatible with a temperature glide of the heating medium shown on FIG. 2. The heater 32 should be designed to provide equality of heat capacity rates of both streams and enable the highest temperature differentials across each stream. The glides and equality of the heat capacity rates provide a higher extent of waste heat utilization and a high entering expander temperature, resulting in improved expander performance. If the hot source is not a waste heat, the equality of heat capacity rates may not be required; the temperature glide provides a higher refrigerant temperature at the expander inlet, which improves the performance characteristics of the expander.

**[0033]** The condenser 24 provides a thermal contact between a cooling medium and the combined refrigerant stream outgoing from the compressor 23 and expander 33. The temperature of the cooling medium in the condenser 24 is always maintained below the refrigerant critical point to enable refrigerant condensation at the heat rejection pressure, with the liquid refrigerant feeding the pump 29.

**[0034]** During periods of operation at higher ambient temperatures, the condenser 24 may be fed by a cooling tower 34 to ensure condensation of the refrigerant vapor. Another option is to use CO<sub>2</sub> and propane or the like in order to elevate the critical point of the fluid sufficiently above the level of ambient temperature to enable the condensation process at the heat rejection pressure.

**[0035]** The heating pressure in the heater 32 is controlled by an expander-to-pump capacity ratio, which is defined by an expander-to-pump rotating speed ratio, a liquid refrigerant temperature at the pump inlet, and a vapor refrigerant state at the expander inlet.

**[0036]** The liquid-to-suction heat exchanger 26 is optional. It slightly sub-cools a liquid stream outgoing from the condenser 24 and substantially superheats a vapor stream flowing from the evaporator 28. The subcooling reduces the pump power due to reduction of the refrigerant density at the pump inlet. Also, it increases the enthalpy difference across the evaporator 28 and increases the evaporator effect. The superheat decreases the refrigerant density at the compressor inlet and reduces the compressor mass flow rate and the evaporator capacity. The superheat effect is usually stronger and the overall effect is usually detrimental. Therefore, the liquid-to-suction heat exchanger 26 is only used if a certain superheat at the compressor inlet is required.

**[0037]** The topping heat exchanger 31 substantially improves thermodynamic efficiency of the system when the hot source temperature is high. When the hot source temperature is low, the topping heat exchanger is not needed.

**[0038]** Power generated in the expander 33 may drive the compressor 23 and the pump 29. All three machines may be placed on the same shaft. There is an option to couple the shaft with a power generator 36 to provide not only cooling or heating duty, but also electrical power. The expander 33 may be coupled with a power generator only, in which case the power generator 36 powers the compressor

23 and pump 29. In addition, optionally, it may generate supplemental electrical power.

**[0039]** The vapor expansion circuit may be implemented as a separate power generation system. Power generated in the power generation system may be used to power a heat pump, air conditioner, refrigerator, or any other electrical device.

**[0040]** All components sitting on the same shaft may be covered by a semi-hermetic or hermetic casing to reduce risk of leakage.

**[0041]** The pump 29 may be a variable or multiple speed device or a constant speed device. Speed variation helps to satisfy the variable demands of refrigeration, air conditioning or heating.

**[0042]** Referring now to FIG. 2, the T-S diagram is shown for both the vapor compression circuit 21 and the vapor expansion circuit 22 of FIG. 1, with the various points of interest in the two figures being shown by the numerals 1-12. As will be seen, the line 9-10 is representative of the temperature and enthalpy increases that occur as the working fluid passes through the heater 32. Also, it should be appreciated that the alternate dash - dot line 37 is indicative of the T-S diagram for the cooled heating fluid passing through the heater 32. In this regard, it is desirable to not only use hot source fluids at temperatures of 180°F and above, as are used in conventional systems, but also enable the use of hot source fluids at temperatures below that level. This is made possible by the “glide” or slope of the line 37 that results from the use of CO<sub>2</sub> as the working fluid. This will be more clearly understood by reference to FIGS. 3A-3C.

**[0043]** Shown in FIG. 3A is a vapor expansion circuit which includes, in serial flow relationship a pump 38, a topping heat exchanger 39, a heater 41, an expander 42 and a condenser 43.

**[0044]** Shown in FIG. 3B is a T-S diagram for the FIG. 3A circuit when operating in a supercritical mode such as with CO<sub>2</sub> as the refrigerant. The numbers 1-8 in FIG. 3B correspond to the positions 1-8 in the FIG. 3A drawing. As will be seen, the line 3-4 in FIG. 3B represents the increases in temperature and enthalpy as the CO<sub>2</sub> passes through the heater 41, and the alternate dash and dot line 44 represents the T-S diagram for the cooled heating fluid. It will be recognized that the “glide”, or the slope of this line is substantial.

[0045] In contrast, the FIG. 3C illustration is a T-S diagram of the FIG. 3A circuit when operating in a subcritical mode, i.e. with a refrigerant other than CO<sub>2</sub>. Here, it will be recognized that the glide/slope of the line 46 is substantially less than that of the line 44 in FIG. 3B. The vertical component of the two lines 44 and 46, as shown by the arrowed lines 47 and 48, respectively, show the degree of waste heat utilization of the two alternatives of FIGS. 3B and 3C. As will be seen, the line 47 extends downwardly further than the line 48 which, in turn, indicates that heat sources (state 7) at lower temperatures may be employed as long as the temperature in state 8 is below the temperature in state 7. Thus, temperatures below 180°F may be suitable, such as, for example, temperatures of 150°F.

[0046] Referring now to FIG. 4, there is shown another embodiment wherein, rather than a single stage expander 33 as shown in FIG. 1, a two stage expander 49 is provided, as well as a second heater 51. The second heater 51 receives the heating fluid along line 52 and returns it to a point of the heater 32 by way of line 53. The temperature of the heating fluid in the heater 51 should be equal to the temperature of the point in the heater 32, where the line 53 is attached to. In operation, the refrigerant passes from the heater 32 to the first stage of the two stage expander 49 and then passes through the second heater 51, after which it passes through the second stage of the two stage expander 49, and then to the topping heat exchanger 31. The remainder of the circuit is as described above. The effect of using the two stage expander 49 and the second heater 51 is shown by the T-S diagram of FIG. 5 wherein the numbers (1-14) are indicative of the locations indicated in FIG. 4. It is known that the method of multi-stage expansion with reheat improves the expander efficiency, and reduces required pump power to thereby enable the use of smaller pumps and to reduce use of pump power to thereby improve the overall efficiency of the system.

[0047] Another embodiment is shown in FIG. 6 wherein a second vapor compression circuit 54 is provided in parallel with the vapor compression circuit 21. This enables the system to provide for both air conditioning, i.e. by way of the second vapor compression circuit 54 and refrigeration, i.e. by way of the vapor compression circuit 21.

**[0048]** The second vapor compression circuit 54 includes a second expansion device 56, a second evaporator or indoor unit 57 and a second compressor 58. The flow of refrigerant for that circuit originates upstream of the expansion device 27, and the discharge flow from the second compressor 58 is combined with the refrigerant flow from the topping heat exchanger 31 prior to the combination being combined with the flow from the discharge of the compressor 23. Thus, each of the vapor compression circuits 21 and 54 has its own compressor and evaporator unit, and all other components are shared between the two circuits. As will be seen both of the compressors are powered by the expander 33.

**[0049]** If the condenser 24 is an outdoor unit and the evaporator 28 is an indoor unit then the thermally activated refrigerant system generates cooling. If the condenser is an indoor unit and the evaporator is an outdoor unit then the thermally activated refrigerant system generates heating. To switch between the two modes of operation, one or more reversing or check valves may be provided as shown in FIGS. 7-15.

**[0050]** In order to allow the system to operate as a heat pump, a pair of reversing valves 59 and 61 are provided as shown in FIG. 7. Further, in addition to the expansion device 27 that is operable for use in the cooling mode, a second expansion device 62 is provided for use in the heating mode. Each of the expansion devices 27 and 62 include a bypass valve, i.e. valves 63 and 64, respectively, to permit operation in the respective cooling and heating modes. The expansion devices 27 and 62 are single directional expansion devices. In order to switch between the cooling and heating modes, the reversing valves 59 and 61, and the bypass valves 63 and 64, are all operated simultaneously.

**[0051]** A suction accumulator 66 maybe provided in order to satisfy the refrigerant charge demands for cooling and heating operation. Also, the suction accumulator 66 provides charge management and capacity control accumulating redundant amount of liquid refrigerant.

**[0052]** Further, a liquid-to-suction heat exchanger 67 may be provided as indicated.

**[0053]** A variation of the FIG. 7 system is shown in FIG. 8 wherein the two expansion devices are replaced by a single expansion device 68 which is designed

for bi-directional use. Thus when switching between the cooling and heating modes, the single expansion device and the reversing valves 59 and 61 are all switched simultaneously.

**[0054]** In FIG. 9A, the respective positions of the reversing valve 59 are shown to provide either cooling or heating operation. Thus, in cooling, the refrigerant passes from the reversing valve 59 through the heat exchanger 67, the expansion device 27, and then to the indoor unit. In heating, refrigerant passes from the reversing valve 59, through the heat exchanger 67, the expansion 27, and then to the outdoor unit.

**[0055]** As will be seen in FIG. 9B, rather than using reversing valves as described hereinabove, check valves maybe substituted to accomplish the same function. Thus, rather than reversing valves, four check vales 71, 72, 73 and 74 are provided. In the cooling mode, the refrigerant passes through the check valve 71, the heat exchanger 67, the expansion device 27, and the check valve 73 to go to the indoor unit, with check valves 72 and 74 being closed. During operation in the heating mode, the check valves 71 and 73 are closed, and refrigerant passes through the check valve 74, the heat exchanger 67, the expansion device 27, and the check valve 72 to pass then to the outdoor unit.

**[0056]** FIG.10 represents a case when two hot sources, high temperature and low temperature sources, are available. A second heater 74 utilizes the high temperature source. The heater 32 utilizes the low temperature source.

**[0057]** A further embodiment is shown in FIG. 11 wherein a multi-stage compressor 76 is provided. After passing through the first stage, the refrigerant passes through a gas cooler 77, and then through the second stage of the two stage compressor 76 before passing to the reversing valve 61 and the condenser 24. In this way, the total compressor power is reduced to thereby improve the thermodynamic efficiency of the compression circuit and therefore that of the total system.

**[0058]** The embodiment of FIG. 12 provides an ejector 78 for boosting the flow of refrigerant vapor to the suction accumulator 66 to thereby improve the thermodynamic efficiencies of the vapor compression circuit and of the total system. The ejector 78 is driven by a high pressure stream along line 79 or, alternatively,

from lines 81 or 82. In this particular embodiment the liquid-to-suction heat exchanger 67 is a mandatory component. The heat exchanger 67 provides completion of evaporation of liquid portion of the refrigerant stream outgoing from the ejector 78.

**[0059]** FIG. 13 embodiment shows a heat pump with an ejector 83 being driven by high pressure refrigerant from line 84 or, alternatively, from line 86. The bi-directional expansion device 87 could be replaced by two one directional expansion devices, i.e. one for the indoor unit and another for the outdoor unit as it was shown above on FIG. 7.

**[0060]** It is known that ejectors improve performance characteristics of vapor compression cycles. The combined vapor compression and vapor expansion cycle improves with a better vapor compression cycle.

**[0061]** Shown in FIG. 14 is an alternative embodiment that includes an economizer cycle which includes an economizing heat exchanger 88, an economizer expansion device 89, and a economizer port 91 leading into a mid-stage of the compressor 23. A further alternative may be that of a multi-stage compressor with intermediate vapor cooling. It is known that economized cycles improve performance characteristics of vapor compression cycles. The combined vapor compression and vapor expansion cycles improves with a better vapor compression cycle.

**[0062]** The FIG. 15 embodiment provides a two-phase expander 92 fluidly interconnected between an inlet to the pump 29 and the reversing valve 59 as shown. Its use tends to increase the cooling effect while recovering additional power to drive the cycle. This, in turn, reduces required pump size and pump power.

**[0063]** Although the present disclosure has been particularly shown and described with reference to a preferred embodiment as illustrated by the drawings, it will be understood by one skilled in the art that various changes in detail made be made thereto without departing from the scope of the disclosure as defined by the claims.

**We Claim:**

1. A thermally activated cooling system comprising:
  - a vapor compression circuit which includes, in serial flow relationship, a compressor, a first heat exchanger, an expansion device and a second heat exchanger;
  - a vapor expansion circuit which includes, in serial flow relationship, a liquid refrigerant pump, a heater, an expander, and said first heat exchanger;
  - said vapor compression circuit and said vapor expansion circuit each having a common refrigerant circulating therethrough as a working fluid; wherein said refrigerant enables supercritical high pressure portion and sub-critical low pressure portion of the vapor compression circuit
  - said compressor having a suction inlet and a discharge outlet, and said expander having an inlet and an outlet, and further wherein said expander outlet is fluidly connected to said compressor outlet to provide a combined flow for circulation of a portion of said working fluid through said first heat exchanger and to said pump wherein said first heat exchanger is so sized, and designed such that the working fluid discharged therefrom is always in a liquid form; and
  - said pump and said expander are so sized and designed that the high pressure portion of said vapor expansion circuit is always super-critical.
  
2. A thermally activated cooling system as set forth in claim 1 where said common refrigerant is CO<sub>2</sub>.
  
3. A thermally activated cooling system as set forth in claim 1 where said common refrigerant is a mixture of CO<sub>2</sub> and propane.
  
4. A thermally activated cooling system as set forth in claim 1 and including a topping heat exchange for causing heat to flow from the expander discharge stream to the stream flowing to the heater.

5. A thermally activated cooling system as set forth in claim 1 and including a liquid-to-suction heat exchanger for causing the flow of heat from the condenser discharge stream to the evaporator discharge stream.

6. A thermally activated cooling system as set forth in claim 1 wherein said expander is a two stage expander and further wherein a second heater is provided between the two stages.

7. A thermally activated cooling system as set forth in claim 1 and including a second vapor compression circuit in parallel with said vapor compression circuit, said second vapor compression circuit having its own expansion device, evaporator, and compressor fluidly interconnected to function with said first heat exchanger.

8. A thermally activated cooling system as set forth in claim 1 and including a plurality of valves for selectively causing the vapor compression system to function as a heat pump.

9. A thermally activated cooling system as set forth in claim 8 wherein said plurality of valves includes two expansion devices, one for the first heat exchanger and another for said second heat exchanger.

10. A thermally activated cooling system as set forth in claim 8 wherein said plurality of valves includes a single bidirectional expansion device which is selectively operated to conduct the flow of refrigerant to either said first or second heat exchanger.

11. A thermally activated cooling system as set forth in claim 8 wherein said plurality of valves comprises a plurality of check valves that are selectively operated to conduct the flow of refrigerant to either the first or second heat exchanger.

12. A thermally activated cooling system as set forth in claim 1 and including a second heater connected in serial flow relationship with said heater.
13. A thermally activated cooling system as set forth in claim 1 wherein said compressor comprises a multi-stage compressor, and further including a gas cooler operably connected between said stages.
14. A thermally activated cooling system as set forth in claim 1 wherein said compression circuit includes an ejector for boosting the flow of refrigerant to said compressor.
15. A thermally activated cooling system as set forth in claim 14 wherein a refrigerant stream intended for cooling duty is split into two portions; said ejector is powered by one portion of said refrigerant stream and ejects another portion of said refrigerant stream processed in said evaporator and then in said liquid-to-suction heat exchanger.
16. A thermally activated cooling system as set forth in claim 14 wherein said compression circuit includes a suction accumulator; a refrigerant stream intended for cooling duty powers said ejector ejects liquid portion of said stream collected in said suction accumulator and processed in said evaporator.
17. A thermally activated cooling system as set forth in claim 1 wherein said vapor compression circuit includes an economizer operably connected therewith.
18. A thermally activated cooling system as set forth in claim 1 and including a two phase expander fluidly interconnected between said condenser and said evaporator.
19. A thermally activated cooling system as set forth in claim 1 wherein said expander, said pump, and said compressor have a common shaft

20. A thermally activated cooling system as set forth in claim 1 wherein a power generator and said expander have a common shaft and said power generator powers said pump and said compressor.

21. A thermally activated cooling system as set forth in claim 1 wherein a power generator, said expander, and said pump have a common shaft and said power generator powers said compressor.

22. A thermally activated cooling system as set forth in claim 1 wherein a power generator, said expander, and said compressor have a common shaft and said power generator feeds said pump.

23. A thermally activated cooling system as set forth in claim 18 wherein said expander, said pump, and said compressor have a common hermetic casing.

24. A power generation vapor expansion circuit which includes, a power generator and, in serial flow relationship, a liquid refrigerant pump, a heater, an expander, and a heat exchanger;

a refrigerant circulating therethrough as a working fluid wherein said refrigerant enables supercritical high pressure portion and sub-critical low pressure portion of the vapor expansion circuit;

said first heat exchanger is so sized, and designed such that the working fluid discharged therefrom is always in a liquid form; and

said pump and said expander are so sized and designed that the high pressure portion of said vapor expansion circuit is always super-critical.

25. A power generation vapor expansion circuit as set forth in claim 24 wherein said refrigerant is CO<sub>2</sub>.

26. A thermally activated cooling system as set forth in claim 24 where said common refrigerant is a mixture of CO<sub>2</sub> and propane.

27. A power generation vapor expansion circuit as set forth in claim 24 and including a topping heat exchanger for causing heat to flow from the expander discharge stream to the stream flowing to the heater.

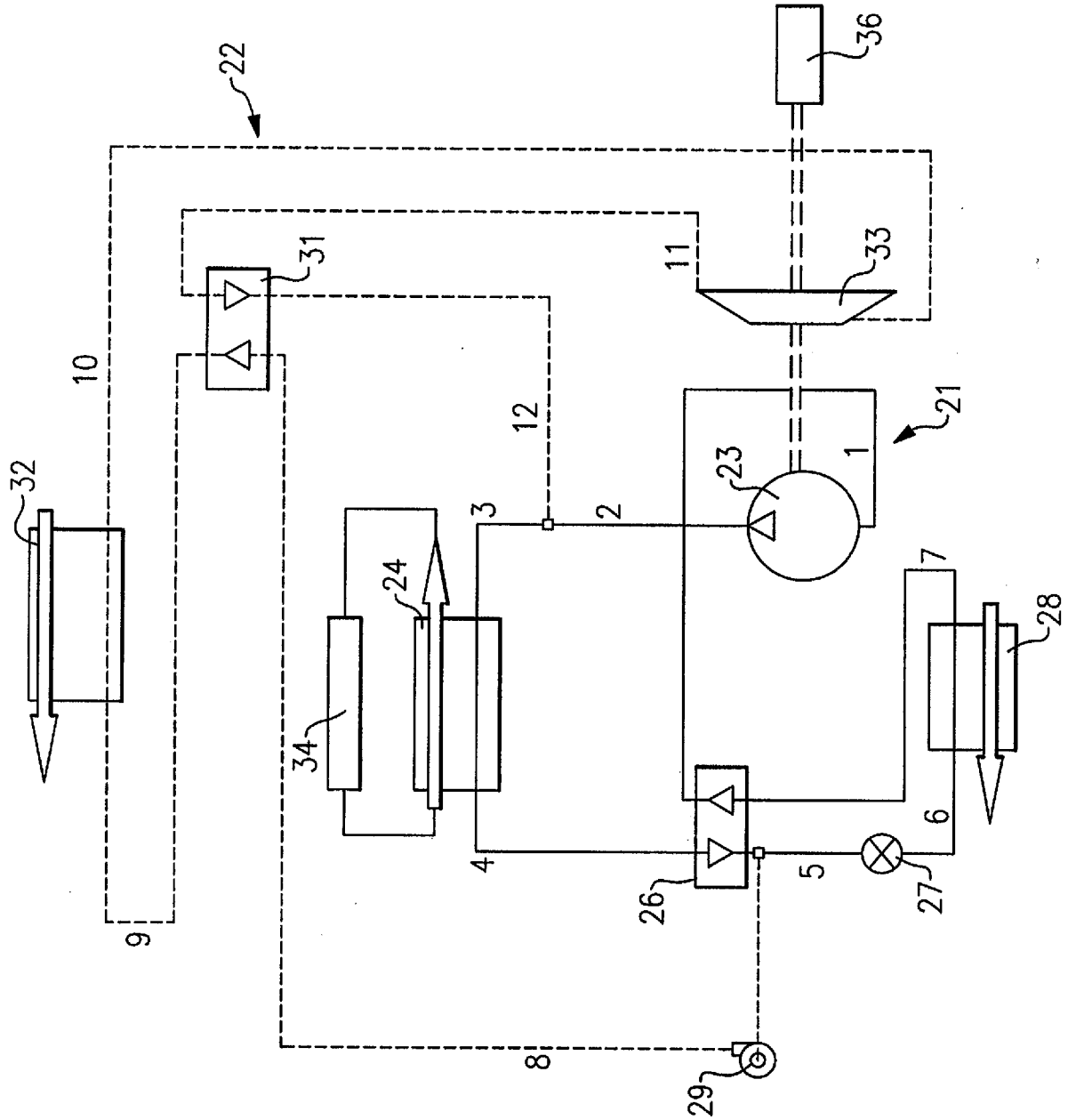
28. A power generation vapor expansion circuit as set forth in claim 24 wherein said expander is a two stage expander and further wherein a second heater is provided between the two stages.

29. A power generation vapor expansion circuit as set forth in claim 24 and including a second heater connected in serial flow relationship with said heater.

30. A power generation vapor expansion circuit as set forth in claim 24 wherein said power generator, said expander, and said pump have a common shaft.

31. A power generation vapor expansion circuit as set forth in claim 24 wherein said power generator, said expander, and said pump have a common hermetic casing.

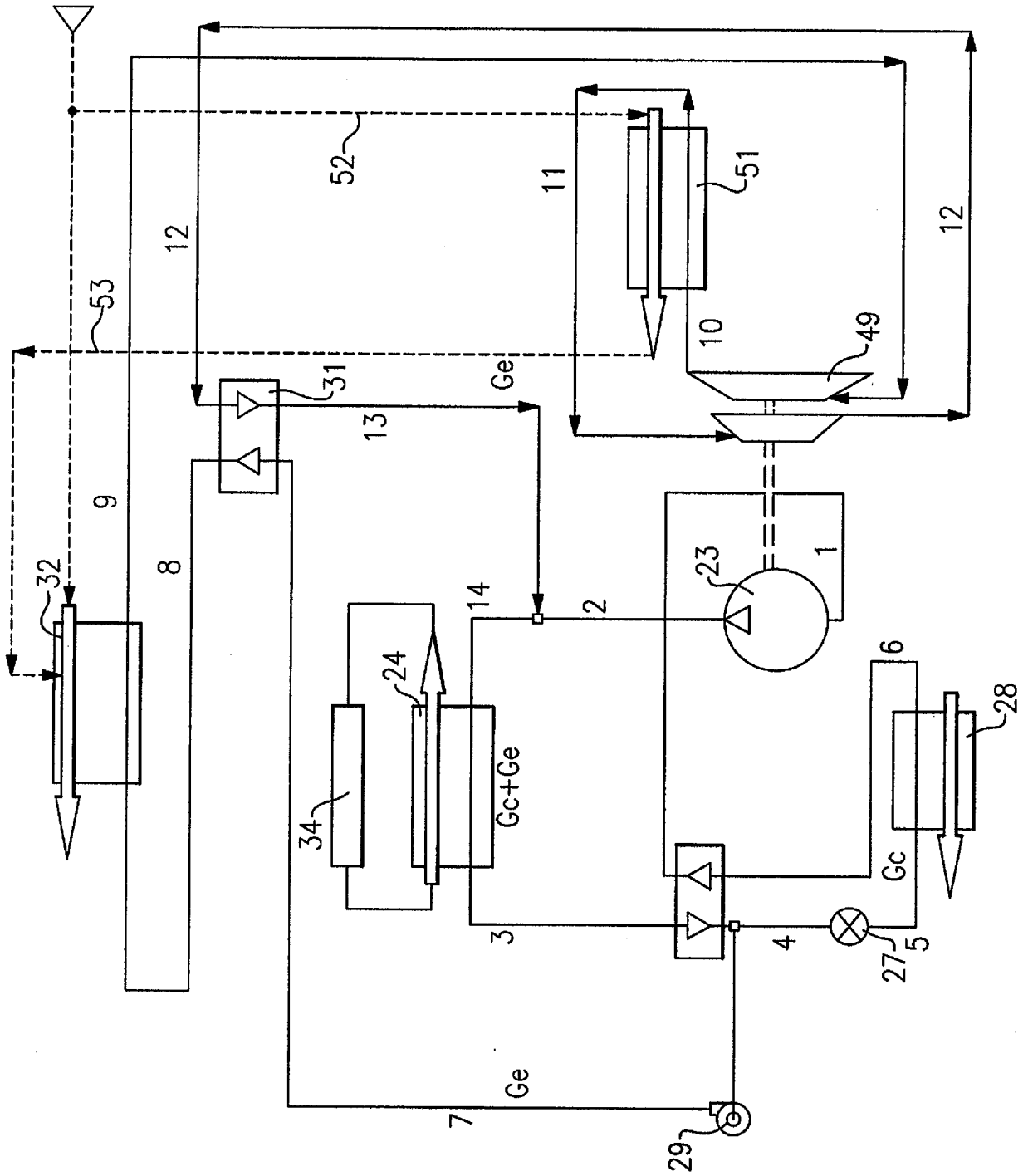
32. A power generation vapor expansion circuit as set forth in claim 24 wherein said power generator powers a refrigerating system.



**FIG. 1**

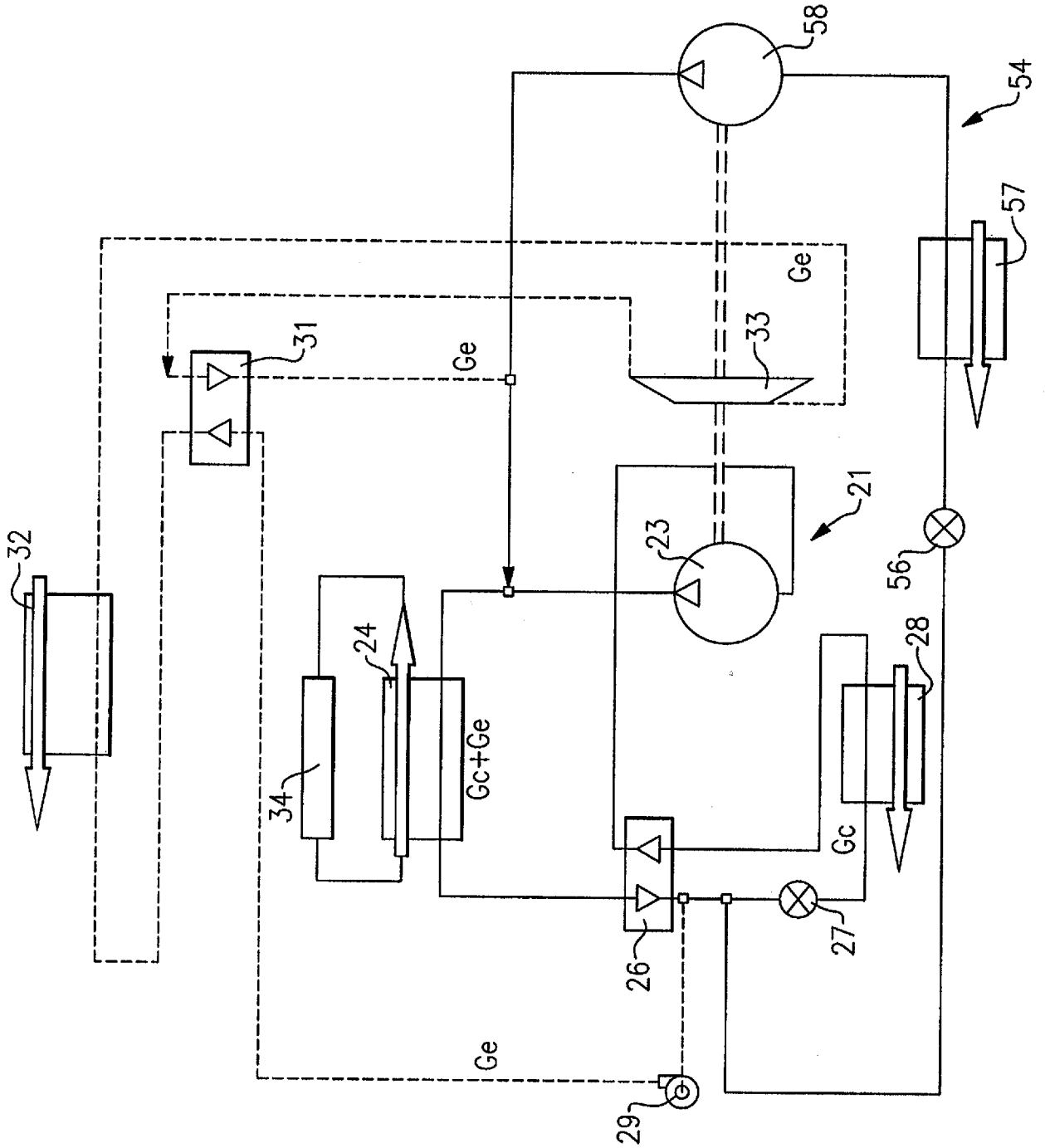




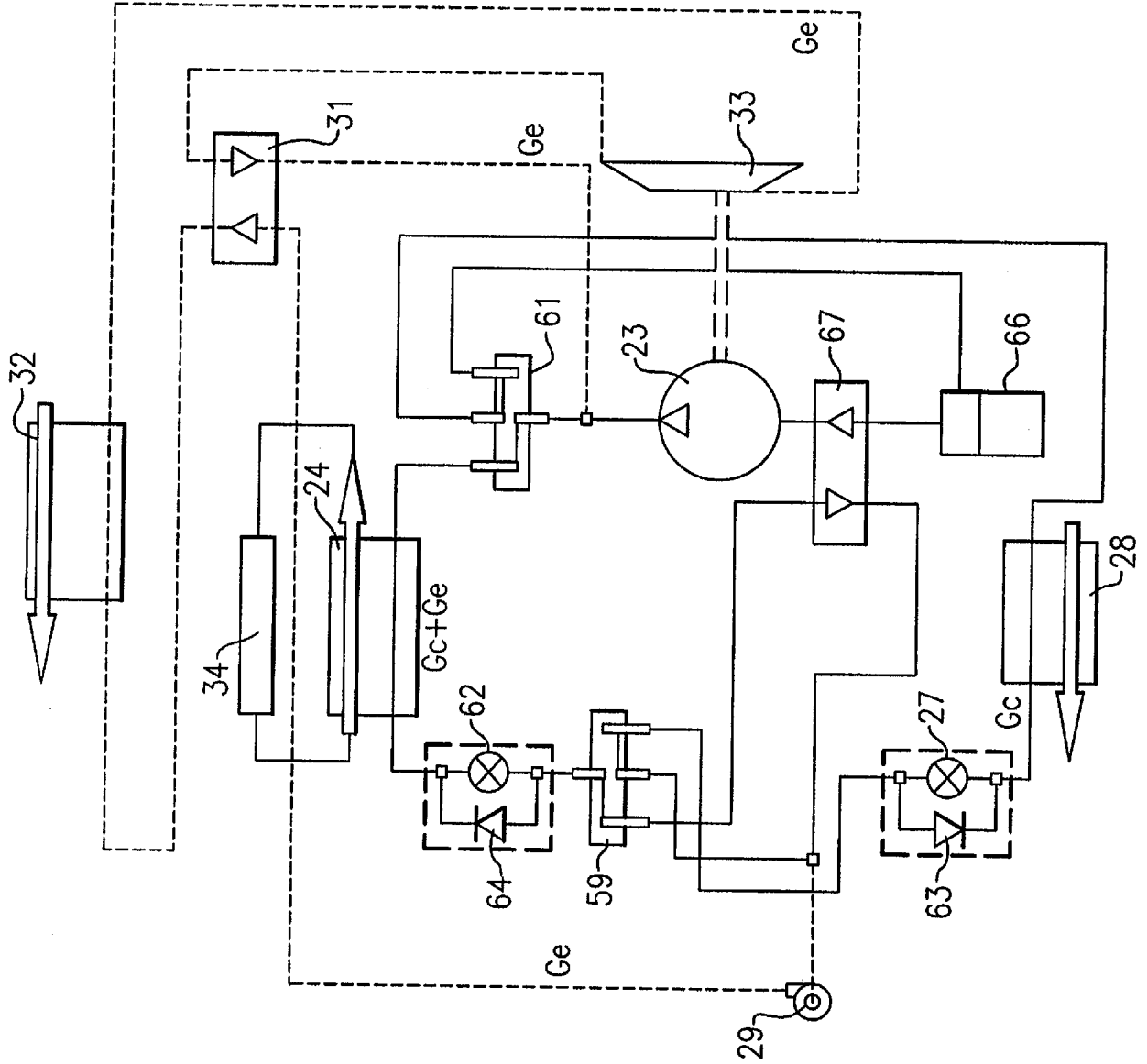


**FIG.4**

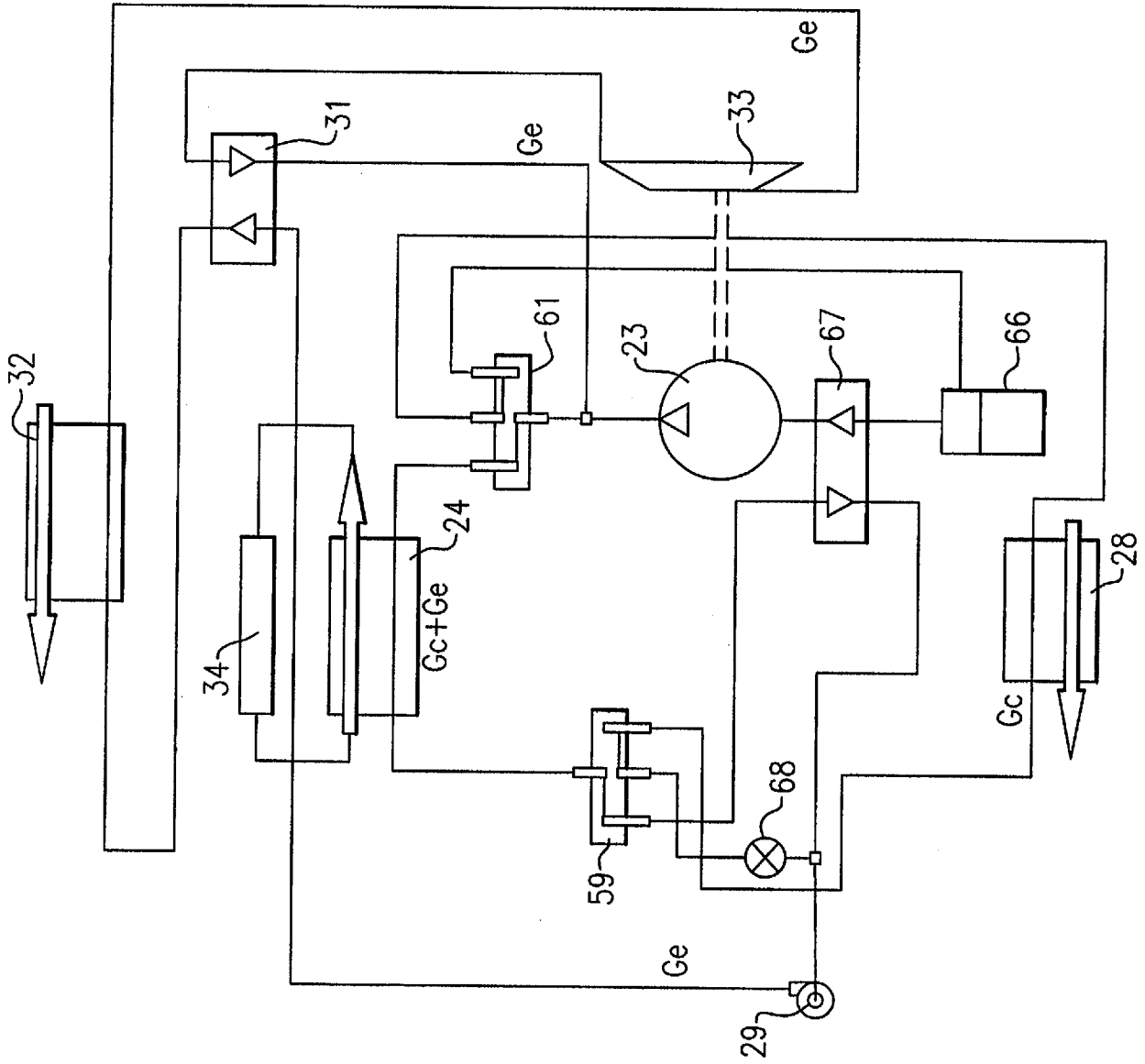




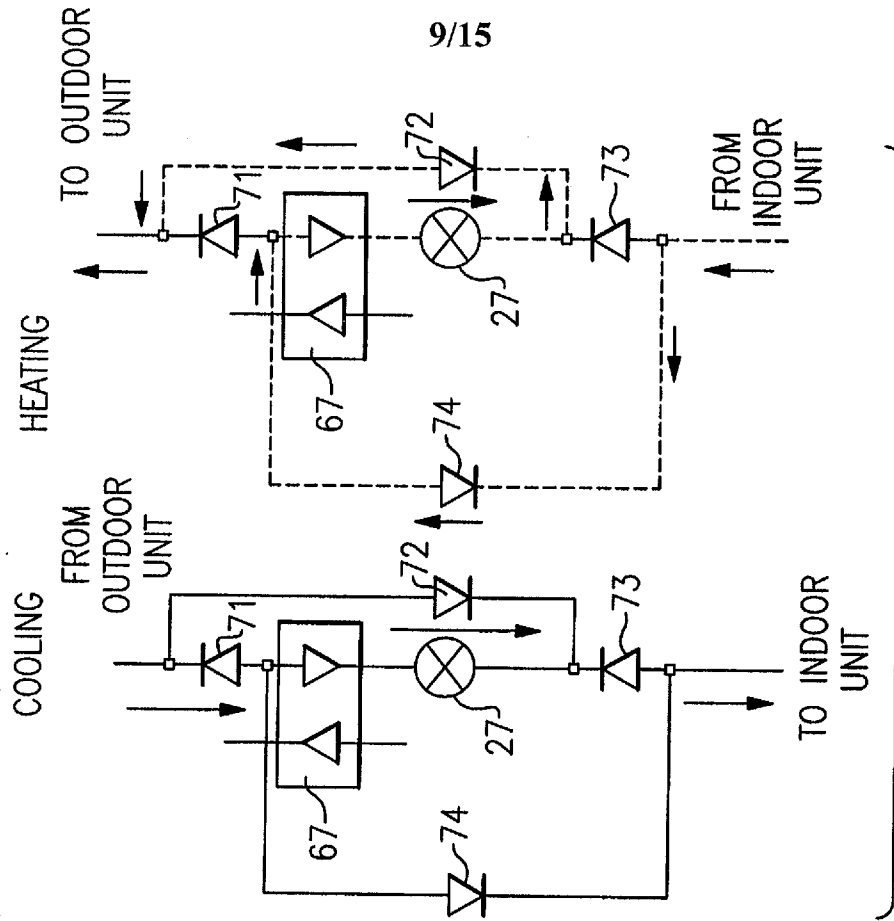
**FIG.6**



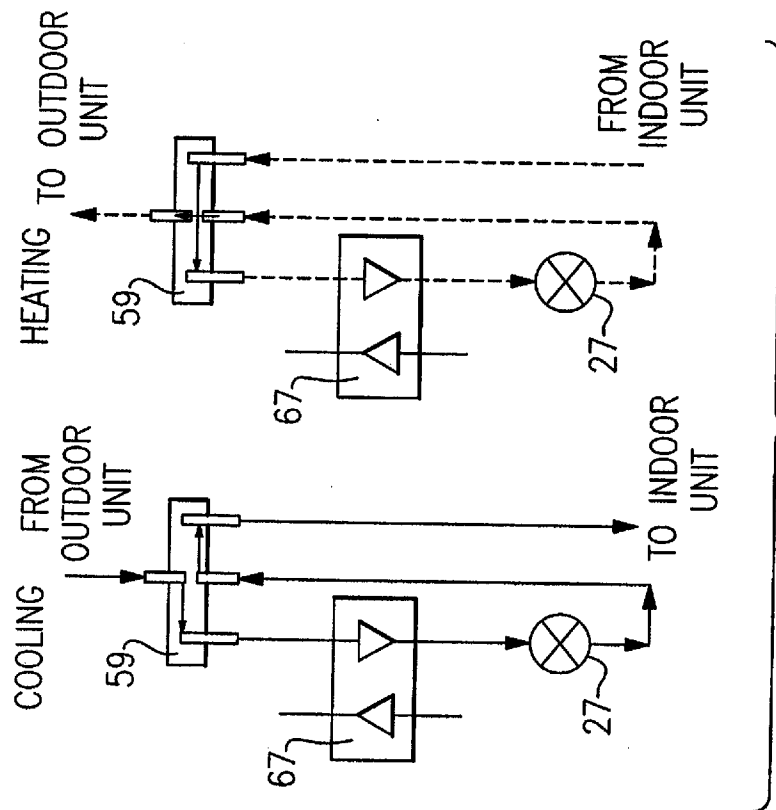
**FIG. 7**



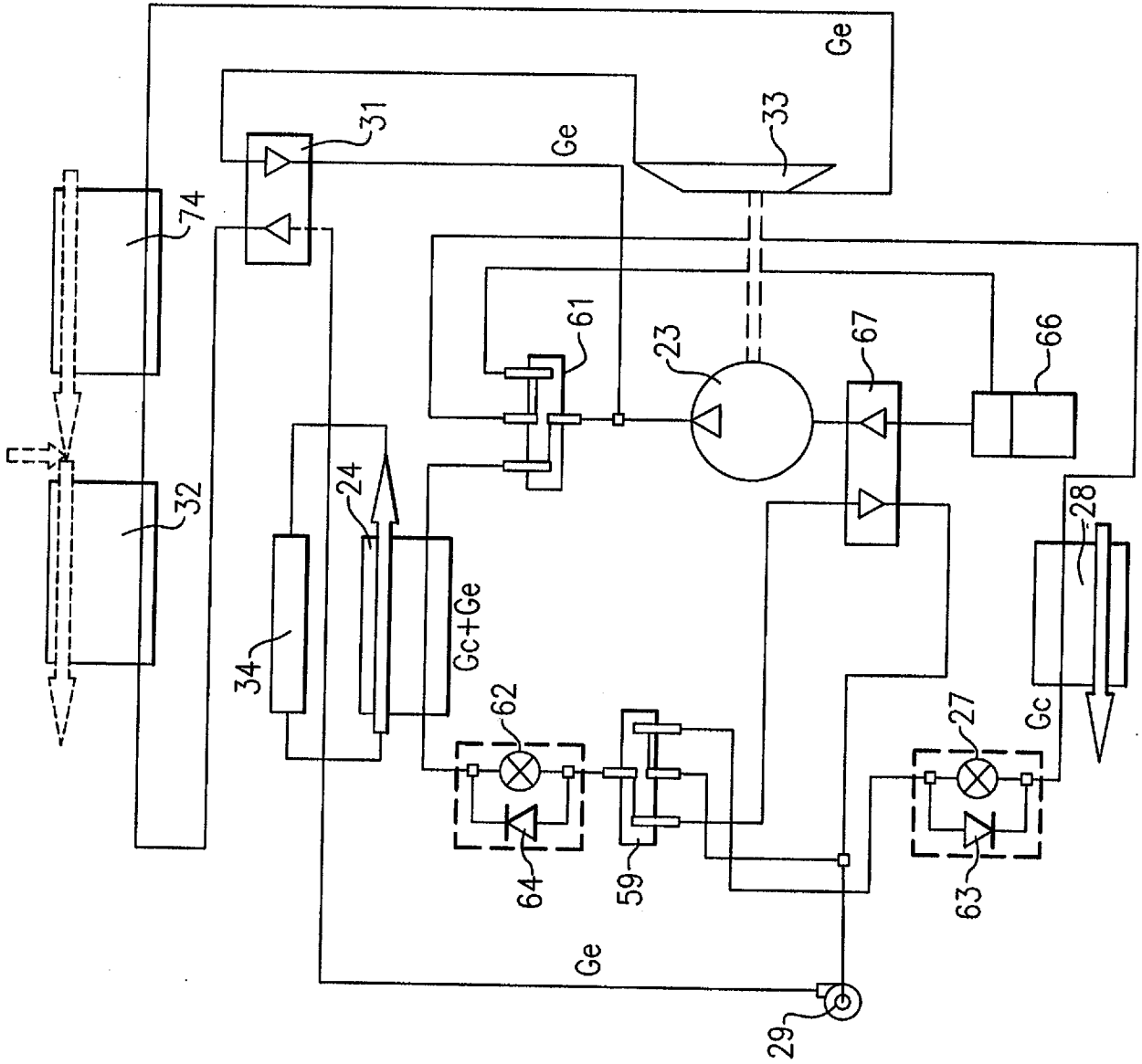
**FIG. 8**



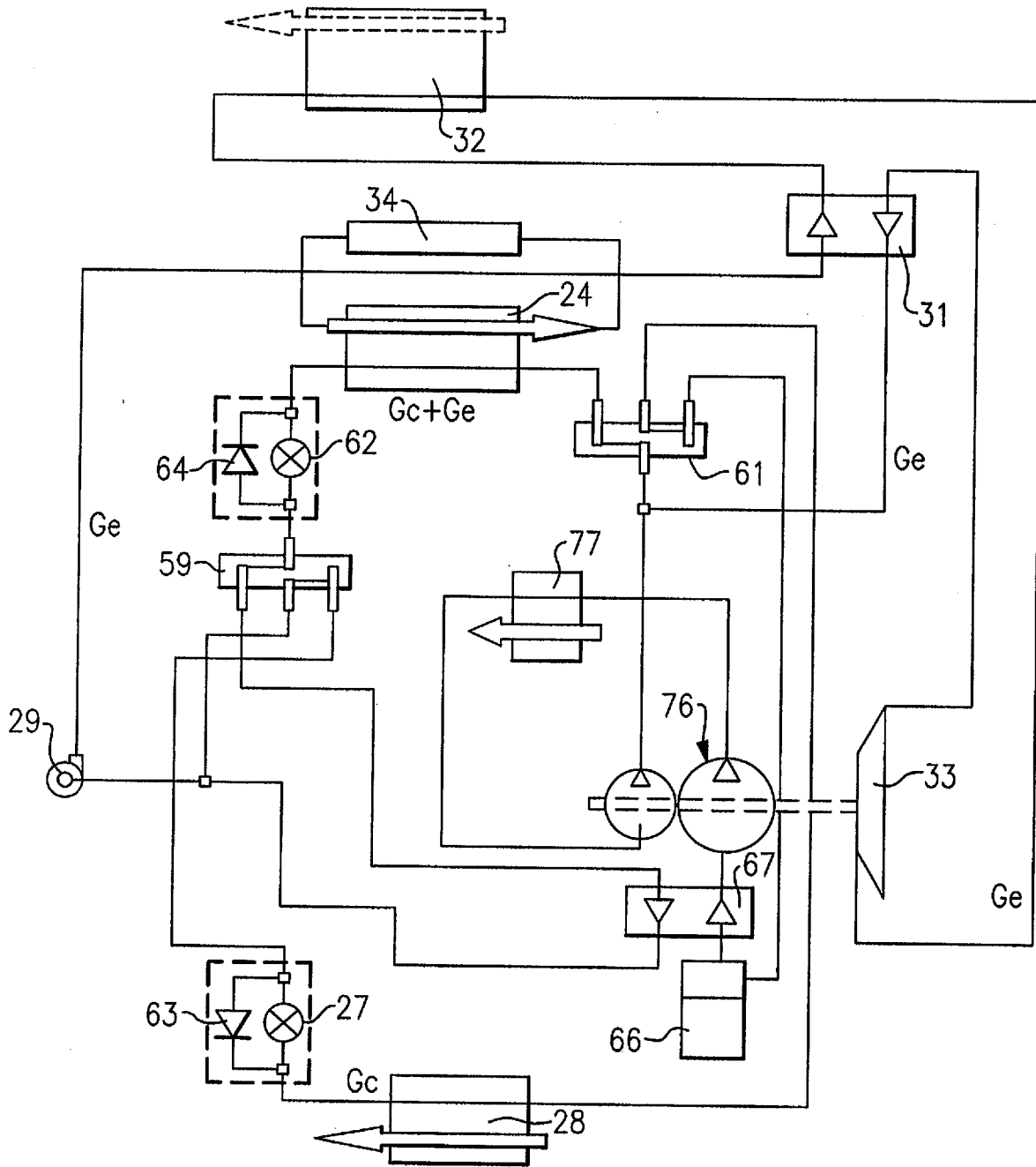
**FIG.9B**



**FIG.9A**



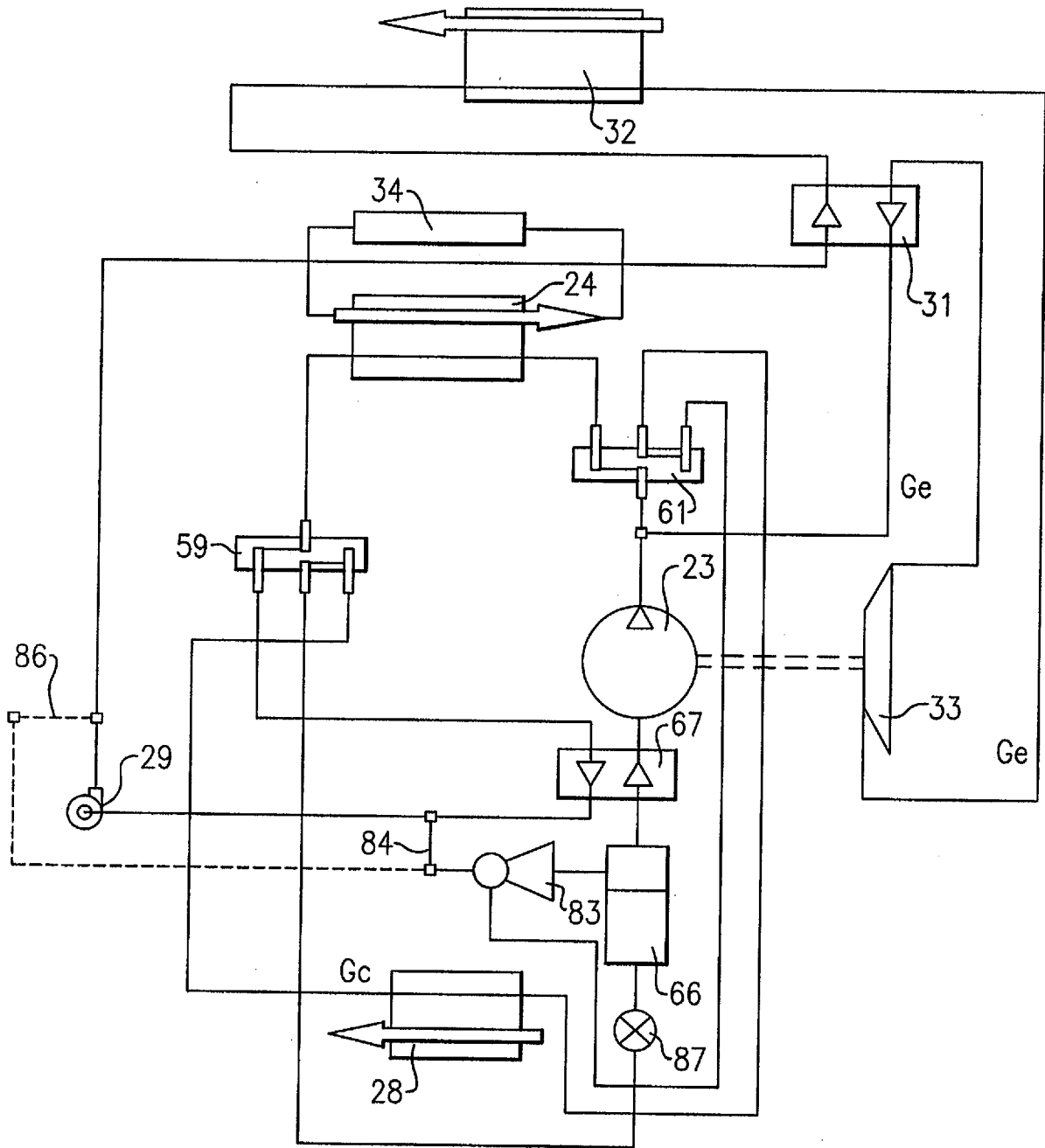
**FIG.10**



**FIG. 11**

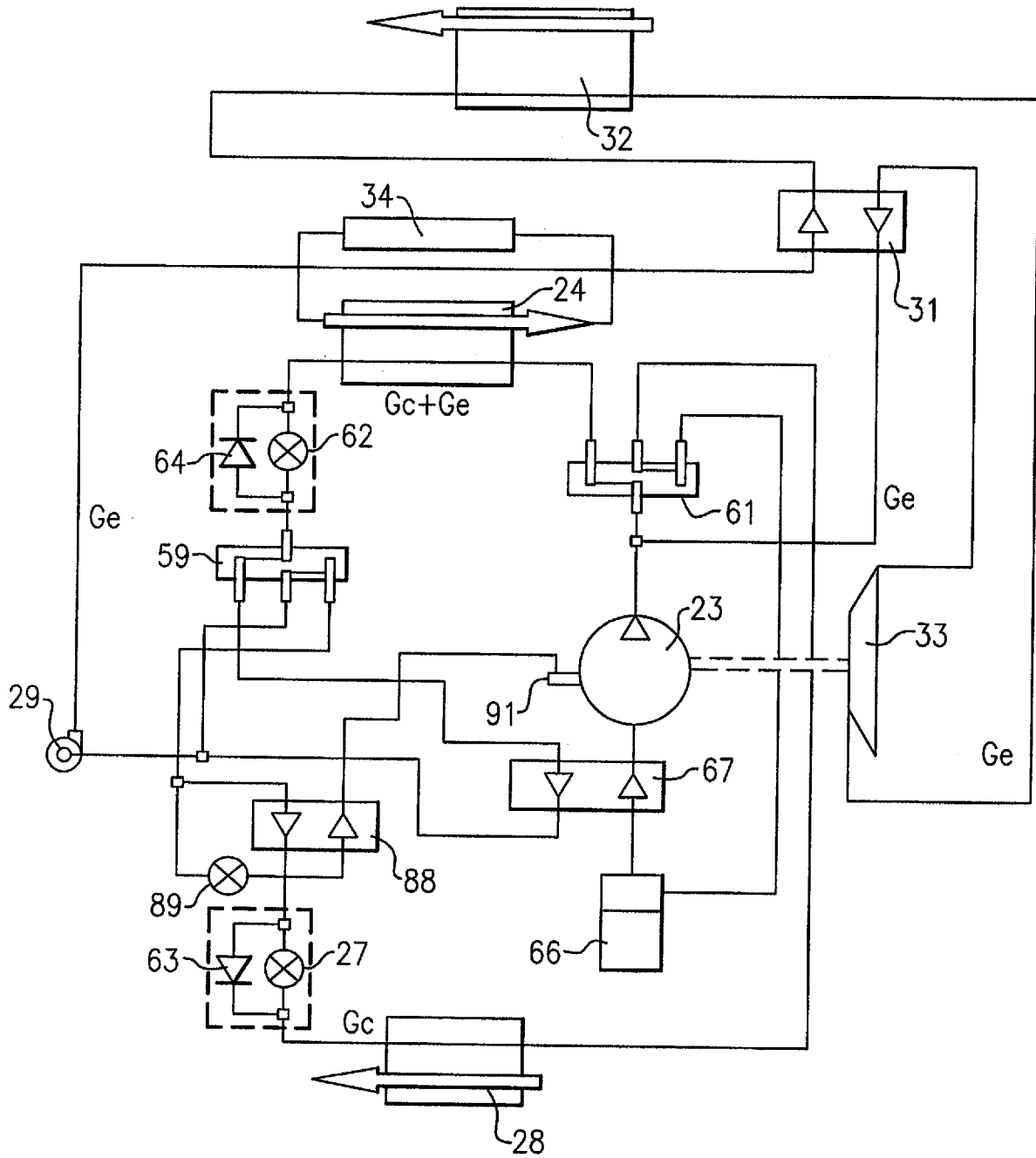


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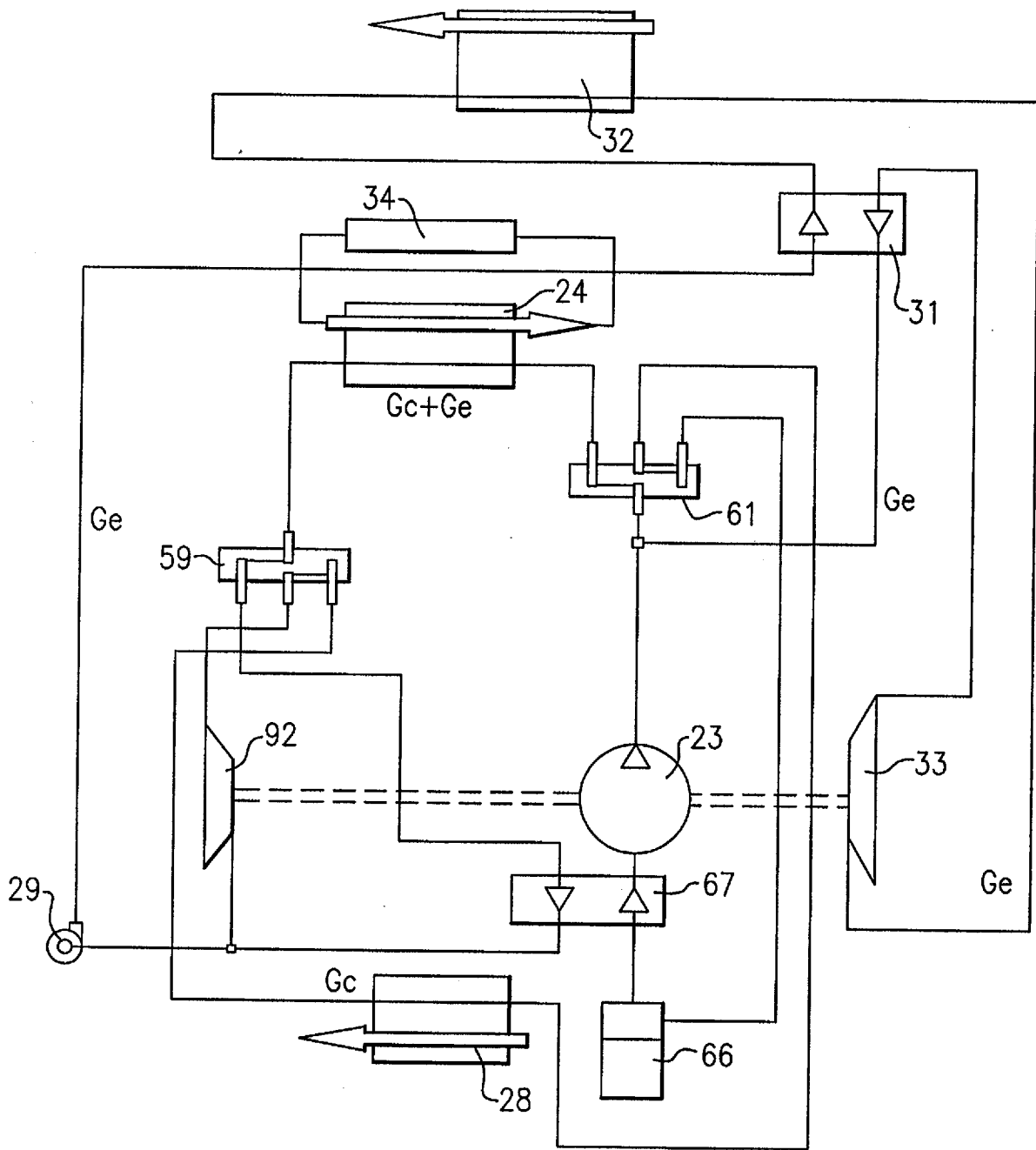
**FIG.13**

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**FIG. 14**

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**FIG. 15**