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(54) Refrigeration system with pulsed ejector and vertical evaporator

(57) A refrigeration system involving a nozzling device (10) which fully opens and closes to create high velocity bursts of unrestricted refrigerant flow through the system, wherein a pulsed ejector (11) is coupled with the nozzling device (10) and refrigerant from the evaporator (36) is recirculated back to the pulse ejector (11), wherein the evaporator (36) is vertical and its tubes (40) are substantially entirely wetted interiorly during operation, wherein the condenser (44) is vertical and its tube surface area are substantially entirely functioning as condensing heat transfer surfaces, wherein an ejector (50) is associated with the condenser (44) and refrigerant from the condenser (44) is recirculated back to the condenser ejector (50), and also wherein internal ejector recirculation effects occur within the evaporator (36) and condenser (44) as a result of pulse flow events.

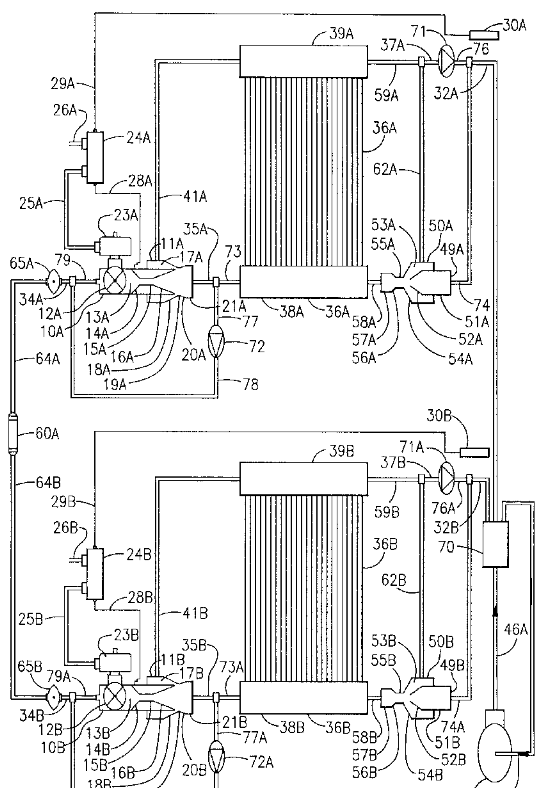
FIG.2

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Description

This application is a continuation-in-part of co-pending application entitled "Non-Steady-State Self-Regulating Intermittent Flow Thermodynamic System" filed March 25, 1993 and given Serial No. 08/036,901, the applicant of which is one of the applicants of the present application, and co-pending continuation-in-part application entitled "Circulation of Oil in Refrigeration Systems with Immiscible Refrigerant-Oil Combinations" filed June 20, 1994 and given Serial No. 08/262,680, the applicants of which are the same as those of the present application.

Background of the Invention

This invention concerns an extension of the technology described in the aforementioned related co-pending applications, particularly with respect to inclusion of pulsed ejectors in combination with nozzling devices which generate high velocity bursts of substantially unrestricted refrigerant flow throughout a refrigeration system. United States Patent No. 5,240,384 describes a refrigeration system utilizing ejectors having partitioned mixing tubes or diffusers to create multiple flow passages in each of which the primary high velocity fluid jet stream effectively pulses. However that pulsation is an internal process within the ejector body and it does not change the conventional steady-state continuous fluid flow throughout the system. In contrast the concept upon which the present invention is based is that of a substantially non-steady-state intermittent flow throughout the entire system.

Refrigeration cycles typically include a vapor compressor, a condenser which changes vapor to liquid as it gives off heat, an expansion device reducing the refrigerant pressure, and an evaporator changing liquid to vapor as it provides cooling. Expansion devices of the prior art are inefficient because of energy loss during the throttling process. Also optimum use of evaporator surface area is limited when conventional expansion devices are used because of the requirement of substantial superheat of refrigerant vapor by a portion of the evaporator. For highest efficiency the entire surface area of an evaporator should be fully wetted but with conventional expansion devices full wetting of the evaporator surface is not possible because the compressor must be protected from wet gas or liquid.

Evaporator and condenser heat exchange tubing of the prior art is typically of serpentine configuration with horizontal tubing having a multiplicity of U-bends. An appreciable pressure drop is inherent in such heat exchanger designs in order to move oil along with the refrigerant and achieve appropriate heat transfer rates. High pressure drops lower overall system efficiency and increase the compressor pumping power requirements. Also, horizontal evaporator and condenser tubing can result in laminar non-turbulent flow with associated low heat transfer rates. Condensed liquid tends to fill the lower half of each horizontal run of tubing thereby reducing the available surface area left for heat transfer. Lengthy runs of heat exchange tubing with a multiplicity of U-bends cannot be arranged vertically because if they were oil flow would be impeded and that could lead to compressor failure.

Defrosting of conventional evaporator heat exchanger tubes in ordinary refrigeration systems is a lengthy energy-intensive process. It is one of the objects of the present invention to render a refrigeration system reversible so that it can function alternatively as a heat pump and dispose quickly and efficiently of accumulated evaporator frost.

Metering systems are known in the prior art for steady-state throttling of refrigerant flow. Typically the mechanical throttling restriction is modulated in response to either temperature or pressure changes in the system related to superheat of the refrigerant vapor. In contrast the technology of self-regulation of a thermodynamic system described in the earlier of the aforementioned related applications is extended by the present invention to the sensing of both pressure and temperature to provide feedback to a setpoint switch which continuously modulates the operating setpoint. As the setpoint modulates, the magnitude of the sensed temperature and sensed pressure at each moment of the system operation can vary with changing environmental conditions. One object of the present invention is to rely upon no fixed temperature or pressure setpoint. Momentary temperature and pressure operating setpoints are relational and their magnitude varies as the system self-regulates with changing environment conditions.

Summary of the Invention

A refrigeration system is the subject of this invention wherein a refrigerant circulates from an evaporator to a compressor to a condenser and thence back to the evaporator through a nozzling device. That device includes a nozzle and a valve automatically fully opened and closed in a binary fashion to create accelerated intermittent high velocity bursts of substantially unrestricted refrigerant flow from an outlet of the nozzle through the system. The system of the invention comprises a pulsed ejector having a pulsed ejection suction port into which the refrigerant is directed form the nozzle outlet. A pulsed ejector suction conduit connects the pulsed ejector suction port with refrigerant flow between the evaporator and the compressor. The evaporator includes a plurality of substantially vertical evaporator tubes interconnected in parallel by lower and upper evaporator headers. The nozzling device and pulsed ejector when the nozzling device is open carries the refrigerant upwardly through the evaporator from the lower to the upper evaporator headers so that liquid refrigerant wets substantially all inner surfaces of the evaporator tubes. The pulsed ejector suction conduit recirculates refrigerant leaving the upper header of the evaporator back to the pulsed ejector when the nozzling device is open.

Sensing means may be included for sensing pressure and temperature of the refrigerant in the system to open fully or close the valve in response to a change in at least one of the system pressure and temperature. The sensing means may comprise a thermostatic bulb sensing refrigerant temperature in the system and a pressure tap sensing refrigerant pressure in the system to infinitely vary a setpoint at which the nozzling device valve opens and closes. As a consequence momentary temperature and pressure operating setpoints are relational and the magnitude of the sensed temperature and sensed pressure at each moment of system operation varies as the system self-regulates with changing environment conditions.

It is preferred that the condenser comprise a plurality of substantially vertical condenser tubes interconnected in parallel by lower and upper condenser headers. A condenser ejector may be included having a condenser ejector suction port for directing refrigerant vapor from the compressor downwardly through the condenser tubes from the upper to the lower condenser headers. A condenser ejector suction conduit may connect the condenser ejector suction port with refrigerant flow from the lower condenser header. The condenser ejector suction conduit recirculates refrigerant leaving the lower condenser header back to the condenser ejector when the nozzling device valve is open.
Recirculated refrigerant intimately mixing with the hot gas entering the ejector from the compressor accomplishes a desuperheating of the superheated hot gas at heat transfer coefficients greater than that of the 'dry-wall' desuperheating of hot gas in the condensers of the prior art.

The condenser ejector suction port may direct refrigerant vapor from the compressor to the lower condenser header. The nozzling device and pulsed ejector when the nozzling device valve is open carries the refrigerant upwardly through the condenser tubes from the lower to the upper condenser header so that liquid refrigerant condenses on inner surfaces of the condenser tubes and flows downwardly in counterflow relation to refrigerant vapor carried upwardly through the condenser tubes.

In one embodiment of the invention the upper condenser header is of dead-end form. Hot gas from the compressor can enter the lower header, which also functions as a refrigerant reservoir. The liquid refrigerant condensing on inner surfaces of the condenser tubes then flows downwardly in counterflow relation to vapor carried upwardly through the condenser tubes to the upper dead-end condenser header. Hot gas bubbling up through the condensed liquid in the lower header is effectively completely desuperheated prior to entering the vertical condenser tubes, leaving the entire tube surface area available for condensing heat transfer.

Means may be included for selectively operating the system alternatively as a heat pump system. This may include reversing valve means for reversing the direction of refrigerant flow from a refrigeration mode to a heat pump mode. The condenser in the refrigeration mode then functions as an evaporator in the heat pump mode and the evaporator in the refrigeration mode then functions as a condenser in the heat pump mode when the reversal of direction of flow occurs. In the heat pump mode the system may rapidly melt frost accumulated on the evaporator tubes during operation in the refrigeration mode.

The reversing valve means may be provided at the discharge of the compressor and the outlet of the pulsed ejector for reversing the direction of refrigerant flow from the refrigeration mode to the heat pump mode while utilizing a single nozzling device and pulsed ejector.

The nozzling device and pulsed ejector may be duplicated as first and second nozzling devices and pulsed ejectors. The reversing valve means for reversing the direction of refrigerant flow from the refrigeration mode to the heat pump mode may be without check valves. The first nozzling device and pulsed ejector meters refrigerant flow in one direction and the second nozzling device and pulsed ejector meters flow in the opposite direction. The condenser in the refrigeration mode then functions as an evaporator in the heat pump mode and the evaporator in the refrigeration mode then functions as a condenser in the heat pump mode when the first nozzling device and pulsed ejector cease operation and the second nozzling device and pulsed ejector meter flow in the opposite direction.

There may be by-pass reservoir means included in the system for withdrawing refrigerant from the system in response to a reduction in system superheat and returning refrigerant to the system in response to an increase in system superheat. As system superheat rises and falls refrigerant alternately enters and leaves the reservoir means and alternately enters and leaves the system.

The system may be an absorption system with an absorbent fluid and a refrigerant each circulating in its own flow circuit. The nozzling device and pulsed ejector are then duplicated as an absorber nozzling device and an absorber pulsed ejector and a refrigerant nozzling device and a refrigerant pulsed ejector. The refrigerant may pass from the refrigerant evaporator to the refrigerant pulsed ejector before entering the absorber, so that the refrigerant pulsed ejector is in series with respect to the absorber. Alternatively, the refrigerant may circulate directly from the evaporator to the absorber without passing through the refrigerant pulsed ejector so that the refrigerant pulsed ejector is in parallel with respect to the absorber.

Also contemplated as part of the invention are certain sub-combinations with the refrigeration system wherein the refrigerant circulates from evaporator to compressor to condenser and thence back to the evaporator through a nozzle device including a nozzle and a valve automatically fully opened and closed in a binary fashion to create accelerated intermittent high velocity bursts of substantially unrestricted refrigerant flow from an outlet of the nozzle through the system. One such sub-combination comprises a thermostatic bulb sensing refrigerant temperature in the system and a pressure tap sensing refrigerant pressure in the system to infinitely vary a setpoint at which the nozzling device opens and closes. Another such sub-combination comprises the condenser with its plurality of substantially vertical condenser tubes interconnected in parallel by lower and upper condenser headers and the condenser ejector with a condenser ejector suction port for directing refrigerant from the compressor upwardly through the condenser tubes from the lower to the upper condenser headers. A condenser ejector suction conduit may connect the condenser ejector suction port with refrigerant flow from the upper condenser header. The condenser ejector suction conduit recirculates refrigerant leaving the upper condenser header back to the condenser ejector when the nozzling device valve is open. When the nozzling device valve is open carrying the refrigerant upwardly through the condenser tubes from the lower to the upper condenser headers, refrigerant liquid then condenses on inner surfaces of the condenser tubes and flows downwardly in counterflow relation to refrigerant vapor carried upwardly through the condenser tubes. In this latter sub-combination the upper condenser header may be a dead-end header.

Also included in the invention is the pulse velocity induced enhancement of heat transfer within the condenser and evaporator without associated increases in heat exchanger pressure drop or flow losses. A pulse flow event initiates external ejector recirculation flows for each ejector. Due to the design of the condenser and evaporator heat exchangers, the pulse velocity of a pulse flow event initiates internal ejector-related recirculation effects within the condenser and evaporator heat exchangers, further enhancing heat transfer.

Also included in the invention is a novel phenomenon resulting from the pulse high velocity, high impulse mass flow rate flow events. The high velocity pulse flows produce a potentially non-equilibrium process characterized by velocity induced subcooling of condenser liquid. The high velocity, high impulse mass flow rate flows can result in the flashing of condenser liquid to vapor. The increased liquid subcooling that results increases overall system efficiency and cooling capacity.

Also included in the invention is a thermodynamic process wherein a heat exchange fluid is circulated and wherein a method is provided of continual thermodynamic efficiency self-optimization in real time as energy is exchanged in the process with an external environment. In this method the heat exchange fluid is directed through a valve and nozzle. The pressure of the heat exchange fluid in the system is sensed and the temperature of the heat exchange fluid is sensed. The sensed temperature is converted to an equivalent sensed pressure. The valve is automatically fully opened or closed in a binary fashion in response to a change in the relation between the first pressure and the sensed temperature thus permitting substantially unrestricted bursts of fluid flow through the valve and permitting acceleration of the intermittent bursts of fluid flow by the nozzle. The opening and closing of the valve in this method functions in a mechanical feedback loop utilizing internal pressure information and internal temperature information to self-regulate the opening and closing of the valve and flow through the nozzle.

Brief Description of the Drawings

FIG. 1 is a schematic system according to the invention showing a continuously self-regulating setpoint switch for regulating a pulsed ejector at the evaporator inlet in parallel arrangement with the compressor and with a simple condenser ejector for condenser overfeed;

FIG. 2 is a schematic similar to FIG. 1 with the system split for alternative heat pump operation;

FIG. 3 is a schematic similar to FIG. 2 but in a unitary rather than split configuration utilizing double pulsed ejectors;

FIG. 4 is a schematic of alternative heat pump configuration utilizing a single pulsed ejector;

FIG. 5 is a schematic of a system according to the invention including refrigerant by-pass storage and a dead-ended vertical condenser;

FIG. 6 is a schematic of a system of the invention for_absorption refrigeration with a series ejector circuit for the absorbent flow pulsed ejector; and

FIG. 7 is an absorption system similar to FIG. 6 with a parallel ejector circuit for the absorbent flow pulsed ejector.

Description of Preferred Embodiment

The term "refrigeration system" as used herein is to be understood as including an air-conditioning system and a heat pump system.

The term "thermodynamic system" as used herein is to be understood as including a refrigeration, air-conditioning system, and a heat pump system.

The term "thermodynamic fluid" as used herein is to be understood to mean a general fluid, including homogeneous, heterogeneous, mixture, non-homogeneous, non-heterogeneous, fraction, component, and blend, as descriptors of the fluid.

The term "pulsed ejector" as used herein is to be understood to mean a nozzling device composed with an ejector body. A nozzling device is fundamentally composed of a valve and a nozzle that are substantially unrestricting to fluid flow when the valve element is open, the nozzle serving to accelerate fluid flow to the maximum attainable velocity. An ejector body is fundamentally composed of a section communicating with the nozzle of the nozzling device, a suction section, and a discharge section.

The term "simple ejector" or "simple pulsed ejector" as used herein is to be understood to mean an ejector composed of a nozzle, a suction section, and a discharge section. Said ejector experiences pulse fluid flow due to the action of a nozzling device within a system without having the valve of said nozzling device situated immediately upstream of said ejector. In general, the nozzle of a nozzling device and the nozzle of said ejector are distinct and seperate entities.

The concept of internal recirculation flows within a heat exchanger as a result of a high velocity pulse flow entering the heat exchanger is a fundamental extension of the dynamical fact that momentum transfer and hence imparted fluid flow will occur whenever fluid streams of different velocities interact. Said momentum transfer and imparted fluid flow phenomenon are the basis upon which ejectors function.

The term "parallel" as used herein with respect to the orientation of an ejector within a system to the other system elements is to be understood as generally utilized with respect to electrical systems and electrical current flow. The analagous ejector flow being the flow from the ejector suction port to the ejector discharge port. For example, a "parallel" ejector orientation with respect to a heat exchanger and a compressor generally increases fluid flow through the heat exchanger. The ejector recirculation flow will impart a greater flow through the heat exchanger than than the flow through the compressor, with the heat exchanger flow provided by the action of both the compressor and the ejector. Were the ejector to cease to function, the heat exchanger would still experience the compressor flow. Both the primary fluid flow entering the ejector through the nozzle and the ejector suction flow is experienced by the heat exchanger as flow through the heat exchanger. The combined nozzle and suction flow leaving the ejector as the ejector discharge flow is experienced by the heat exchanger as flow through the heat exchanger.

The term "series" as used herein with respect to the orientation of an ejector within a system to the other system elements is to be understood as generally utilized with respect to electrical systems and electrical current flow. The analagous ejector flow being the flow from the ejector suction port to the ejector discharge port. For example, a "series" ejector orientation with respect to a heat exchanger and a compressor generally limits the fluid flow through the heat exchanger to that provided by the ejector suction flow. The ejector suction flow can impart less flow through the heat exchanger than than the flow through the compressor. Were the ejector to cease to function, the heat exchanger would not experience the compressor flow. The primary fluid flow entering the ejector through the nozzle is not experienced by the heat exchanger as flow through the heat exchanger. Only the suction flow component of the combined nozzle and suction flow leaving the ejector as the ejector discharge flow is experienced by the heat exchanger as flow through the heat exchanger. The "no ejector suction flow, no heat exchanger flow" aspect of "series" ejector piping has severely limited the use of ejectors in refrigeration systems where the ejector efficiency and performance

In the refrigeration system shown in FIG. 1 a nozzling device 10 is associated with an ejector body 11 to form a pulsed ejector. The nozzling device 10 includes a valve element 12, a valve-nozzle transition section 13, and a nozzle composed of converging nozzle inlet section 14, nozzle throat 15, and a diverging nozzle outlet section 16. For a complete description of such a nozzling device and its operation see the aforementioned copending application entitled Non-Steady-State Self-Regulating Intermittent Flow Thermodynamic System. The ejector body 11 is composed of an ejector suction port 17, a converging momentum transfer section 18, an ejector throat and mixing section 19, a diverging diffuser pressure recovery section 20, and an ejector outlet section 21. Depending on design considerations, the size, shape, and inclusion of nozzle, ejector, and associated sections may vary.

The valve element 12 opens fully with substantially no restriction to fluid flow and closes fully with no intermediate positions. The nozzle sections 14, 15, and 16 accelerate fluid flow to the maximum attainable velocity with substantially no restriction to fluid flow. The nozzling device 10 achieves substantially isentropic flow when open. High velocity fluid flow from the nozzle outlet 16 transfers momentum to fluid within the ejector suction port 17 at the ejector converging momentum transfer section 18. High velocity fluid and entrained fluid from the ejector suction port 17 flow through the converging section 18 to the ejector throat and mixing section 19 and out of the ejector diverging diffuser section 20. In the ejector throat and mixing section 19 the primary fluid and the entrained fluid mix. In the ejector diverging diffuser section 20 some of the velocity of the fluid flow is recovered as a pressure rise. Flow through the ejector is partially isentropic for minimal fluid flow losses, the combined flow leaving through ejector outlet section 21.

The nozzling device 10 is actuated by a solenoid coil 23 which fully opens the valve element 12 when energized and fully closes the valve element when deenergized. A setpoint self-regulating pressure switch 24 regulates the operation of the solenoid coil 23. An electrical conduit 25 transfers power between the electric contacts of the setpoint switch 24 and the solenoid coil 23. An electrical conduit 26 supplies power to the solenoid coil 23 through the contacts of the switch 24 and the conduit 25. Power from the conduit 26 fully opens the nozzling device 10 when the contacts of the switch 24 complete an electrical circuit between 26, 25, and 23. When that circuit is broken by the opening of the contacts of the switch 24, the solenoid coil 23 is deenergized and the nozzling device 10 returns to its normally closed condition.

A conduit 28 transfers pressure information from the valve-nozzle transition section 13 within the nozzling device 10 to the setpoint switch 24. The conduit 28 is placed close to the outlet of the valve element 12 so that there is an immediate sensing of flow leaving the valve element 12. A conduit 29 transfers temperature information from a thermostatic bulb 30 to the setpoint switch 24. Temperature information from within a conduit 32 is transferred to the conduit 29 by the thermostatic bulb 30. The conjunction of the temperature and pressure information continuously modulates the momentary pressure and temperature setpoint of the setpoint switch 24. A compressor 33 functions to lower the pressure in the suction side of the system, and heat transferred from the ambient to an evaporator heat exchanger 34 functions to raise the temperature within the suction side of the system. The setpoint switch 24 opens the nozzling device 10 when the pressure-temperature relation rises above the switch setpoint, permitting fluid to flow from within an upstream conduit 34 through the valve element 12 to the valve-nozzle transition section 13. As the burst of fluid enters the transition section 13 and the high velocity fluid flows through an outlet conduit 35 it produces a pressure rise within the suction side of the system, changing the pressure-temperature relation between the sensed pressure and the sensed temperature. When the pressure-temperature relation is below the switch setpoint the contacts of the setpoint switch 24 open and the solenoid coil 23 deenergizes closing the nozzling device 10 and stopping fluid flow through the nozzling device 10. With the nozzling device 10 closed the compressor 33 lowers the suction side pressure as heat transfer from the ambient raises the suction side temperature until the pressure-temperature relation is above the switch setpoint, resulting in the reopening of the nozzling device 10. As the nozzling device 10 alternates between fully open and fully closed conditions, fluid alternately flows and does not flow within the thermodynamic system.

The high velocity burst of fluid flows into the evaporator heat exchanger 36 through the conduit 35 and out through a conduit 37. The evaporator heat exchanger 36 includes a lower header 38 and an upper header 39 interconnecting in parallel an array of closely spaced vertical evaporator tubes 40.

When the nozzling device 10 is open, suction within the pulsed ejector suction port 17 pulls refrigerant from an ejector suction conduit 41 to the ejector suction port 17. When the nozzling device 10 is closed the compressor 33 pulls refrigerant in the reversed direction out of the ejector suction port 17 through the conduit 41, back through the conduit 32, a counter-flow heat exchanger 42, a suction conduit 43, to the inlet of the compressor 33. Accompanying the reversal of flow direction within the ejector suction conduit 41 as the pulsed ejector opens and closes, pulling suction at the ejector suction port 17 and ceasing to pull suction, is a pseudo reversal of flow within the evaporator heat exchanger 36. The high velocity pulses of refrigerant proceed through the tubes 40 of the evaporator heat exchanger 36 with virtually no pressure drop. This results in a marked improvement in evaporator efficiency as compared to prior art evaporator heat exchangers which include long lengths of tubing with many U-bends. The liquid phase of the refrigerant passing through the evaporator heat exchanger 36 thoroughly wets the inside of the surface of the tubes 40 throughout their length which increases heat transfer. When the high velocity liquid contacts the inside of the tubes 40, and the refrigerant within the tubes 40, the bulk flow can change from laminar to turbulent which provides an additional increase in heat transfer. The vapor phase of the refrigerant passes centrally through the tubes 40.

The compressor 33 continuously acts to remove refrigerant from the outlet of the evaporator 36 through the conduit 37 and the inlet of the evaporator 36 as reversed flow through the ejector suction conduit 41. The flow of refrigerant within the ejector suction conduit 41 and the evaporator 36 experiences partial reversals with respect to the continuous direction of flow caused by the compressor as the pulsed ejector opens and closes. When the evaporator 36 is fabricated as a completely parallel heat exchanger with upper and lower headers, the high velocity flow through the conduit 35 entering the lower header of the evaporator 36 can cause an ejector-type suction by momentum transfer to the fluid within the vertical tubes 40. Thus the tubes of the evaporator 36 become multiple ejector stages, resulting in recirculating flow within the heat exchanger itself during a pulsed high velocity flow event, and a reversal of flow direction within the multiple ejector tubes when a pulse event ceases.

Refrigerant from the conduit 41 and the conduit 37 flows through the conduit 32 to the counter-flow heat exchanger 42. Fluid flows out of the counter-flow heat exchanger 42 through the conduit 43 to the compressor 33. The counter-flow heat exchanger 42 serves to further lower the temperature of the refrigerant leaving a condenser heat exchanger 44 and entering the nozzling device 10 by exchanging heat with the lower temperature refrigerant leaving the evaporator heat exchanger 36. The counter-flow heat exchanger 42 need not be used in all applications.

The condenser heat exchanger 44 includes a lower header 45 and an upper header 46 interconnected by a plurality of vertical closely spaced tubes 47 much like the configuration of the evaporator heat exchanger 36.

The compressor 33 transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through a conduit 48 to a nozzle inlet 49 of a condenser refrigerant overfeed simple pulsed ejector 50. When the nozzling device 10 opens to allow high impulse mass flow through the pulsed ejector body 11 and through the overall system, fluid flowing from a condenser overfeed ejector nozzle body 51 through a converging nozzle section 52 increases in velocity. The high velocity flow from the converging nozzle section 52 transfers momentum to fluid within a condenser overfeed ejector suction port 53. High velocity fluid flow from the nozzle outlet 52 transfers momentum to fluid within the ejector suction port 53 at an ejector converging momentum transfer section 54. High velocity fluid and entrained fluid from the ejector suction port 53 flow through the converging section 54 to an ejector throat and mixing section 55 and out of an ejector diverging diffuser section 56. In the ejector throat and mixing section 55 the primary fluid and the entrained fluid mix. In the ejector diverging diffuser section 56 some of the velocity of the fluid flow is recovered as a pressure rise. Fluid flows from the diverging section 56 through an ejector outlet 57 and a conduit 58 to the upper header 46 of the condenser heat exchanger 44. Depending on operating conditions, the size and shape of the ejector and nozzle sections and the relative position of the nozzle to the ejector sections can vary. Flow through the ejector is partially isentropic in the spirit of design for minimal fluid flow losses.

Fluid flows out of the lower header 45 of the condenser heat exchanger 44 through a conduit 59 to a conduit 60 to a filter-drier 61 which functions to filter out contaminants and remove moisture from the refrigerant. A simple ejector suction conduit 62 supplies refrigerant overfeed to the condenser overfeed ejector suction port 53. Filtered refrigerant flows out of the filter-drier 61 through a conduit 63 to the counter-flow heat exchanger 42. The fluid that enters the counter-flow heat exchanger 42 through the conduit 63 in counter-flow heat relationship with fluid flowing from the heat exchanger 36 to the compressor 33 emerges through the conduit 64, flows through a sight glass 65 and the conduit 34 and returns to the nozzling device 10 to complete a thermodynamic cycle. The sight glass 65 indicates the quality of the refrigerant in the system, and is not required in all applications.

As the thermodynamic system functions as a mechanical feedback loop, the self-regulating setpoint switch 24 will self-regulate the pressure and temperature setpoints at the valve-nozzle transition section 13 and the thermostatic bulb 30 to maintain the differential pressure between the conduit 29 and the conduit 28. As the self-regulation is relational, the magnitude of the sensed pressures can be from the vacuum range to the high pressure range, representing the entire range of pressures and temperatures that the refrigerant and the thermodynamic system are able to maintain.

The condenser 44 in FIG. 1 may be fed with refrigerant in its lower header 45 as will be described in reference to FIG. 2.

In the reversible heat pump system shown in FIG. 2, the fundamental components of the FIG. 1 embodiment are arranged in a split heat pump system. Component parts 10 to 21, 23 to 26, 28 to 30, 32, 36, and 41 of FIG. 1, composing a pulsed ejector, a setpoint self-regulating superheat switch, a heat exchanger, and interconnecting conduit are repeated twice as A and B sub-systems in FIG. 2 as component parts having the same reference numerals with A and B suffixes. Each of these functions in an analogous fashion to the corresponding component parts of FIG. 1.

Similarly, component parts 49 to 57 of FIG. 1, composing a simple ejector, are repeated twice in the A and B systems of FIG. 2 as component parts having the same reference numerals with A and B suffixes. Each of these parts in the A and B sub-systems in FIG. 2 functions in an analogous fashion to the corresponding component parts of FIG. 1.

Similarly, compressor 33, filter-drier 61, and sight glass 65 of FIG. 1 are repeated in FIG. 2 as compressor 33A, filter-drier 61A and sight glasses 65A and 65B respectively and they function in an analogous fashion to the corresponding component parts of FIG.1. The critical elements that make FIG. 1 into a reversible heat pump in FIG. 2 are: a reversing valve 70, and check valves 71, 71A, 72 and 72A which enable the direction of fluid flow through the system to reverse, switching the heat exchangers from being condenser and evaporator respectively to being evaporator and condenser respectively. In one operating mode, for example, the 'cooling' mode, the nozzling device 10B remains closed while the nozzling device 10A pulses. In the other operating mode, for example, the 'heating' mode, the nozzling device 10A remains closed while the nozzling device 10B pulses. The respective heating and cooling mode refrigerant flows are switched by the reversing valve 70.

One distinct difference in the circuit in FIG. 2 with respect to the circuit in FIG. 1 is the placement of the condenser overfeed ejector 50 of FIG. 1 relative to the upper and lower header of the condenser 44 of FIG. 1. In FIG. 1, the condenser overfeed ejector 50 is placed with its discharge section 57 at the upper header 46, and with its suction port 53 in communication with the lower header 45 through the ejector suction conduit 62. In FIG. 2, the condenser overfeed ejector 50A and the condenser overfeed ejector 50B are placed with their respective ejector discharge sections 57A and 57B in communication with the lower header 38A and 38B of the heat exchanger 36A and 36B respectively; and with the suction ports 53A and 53B in communication with the upper header 39A and 39B of the heat exchanger 36A and 36B respectively through ejector suction conduits 62A and 62B respectively. In FIG. 1, the condenser 44 is a 'top feed,' or 'upper header feed' condenser, with refrigerant entering the upper header 46 first.

In FIG. 2, the purpose of the check valve 71 and the check valve 71A is to route refrigerant so that the heat exchanger 36A and the heat exchanger 36B become 'bottom feed,' or 'lower header feed' condensers, with refrigerant entering the lower header 38A or 38B first.

In any particular version of the circuits represented in FIG. 1 and FIG. 2, the position and orientation of the condenser overfeed ejector can be in either the 'top feed' or 'bottom feed' placements. For overall simplicity, the condenser overfeed ejector can be removed completely from the system schematics, which would enable the removal of the check valves 71 and 71A, and unnecessary associated conduits in FIG. 2 as well. The spirit of FIG. 1 and FIG. 2 is to show representative piping schematics including a condenser overfeed ejector that can be further simplified as required by circumstance In either figure, the condensers can be piped as 'top feed or 'bottom feed' as required, with or without condenser overfeed ejectors.

In the 'cooling mode,' the nozzling device 10B remains closed. As the nozzling device 10A alternates between fully open and fully closed conditions, fluid alternately flows and does not flow within the thermodynamic system. With each pulse, a high velocity burst of fluid flows from the conduit 35A through a condui 73 into the evaporator heat exchanger 36A. When nozzling device 10A is open, suction within the pulsed ejector suction port 17A pulls refrigerant from the upper header 39A of the evaporator 36A through the ejector suction conduit 41A to the ejector suction port 17A. When the nozzling device 10A is closed the compressor 33A pulls refrigerant in the reversed direction out of ejector suction port 17A through the conduit 41A back to the inlet of the compressor 33A. Similar flow reversal and secondar ejector effects within the heat exchanger 36A occur as described with respect to heat exchanger 36 in FIG. 1.

Refrigerant flows out of the evaporator 36A from the conduit 58A and the conduit 59A. Refrigerant flowing through the conduit 58A flows backwards through the ejector 50A, flowing to the conduit 62A and a conduit 74. Refrigerant flow from the conduit 59A and the conduit 62A combines to flow through the conduit 37A. Refrigerant from the conduit 37A flows through the check valve 71 to a conduit 76. Refrigerant from the conduit 74 and the conduit 76 flows through the conduit 32A to the reversing valve 70. Refrigerant flows from the reversing valve 70 through the conduit 43A to the compressor 33A. The compressor 33A transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through the conduit 48A to the reversing valve 70. High pressure, high temperature refrigerant leaving the reversing valve 70 through the conduit 32B to a conduit 76A is prevented from entering the upper header 39B of the condenser 36B by the check valve 71A, resulting in the flow of refrigerant through a conduit 74A to the condenser overfeed ejector 50B.

When the nozzling device 10A opens to allow high impulse mass flow through the pulsed ejector body 11A and through the overall system, the condenser overfeed ejector 50B functions in a manner similar to the condenser overfeed ejector 50 of Fig. 1. Entrained fluid from the ejector suction port 53B flows from the ejector suction conduit 62B which flows from the conduit 59B which flows from the upper header 39B of the condenser 36B.

When the condenser 36B is fabricated as a completely parallel heat exchanger with upper and lower headers, and substantially vertical tubes, fluid flowing from the conduit 58B enters the lower header 38B of the condenser 36B. The condensation process becomes one of hot gas rising within the vertical tubes 40B, with condensed liquid forming and falling down the tubes, establishing an internal counterflow of rising gas and falling liquid. Condensed liquid is able to drain from the tube surface area into the lower header 38B, which acts as a liquid receiver, leaving the inner tube surface area available for condensing heat transfer. The entry of hot gas into a partially liquid filled lower header enables a very rapid de-superheating of the hot gas due to intimate contact with the liquid refrigerant, leaving the internal surface area of the condenser available for condensing heat transfer and subcooling heat transfer, which occurs at higher heat transfer coefficients than de-superheating heat transfer. Thus the internal surface area becomes more effective than in heat exchangers that devote a portion of their active internal tube surface area to the 'dry wall' de-superheating heat transfer process. Condensed liquid and vapor overfeed flow out of the condenser 36B upper header 39B through the conduit 59B and through the ejector suction conduit 62B serves to further enhance heat transfer within the condenser 36B. The conduit 62B supplies refrigerant overfeed to the condenser overfeed ejector suction port 53B.

High velocity flow leaving the condenser overfeed ejector 50B through the conduit 58B entering the lower header 38B of the condenser 36B can cause an ejector-type suction by momentum transfer to the fluid within the vertical tubes 40B. Thus the tubes of the condenser 36B become multiple ejector stages, resulting in recirculating flow within the heat exchanger itself during a pulsed high velocity flow event, and a reversal of flow direction within the multiple ejector tubes when a pulse event ceases. The flow reversals and counter-flow characteristics within the condenser tubes can be considered natural convection processes when a pulse ceases, and forced convection processes when a pulse flow occurs.

Fluid flows out of the condenser heat exchanger 36B through a conduit 73A, and is prevented from flowing through the conduit 35B and the ejector body IIB by pulsed ejector valve element 12B, which remains closed. Condensed refrigerant from conduit 73A flows through a conduit 77A, through the check valve 72A, through a conduit 78A, through a conduit 34B, through the sight glass 65B, through the conduit 64B, to the filter-drier 61A which functions to filter out contaminants and remove moisture from the refrigerant.

Filtered refrigerant flows through the conduit 64A, through the sight glass 65A to the conduit 34A. Refrigerant is prevented from flowing through a conduit 78 by the check valve 72. Refrigerant from the conduit 34A flows through a conduit 79, returning to the nozzling device 10A to complete a thermodynamic cycle. The sight glass 65A and the sight glass 65B are not required in all applications.

As the thermodynamic system functions as a mechanical feedback loop, the self-regulating setpoint switch 24A will self-regulate the pressure and temperature setpoints at the valve-nozzle transition section 13A and the thermostatic bulb 30A to maintain the differential pressure between the conduit 29A and the conduit 28A which can be related to a thermodynamic superheat.

To switch between 'cooling mode' and 'heating mode', reversing valve 70 is actuated. Due to the substantial lack of flow restriction within the valve elements of both pulsed ejectors, upon reversing modes the heat exchanger pressures equalize extremely rapidly. This enables a rapid changeover between operating modes. In reversible heat pumps of the prior art utilizing substantial flow restricting metering devices, a significant delay is often required between switching heat pump modes to allow for heat exchanger pressure equalization.

In the 'heating mode,' the nozzling device 10A remains closed. As nozzling device 10B alternates between fully open and fully closed conditions, fluid alternately flows and does not flow within the thermodynamic system. With each pulse, a high velocity burst of fluid flows from the conduit 35B through the conduit 73A into the lower header 38B of the evaporator heat exchanger 36B. When the nozzling device 10B is open, suction within the pulsed ejector suction port 17B pulls refrigerant from the upper header 39B of the evaporator 36B through the ejector suction conduit 41B to the ejector suction port 17B. When the nozzling device 10B is closed the compressor 33A pulls refrigerant in the reversed direction out of the ejector suction port 17B through the conduit 41B back to the inlet of the compressor 33A. Similar flow reversal and secondary ejector effects within the heat exchanger 36B occur as described with respect to the heat exchanger 36 in FIG. 1.

Refrigerant flows out of the'evaporator 36B from the conduit 58B and the conduit 59B. Refrigerant flowing through the conduit 58B flows backwards through the ejector 50B, flowing to the conduit 62B and the conduit 74A. Refrigerant flow from conduit 59B and the conduit 62B combines to flow through the conduit 37B. Refrigerant from conduit 37B flows through the check valve 71A to the conduit 76A. Refrigerant from the conduit 74A and the conduit 76A flows through the conduit 32B to the reversing valve 70. Refrigerant flows from the reversing valve 70 through the conduit 43A to the compressor 33A.

The compressor 33A transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through conduit 46A to the reversing valve 70. High pressure, high temperature refrigerant leaving the reversing valve 70 through the conduit 32A to the conduit 76 is prevented from entering the upper header 39A of the condenser 36A by the check valve 71, resulting in the flow of refrigerant through the conduit 74 to the nozzle inlet 49A of the condenser overfeed ejector 50A.

When the nozzling device 10B opens to allow high impulse mass flow through the pulsed ejector body 11B and through the overall system, fluid flows through the ejector 50A and its associated conduits in a manner similar to that described for the ejector 50B and its associated conduits in the 'cooling mode' mentioned previously.

When the condenser 36A is fabricated as a completely parallel heat exchanger with upper and lower headers, and substantially vertical tubes, fluid flowing from the conduit 58A enters the lower header of the condenser 36A. The condensation process becomes that described by the condenser 36B in the 'cooling mode' heretofore described.

Condensed liquid and vapor overfeed flow out of the condenser 36A upper header 39A through the conduit 59A and through the ejector suction conduit 62A serves to further enhance heat transfer within the condenser 36A. The conduit 62A supplies refrigerant overfeed to the condenser overfeed ejector suction port 53A.

High velocity flow leaving the condenser overfeed ejector 50A through the conduit 58A entering the lower header 38A of the condenser 36A can cause an ejector-type suction by momentum transfer to the fluid within the vertical tubes 40A as previously described.

Fluid flows out of the condenser heat exchanger 36A through the conduit 73, and is prevented from flowing through the conduit 35A and the ejector body 11A by the pulsed ejector valve element 12A, which remains closed. Condensed refrigerant from the conduit 73 flows through a conduit 77, through the check valve 72, through the conduit 78, through the conduit 34A, through the sight glass 65A, through the conduit 64A, to the filter-drier 61A. Filtered refrigerant flows through the conduit 64B, through the sight glass 65B to the conduit 34B. Refrigerant is prevented from flowing through the conduit 78A by the check valve 72A. Refrigerant from the conduit 34B flows through a conduit 79A, returning to the nozzling device 10B to complete a thermodynamic cycle.

As the thermodynamic system functions as a mechanical feedback loop, the self-regulating setpoint switch 24B will self-regulate the pressure and temperature setpoints at the valve-nozzle transition section 13B and the thermostatic bulb 30B to maintain the differential pressure between the conduit 29B and the conduit 28B which can be related to a thermodynamic superheat.

In the reversible heat pump system shown in FIG. 3, the fundamental components of FIG. 2 are arranged in a unitary heat pump system with a bi-directional pulsed ejector 10C made possible by bi-directional flow through a valve element 12C, which is actuated by a solenoid coil 23C.

Component parts in FIG. 3 with the same reference numerals as in FIG. 1 and FIG. 2 function in an analogous fashion as in those other embodiments. Suffixes C and D are used here in FIG. 3 for parts corresponding to the numerals with A and B suffixes in FIG. 2.

The critical element that makes the FIG. 3 embodiment into a reversible unitary heat pump is a bi-directional flow through valve element 12C. This enables a reversing valve 70A to change the diredtion of fluid flow through the system without the need for check valves. In one operating mode, for example, the 'cooling' mode, a bi-directional pulsed ejector 10C meters refrigerant flow in one direction, from the condenser heat exchanger 36D into the evaporator heat exchanger 36C, actuated by the self-regulating setpoint switch 24C. In the other operating mode, for example, the 'heating' mode, the bi-directional pulsed ejector 10C meters refrigerant flow in the opposite direction, from the condenser heat exchanger 36C into the evaporator heat exchanger 36D, actuated by the self-regulating setpoint switch 24D. The respective heating and cooling mode refrigerant flows are switched by the reversing valve 70A.

In FIG. 3, the purpose of the check valve 71B and the check valve 71C is to route refrigerant so that the heat exchanger 36C and the heat exchanger 36D become 'bottom feed', or 'lower header feed' condensers, with refrigerant entering the lower header first.

In FIG. 3, a condenser overfeed ejector could be installed in either the 'top feed' or 'bottom feed' placements as described in FIG. 1 and FIG. 2. For overall simplicity, the condensers could be piped as 'top feed', which would enable the removal of the check valves 71B and 71C, and the connecting conduits 74C, 37C, 76B, and 74D, 37D, 76C as well. The spirit of FIG. 1, FIG. 2, and FIG. 3 is to show representative piping schematics, including condenser overfeed ejectors, that can be further simplified or augmented as required by circumstance. In each figure, the condensers can be piped as 'top feed' or 'bottom feed' as required.

In the 'cooling mode,' as the bi-directional pulsed ejector 10C alternates between fully open and fully closed conditions, fluid alternately flows and does not flow within the thermodynamic system. With each pulse, a high velocity burst of fluid flows from the conduit 35C into the lower header 38C of the evaporator heat exchanger 36C. When the valve element 12C is open, suction within the pulsed ejector suction port 17C pulls refrigerant from the upper header 39C of the evaporator 36C through the ejector suction conduit 41C to the ejector suction port 17C. When the valve element 12C is closed the compressor 33B pulls refrigerant in the reversed direction out of the ejector suction port 17C through the conduit 41C back to the inlet of the compressor 33B. Similar flow reversal and secondary ejector effects within the heat exchanger 36C occur as described with respect to the heat exchanger 36A in FIG. 2.

Refrigerant flows out of the evaporator 36C from the conduit 74C and the conduit 37C. Refrigerant from the conduit 37C flows through the check valve 71B to the conduit 76B. Refrigerant flow from the conduit 76B and the conduit 74C combines to flow through the conduit 32C. Refrigerant from the conduit 32C flows to the reversing valve 70A. Refrigerant flows from the reversing valve 70A through the conduit 43B to the compressor 33B.

The compressor 33B transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through the conduit 48B to the reversing valve 70A. High pressure, high temperature refrigerant leaving the reversing valve 70A through the conduit 32D to the conduit 76C is prevented from entering the upper header 39D of the condenser 36D by the check valve 71C, resulting in the flow of refrigerant through the conduit 74D to the lower header of the condenser 36D. When the condenser 36D is fabricated as a completely parallel heat exchanger with upper and lower headers, the condensing process is similar to that described for the condenser 36B of FIG. 2.

Fluid flows out of the condenser heat exchanger 36D through the conduit 35D. Refrigerant from the conduit 35D flows backwards through the ejector body 11D which consists of sections 21D, 20D, 19D, 18D and 17D, and backwards through the ejector nozzle which consists of 16D, 15D and 14D, until the flow reaches the valve-nozzle transition section 13D. When the valve element 12C opens, fluid from the valve-nozzle transition section 13D flows through valve element 12C to the valve-nozzle transition section 13C. The pulse of fluid flows through the nozzle and the ejector body into the evaporator 36C as previously described.

To switch between 'cooling mode' and 'heating mode,' the reversing valve 70A is actuated. When the heat exchanger 36C functions as an evaporator, the self-regulating setpoint switch 24C actuates the bi-directional pulsed ejector 10C. When the heat exchanger 36D functions as an evaporator, self-regulating setpoint switch 24D actuates the bi-directional pulsed ejector 10C.

To accomplish the rapid hot gas defrost, a rapid defrost switch 80 transfers electrical power from the conduit 26E to the conduit 25E which transfers power to actuate reversing valve 70A to reverse the direction of refrigerant flow. This reversal of flow sends hot gas to what was previously the evaporator, to accomplish the defrosting of the heat exchanger. When the defrosting is substantially completed the rapid defrost switch 80 actuates the reversing valve 70A to return to the prior flow direction, allowing the defrosted heat exchanger to resume function as an evaporator.

flows and does not flow within the thermodynamic system. With each pulse, a high velocity burst of fluid flows from the conduit 35D into the evaporator heat exchanger 36D. when the valve element 12C is open, suction within the pulsed ejector suction port 17D pulls refrigerant from the upper header 39D of the evaporator 36D through the ejector suction conduit 41D to the ejector suction port 17D. When the valve element 12C is closed the compressor 33B pulls refrigerant in the reversed direction out of the ejector suction port 17D through the conduit 41D back to the inlet of the compressor 33B. Similar flow reversal and secondary ejector effects within the heat exchanger 36D occur as described with respect to the heat exchanger 36A in FIG. 2.

Refrigerant flows out of the evaporator 36D from the conduit 74D and the conduit 37D. Refrigerant from the conduit 37D flows through the check valve 71C to the conduit 76C. Refrigerant flow from the conduit 76C and the conduit 74D combines to flow through the conduit 32D. Refrigerant from the conduit 32D flows to the reversing valve 70A. Refrigerant flows from the reversing valve 70A through the conduit 43B to the compressor

33B.

The compressor 33B transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through the conduit 48B to the reversing valve 70A. High pressure, high temperature refrigerant leaving the reversing valve 70A through the conduit 32C to the conduit 76B is prevented from entering the upper header 39C of the condenser 36C by the check valve 71B, resulting in the flow of refrigerant through the conduit 74C to the lower header 38C of the condenser 36C. When the condenser 36C is fabricated as a completely parallel heat exchanger with upper and lower headers, the condensing process is similar to that described for the condenser 36B of FIG. 2.

Fluid flows out of the condenser heat exchanger 36C through the conduit 35C. Refrigerant from the conduit 35C flows backwards through the ejector body 11C which consists of the sections 21C, 20C, 19C, 18C and 17C and backwards through the ejector nozzle which consists of the sections 16C, 15C and 14C until the flow reaches the valve-nozzle transition section 13C. When the valve element 12C opens, fluid from the valve-nozzle transition section 13C flows through the valve element 12C to the valve-nozzle transition section 13D. The pulse of fluid flows through the nozzle and the ejector body into the evaporator 36D as previously described.

As the thermodynamic system functions as a mechanical feedback loop, when the heat exchanger 36C functions as an evaporator, the self-regulating setpoint switch 24C will self-regulate the pressure and temperature setpoints at the valve-nozzle transition section 13C and the thermostatic bulb 30C to maintain the differential pressure between the conduit 29C and the conduit 28C which can be related to a thermodynamic superheat.

As the thermodynamic system functions as a mechanical feedback loop, when the heat exchanger 36D functions as an evaporator, the self-regulating setpoint switch 24D will self-regulate the pressure and temperature setpoints at the valve-nozzle transition section 13D and the thermostatic bulb 30D to maintain the differential pressure between the conduit 29D and the conduit 28D which can be related to a thermodynamic superheat.

In the reversible heat pump system shown in FIG. 4, the fundamental components of FIG. 2 are arranged in a unitary heat pump system with a single pulsed ejector 10D and two reversing valves, 70B and 70C. The reversing valve 70B switches the compressor 33C hot gas discharge flow to the heat exchanger 36E or to the heat exchanger 36F depending on the mode of operation. The reversing valve 70C switches pulsed ejector discharge flow to the heat exchanger 36F or to the heat exchanger 36E depending on the mode of operation. Thus a single pulsed ejector can be utilized in a reversible heat pump circuit. Any liquid refrigerant returning to a suction accumulator 82 is recirculated through the heat exchanger acting as an evaporator due to suction flow from the pulsed ejector 10D. The valve element 12D of the pulsed ejector 10D is actuated by the solenoid coil 23D.

Component parts 24E, 25F, 26F, 28E, 29E, 30E and 11E of FIG. 4, function in an analogous fashion to the corresponding component parts 24B, 25B, 26B, 28B, 29B, 30B and 11B, respectively, of FIG. 2.

Other component parts in FIG. 4 with the same reference numerals as in FIG. 1 function in an analogous fashion as in FIG. 1.

In one operating mode, for example, the 'cooling' mode, the pulsed ejector 10D meters refrigerant flow through the conduit 37E to the reversing valve 70C. Refrigerant from the reversing valve 70C flows through the conduit 73E into the evaporator heat exchanger 36E. Refrigerant leaving the evaporator 36E through the conduit 32F enters the reversing valve 70B. Refrigerant entering the reversing valve 70B from the conduit 32F leaves the reversing valve 70B through the conduit 83. The thermostatic bulb 30E senses the temperature of the refrigerant within the conduit 83. Refrigerant from the conduit 83 flows into the suction accumulator 82.
phase leaves suction accumulator 82 through the conduit 43C to enter the compressor 33C. Compressed refrigerant leaves compressor 33C through the conduit 48C to enter the reversing valve 70B. Refrigerant entering the reversing valve 70B from the conduit 48C leaves the reversing valve 70B through the conduit 32E. Refrigerant from the conduit 32E enters the condenser heat exchanger 36F. Refrigerant leaves the condenser 36F through the conduit 73F to enter the reversing valve 70C. Refrigerant entering the reversing valve 70C from the conduit 73F leaves the reversing valve 70C through the conduit 34C to enter pulsed ejector 10D.

Liquid or vapor refrigerant from the suction accumulator 82 leaves through the conduit 41E due to the suction action of the pulsed ejector suction port 17E. An optional check valve may be placed within the conduit 41E to prevent refrigerant liquid from reversing direction and flowing from the ejector suction port 17E through the conduit 41E back into the accumulator 82.

The conjunction of bulb temperature 30E and system pressure at 13E determines the actuation of the pulsed ejector 10D for metering refrigerant into the evaporator 36E from the condenser 36F.

In the other operating mode, for example, the 'heating' mode, the pulsed ejector 10D meters refrigerant flow through the conduit 35E to the reversing valve 70C. Refrigerant from the reversing valve 70C flows through the conduit 73F into the evaporator heat exchanger 36F. Refrigerant leaving the evaporator 36F through the conduit 32E enters the reversing valve 70B. Refrigerant entering the reversing valve 70B from the conduit 32E leaves the reversing valve 70B through the conduit 83. The thermostatic bulb 30E senses the temperature of the refrigerant within the conduit 83. Refrigerant from the conduit 83 flows into the suction accumulator 82. Refrigerant in the vapor phase leaves suction accumulator 82 through the conduit 43C to enter the compressor 33C. Compressed refrigerant leaves the compressor 33C through the conduit 48C to enter the reversing valve 70B. Refrigerant entering the reversing valve 70B from the conduit 48C leaves the reversing valve 70B through the conduit 32F. Refrigerant from the conduit 32F enters the condenser heat exchanger 36E. Refrigerant leaves the condenser 36E through the conduit 73E to enter the reversing valve 70C. Refrigerant entering the reversing valve 70C from the conduit 73E leaves the reversing valve 70C through the conduit 34C to enter the pulsed ejector 10D.

The conjunction of bulb temperature 30E and system pressure at 13E determines the actuation of the pulsed ejector 10D for metering refrigerant into the evaporator 36F from the condenser 36E.

Recirculation of liquid refrigerant from suction accumulator 82 through an evaporator heat exchanger enables the liquid refrigerant to evaporate and provide cooling capacity for the system.

When necessary, rapid defrosting may be accomplished by switching from heating mode to cooling mode for the duration of the defrost cycle, and then switching back to heating mode. Condenser overfeed ejectors and associated piping can be added as required.

The part load refrigerant storage system shown in FIG. 5 is a result of the ability of pulsed metering devices to effectively meter refrigerant of any quality; subcooled liquid, saturated liquid, two phase, and vapor. The purpose of the part load refrigerant management system is to vary the cooling capacity of a system by varying the quality of the refrigerant leaving the condenser by varying the active refrigerant charge within the system. The cooling capacity in the evaporator is relative to the condenser leaving liquid quality, with the most cooling capacity for subcooled liquid, less for saturated liquid, less for two phase, and less for vapor.

In a typical system, one charges the system until there is subcooled liquid present leaving the condenser. This is accomplished by adding refrigerant charge until a liquid line sight glass is full, and the liquid temperature is below the saturation temperature of the liquid pressure, indicating thermodynamic subcooling. A superheat measurement is made at the evaporator outlet for the desired saturation evaporator temperature, in order to determine whether there is sufficient cooling capacity. In order to lower superheat and increase cooling capacity, more refrigerant charge is added to the system to increase the condenser subcooling. In order to raise the superheat and decrease cooling capacity, refrigerant is removed from the system to decrease the condenser subcooling.

In the expansion systems of the prior art, a "liquid seal" at the expansion device inlet is typically required, which requires some degree of liquid subcooling as a result. Thus the removal of refrigerant charge to decrease cooling capacity at low load is not practical, lest saturated or two phase refrigerant enter the expansion device causing faulty system operation. As a result, refrigeration systems of the prior art have difficulty operating at low load conditions effectively, often utilizing inefficient means of false loading the compressor such as hot gas bypass, where hot gas is bypassed from the compressor outlet directly back to the compressor inlet, forcing the compressor to do pumping work without doing any cooling with the bypassed flow. The other method of lowering load is to reduce the evaporator pressure, and thus temperature. This can lead to dropping the evaporator temperature below 32 F, resulting in frosting over, and blockage of the evaporator, which can lead to compressor failure due to excessive liquid floodback. In compressors that have performance curves that are very sensitive to suction pressure and density, such as centrifugal compressors, going to lower load conditions by lowering evaporator pressure can result in very poor efficiency and performance at low loads.

The present invention functions by bypassing condenser outlet refrigerant into a reservoir when evaporator outlet superheat drops, which lowers the condenser pressure and liquid subcooling by removing refrigerant charge from the active system loop. As necessary due to low load and low superheat, refrigerant can be bypassed into the reservoir until the condensing leaving refrigerant is two phase, and even just vapor, resulting in lower cooling capacity in the evaporator. As superheat rises, refrigerant from the reservoir is returned to the active refrigerant system loop by entering the evaporator, increasing the active refrigerant charge, which increases the condenser pressure and lowers the quality of the refrigerant leaving the condenser. Just as in charging the system initially, the condenser leaving refrigerant will go from vapor to two phase to saturated to subcooled, as refrigerant is added to the active system loop. The lower the quality of the condenser leaving refrigerant, the higher the cooling capacity in the evaporator. Thus the refrigerant bypass storage system manages low and high load conditions by varying the quality of the refrigerant leaving the condenser. As the pulsed ejector can effectively meter any quality refrigerant, performance of the overall system remains within required operating realms.

In the embodiment of the refrigerant bypass storage system shown in FIG. 5, the pulsed ejector 10E is actuated by the pressure switch 24F. Electric power from the conduit 26G enters the pressure switch 24F, and is transferred through the conduit 25G to the solenoid coil 23E when the switch contacts within the pressure switch 24F are closed, completing an electric circuit between the conduit 26G, the pressure switch 24F, the conduit 25G, and the solenoid coil 23E. When the switch contacts within the pressure switch 24F are opened, breaking the electric circuit between the conduit 26G, the pressure switch 24F, the conduit 25G, and the solenoid coil 23E electric power ceases to flow. Pressure switch 24F actuates the solenoid coil 23E, which opens and closes the valve element 12E, metering refrigerant flow to maintain a pressure setpoint. The pressure switch 24F receives pressure information from the system through the conduit 28F, which senses pressure at the valve-nozzle transition section 13F. The pressure switch 24F opens the valve element 12E on a drop in sensed pressure below the pressure setpoint, and closes the valve element 12E on a rise in sensed pressure above the pressure setpoint. The opening and closing of the substantially unrestricted valve element 12E causes high velocity pulsed flow events through the valve body that are sensed by the pressure switch 24F, resulting in a mechanical feedback loop self-regulation of pulse rate and flow to maintain the pressure setpoint.

The pressure setpoint of the pressure switch 24F can be, for example, the pressure at which the compressor 33D achieves optimum performance. As load variations change, the pressure switch 24F will maintain the evaporator pressure at the optimum point, and the refrigerant bypass system will vary the condenser leaving refrigerant quality based on evaporator outlet superheat, effectively modulating evaporator cooling capacity. Regulating system performance by refrigerant bypass based on a superheat determination is useful in that the compressor requires a certain minimum superheat to avoid damage, and the effective use of the evaporator surface area depends on a minimum, regulated superheat. For example, given that a means of providing superheat to the evaporator outlet refrigerant is provided within the system, the evaporator could be run with a fully wetted surface area, increasing evaporator performance and cooling capacity, increasing system performance with the compressor still protected from damage by refrigerant liquid or wet vapor. Load management with the refrigerant bypass modulation of condenser leaving refrigerant quality maintains the system performance and efficiency within operating requirements as operating conditions vary.

In the refrigeration system, the practical requirement of the refrigerant bypass system and bypass reservoir is to have sufficient refrigerant to return to the active system at high loads, and sufficient volume to store refrigerant at low loads. Most importantly, the relative levels of refrigerant in the active system and in the reservoir should be self-regulated to maintain optimum system performance and efficiency as operating conditions vary. This self-regulated balance can occur due to the conjunction of a pressure switch for regulating evaporator pressure and a superheat switch for regulating refrigerant bypass into the reservoir. Other combinations of system variables, such as pressure, temperature, superheat, subcooling, and concentration, can be utilized to self-regulate system operation and refrigerant bypass storage and release.

In the part load refrigeration storage system shown in FIG. 5, the thermostatic bulb 30F transfers temperature information from the evaporator 36G outlet conduit 32G to the superheat switch 24G through the conduit 29F. System pressure information is transferred to the superheat switch 24G through the conduit 28G and the conduit 85, which senses pressure at the valve-nozzle transition element 13F. The superheat switch 24G can be composed of a differential pressure switch acting on the difference in pressure between the conduit 28G representing system pressure and the conduit 29F representing pressure within the thermostatic bulb 30F. The superheat switch 24G acts to allow refrigerant to enter a bypass reservoir 86 on a drop in the sensed differential pressure, which can be related to a drop in superheat, and acts to allow refrigerant to leave the bypass reservoir 86 on a rise in the sensed differential pressure, which can be related to a rise in superheat.

On a rise in sensed superheat, electrical power from the conduit 26H is transferred by the superheat switch 24G through the conduit 25I to the solenoid coil 23G, which opens the solenoid valve 10G, allowing refrigerant from the bypass reservoir 86 to leave through a conduit 87, flow through solenoid valve 10G, flow through a conduit 85 and enter the active system loop through the valve-nozzle transition element 13F. The solenoid valve 10F is closed.

On a drop in sensed superheat, electrical power from the conduit 26H is transferred by the superheat switch 24G through the conduit 25H to the solenoid coil 23F, which opens the solenoid valve 10F, allowing condensed refrigerant from the conduit 90 to flow through the solenoid valve 10F, through the conduit 91 and enter the bypass reservoir 86. The solenoid valve 10G is closed.

As superheat rises and falls, refrigerant alternately enters and leaves the reservoir 86, alternately entering and leaving the active system circuit.

As the pressure switch 24F maintains evaporator pressure, the superheat switch 24G maintains evaporator superheat. The conjunction of the pressure switch 24F and the superheat switch 24G act to handle load variations.

In FIG. 5, components 33D, 48D, 61B, 64C, 65C, 34D and 79B function in an analogous fashion to the corresponding component parts 33A, 48A, 61A, 64B, 65B, 34B and 79A respectively, of FIG. 2. The condenser heat exchanger 36H includes a lower header 38E, parallel vertical tubes 40C and a dead-ended upper header 39E. It is a 'bottom feed' condenser fed with hot gas from the conduit 48D, with condensed liquid leaving from the bottom header through the conduit 59A. Hot superheated gas from the compressor discharge enters the lower header 38E from one end, rises into the tubes 40C, condenses, setting up an internal turbulent counterflow of hot gas rising within the tubes 40C and condensed liquid falling down the tube walls to collect in the lower header 38E. As liquid fills the lower header 38E the hot gas very rapidly de-superheats by the intimate mixing process of bubbling through the condensed liquid. Condensed, and even subcooled liquid refrigerant leaves the lower header 38E through the conduit 59A at the end opposite to the hot gas inlet from the conduit 48D with each high velocity pulse event.

The pulsed ejector 10E, ejector body 11E, and their component parts function in analogous fashion to the pulsed ejector 10A, ejector body 11A, and their component parts of FIG. 2. The only substantial difference is that the pulsed ejector 10E is actuated by the pressure switch 24F, whereas the pulsed ejector 10A is actuated by the superheat switch 24A. The heat pump system in FIG. 2 will vary load by self-regulating superheat, and by raising and lowering evaporator pressure and temperature as required. The refrigeration system in FIG. 5 will vary load by self-regulating at a relatively fixed evaporator pressure and temperature.

In FIG. 6, the pulsed ejector 10H regulates the flow of a refrigerant to an evaporator heat exchanger 36I in which the refrigerant takes on heat from the environment to be cooled. The refrigerant vapor then flows into an absorber 93 where it is absorbed by thermodynamic fluid in the liquid state, releasing heat energy in an exothermic process to the ambient environment. A pump 94 pressurizes the liquid from the absorber 93, pumping it through a counter-flow heat exchanger 95 into a vapor generator 96. The high pressure liquid in the vapor generator 96 absorbs heat energy from a higher temperature ambient source, releasing the refrigerant vapor absorbed into the liquid in the absorber 93 as a high pressure and high temperature vapor. The high pressure and high temperature refrigerant vapor flows from the vapor generator 96 to a rectifier 97 which removes any water in the liquid or vapor phase from the refrigerant vapor. In some applications the rectifier 97 is not required; for example, when water is the refrigerant. The dry vapor leaving the rectifier 97 condenses into liquid refrigerant in the condenser heat exchanger 44A which releases heat energy to the ambient environment. Liquid refrigerant from the condenser 44H flows to the inlet of the pulsed ejector 10H to complete a thermodynamic cycle.

Liquid absorbent fluid from the vapor generator 96 flows back through a conduit 98 to the counter-flow heat exchanger 95, where it exchanges heat energy with fluid flowing from the pump 94 to the vapor generator 95, and then through a conduit 99 to the pulsed ejector 10I which opens and closes to meter absorbent fluid flow back to the absorber 93.
temperature information, alternately opening and closing the pulsed ejector 10H. The pulsed ejector 10H replaces a throttling expansion valve in systems of the prior art.

Absorbent fluid continually cycles through its system loop in an intermittent fashion as the differential pressure switch 24J responds to internal pressure information, alternately opening and closing the pulsed ejector 10I. The pulsed ejector 10I replaces a throttling metering valve in systems of the prior art.

Both the pulsed ejector 10H and the pulsed ejector 10I recover energy wasted in the throttling devices of the systems of the prior art. In systems where there are additional throttling devices, the throttling devices can be replaced with pulsed ejectors, or simple ejectors that do not include actuated valve elements.

The pulsed ejector absorption refrigeration system in FIG. 6 includes a condenser overfeed ejector 50C, the pulsed ejector 10H feeding the evaporator 36I and providing for recirculation, and the pulsed ejector 10I feeding the absorber 93 and providing flow assist in moving fluid from the evaporator 36I to the absorber 93.

The solenoid-actuated pulsed ejector 10H is actuated by the solenoid coil 23J which fully opens the valve element 12H when electrically energized and fully closes the valve element 12H when deenergized. The superheat switch 24I regulates the energization and de-energization of the solenoid coil 23J. The electrical conduit 25K transfers electrical power between the electric contacts of the superheat switch 24I and the solenoid coil 23J. The electrical conduit 26J supplies electrical power to the solenoid coil 23J through the electric contacts of the switch 24I and the electrical conduit 25K.

Electric power from the conduit 26J fully opens the pulsed ejector 10H when the contacts of the switch 24I complete an electrical circuit between 26J, 25K, and 23J. When the electrical circuit between them is broken by the opening of the electrical contacts of the switch 24I, the solenoid coil 23J is deenergized and the pulsed ejector 10H returns to its normally closed condition.

The conduit 28H transfers pressure information from within the valve-ejector transition section 13I to the superheat switch 24I. The conduit 29I transfers pressure information from within the thermostatic bulb element 30I to the superheat switch 24I. The thermostatic bulb 30I senses system temperature at the evaporator outlet conduit 32I. The differential pressure at which the superheat switch 24I is set to open the pulsed ejector 10H is chosen by design criterion. Thermodynamic criterion other than superheat can be utilized to the actuate pulsed ejector 10H.

As the pump 94 lowers the pressure in the suction side of the thermodynamic system, and heat energy is added to the evaporator 36I, the superheat switch 24I opens the pulsed ejector 10H when the differential pressure between the bulb 30I pressure and the system pressure at the valve-ejector transition section 13I, which can be related to a superheat equivalent, rises above the differential pressure setting of the superheat switch 24I, permitting flow of the thermodynamic fluid from within the upstream system conduit 34E through the pulsed ejector 10H to the downstream system conduit 35I.

As the high velocity burst of thermodynamic fluid enters the valve-ejector transition section 13I and flows to the downstream conduit 35I it produces an internal system pressure rise within the suction side of the system. Refrigerant flowing through the evaporator 36I to the outlet conduit 32I can lower the temperature as sensed by the thermostatic bulb 30I, lowering its internal bulb pressure.

When the differential pressure sensed by the superheat switch 24I drops below the differential pressure setpoint of the superheat switch 24I, the electric contacts of the superheat switch 24I open and the solenoid coil 23J deenergizes closing the valve element 12H within the pulsed ejector 10H, and stopping thermodynamic fluid flow through the pulsed ejector 10H.

With the pulsed ejector 10H closed, the pump 94 lowers the suction side pressure, and heat addition to the evaporator 36I can raise the temperature sensed at the evaporator outlet conduit 32I by the thermostatic bulb 30I, the differential pressure sensed by the superheat switch 24I rises above its differential pressure setpoint, resulting in the reopening of the pulsed ejector 10H.

As the pulsed ejector 10H alternates between fully open and fully closed conditions, refrigerant fluid alternately flows and does not flow within the thermodynamic system.

The pulsed ejector 10H component parts of FIG. 6 function in an analogous fashion to the corresponding pulsed ejector 10 component parts of FIG. 1.

In FIG. 6, the ejector suction conduit 41G is shown recirculating fluid from what can be construed to be the upper header 39F of a parallel, vertical, evaporator 36I.

The high velocity burst of thermodynamic fluid flows into the lower header 38I of the evaporator heat exchanger 36I through the conduit 35I, resulting in internal recirculation flows within evaporator 36I as previously described. Fluid flows out of evaporator 36I through the ejector suction conduit 32I to the ejector suction port 17J of the pulsed ejector 10I. When the pulsed ejector 10I is closed, refrigerant fluid from the conduit 32I may still flow through the body of the pulsed ejector 10I, flowing through the ejector suction port 17J, the converging section 18J, the ejector throat/bore section 19J, the diverging diffuser section 20J, the ejector outlet section 21J, flowing through the conduit 35J to enter the absorber 93.

Refrigerant from the evaporator 36I is absorbed by the liquid absorbent fluid within the absorber 93 in what is typically an exothermic process, releasing heat energy to the external environment. Liquid absorbent fluid is pumped out of the absorber 93 through a conduit 100 by pump 94.

The pump 94 raises the pressure of the liquid absorbent fluid and discharges the pressurized liquid through a conduit 101 to the counter-flow heat exchanger 95. Pressurized liquid absorbent fluid that enters the counter-flow heat exchanger 95 through the conduit 101 leaves through a conduit 102 and enters the vapor generator 96. Heat energy from a higher temperature ambient environment is transferred to the vapor generator 96 so that refrigerant vapor is released from the absorbent fluid in an endothermic process.

The high pressure refrigerant vapor leaves the vapor generator 96 through a conduit 103 and flows to the rectifier 97 which functions as a desiccant to remove any water in the liquid or vapor phase from the refrigerant vapor. Dry refrigerant vapor leaves the rectifier 97 through a conduit 104 and enters a simple condenser overfeed ejector 50C prior to entering the condenser heat exchanger 44A.

In FIG. 6, simple pulsed ejector 50C and its component parts function in an analogous fashion to simple pulsed ejector 50 and component parts in FIG. 1. Discharge flow from simple pulsed ejector 50C flows through conduit 58C to the upper header 39G of condenser heat exchanger 44A.

The ejector suction conduit 62C is shown recirculating fluid from what can be construed to be the lower header 45A of a parallel, vertical, condenser 44A.

Dry refrigerant vapor within the condenser heat exchanger 44A changes thermodynamic state to refrigerant liquid as it releases heat energy to heat the external environment. Liquid refrigerant leaves the condenser 44A through the conduit 34E and flows to the inlet of the pulsed ejector 10H to complete a thermodynamic cycle.

High pressure liquid absorbent fluid from the vapor generator 96 leaves through the conduit 98 and flows through the counter-flow heat exchanger 95, where it transfers heat energy to absorbent fluid flowing from the pump 94 to the vapor generator 96, preheating the absorbent fluid before it enters the vapor generator 96. Liquid absorbent fluid entering the counter-flow heat exchanger 95 through the conduit 98 leaves the counter-flow heat exchanger 95 through the conduit 99 and flows to the inlet of the pulsed ejector 10I.

The pulsed ejector 10I opens and closes to meter absorbent fluid flow back to the absorber 93, with the pulsed suction flow assisting the pump in moving fluid from the evaporator 36I to the absorber 93.

The solenoid-actuated pulsed ejector 10I is actuated by the solenoid coil 23K which fully opens the valve element 12I when electrically energized and fully closes the valve element 12I when deenergized. The differential pressure switch 24J regulates the energization and de-energization of the solenoid coil 23K. The electrical conduit 25L transfers electrical power between the electric contacts of the differential switch 24J and the solenoid coil 23K. The electrical conduit 26K supplies electrical power to the solenoid coil 23K through the electric contacts of the switch 24J and the electrical conduit 25L.

Electric power from the conduit 26K fully opens the pulsed ejector 10I when the contacts of the switch 24J complete an electrical circuit. When the electrical circuit is broken by the opening of the electrical contacts the solenoid coil 23K is deenergized and the pulsed ejector 10I returns to its normally closed condition.

The basic premise of the self-regulated pulsed actuation of the absorbent flow pulsed ejector 10I is as follows:

With the pulsed ejector 10H and pulsed ejector 10I normally closed, the pump 94 acts to lower the pressure within the suction side of the system, seeking to maintain an evaporator pressure and temperature as self-regulated by the superheat switch 24I as it meters refrigerant flow. The pulsed ejector 10I has the active criterion of metering absorbent fluid flow back to the absorber, at the absorber pressure that is fundamentally determined by the superheat switch 24I. With the expectation of a minimal amount of pressure drop from the evaporator to the absorber, pulsed ejector 10I has the active criterion of metering absorbent fluid back to the absorber at a pressure slightly lower than the evaporator pressure.

Take, for example, a pressure difference of 1 PSIA, pounds per square inch absolute, between the evaporator and the absorber; with the evaporator 1 PSIA higher than the absorber. To maintain this, one places the differential pressure switch 24J in communication with the evaporator pressure, in communication with the absorber pressure, and in communication with the absorber pulsed ejector 10I.

The conduit 28I transfers pressure information from within the valve-ejector transition section 13J, sensing absorber inlet pressure, to the differential pressure switch 24J. The conduit 29J transfers pressure information from within the valve-ejector transition section 13I, sensing evaporator inlet pressure, to the differential pressure switch 24J.

The differential pressure at which the differential pressure switch 24J is set to open the pulsed ejector 10I is chosen by design criterion. Thermodynamic criterion other than differential pressure may be utilized to actuate the pulsed ejector 10I. For example, a differential pressure switch seeking to maintain discharge side pressures, such as condenser outlet pressure and absorbent pressure at the inlet of absorbent pulsed ejector 10I.

As the pump 94 lowers the pressure in the suction side of the thermodynamic system, and heat energy is added to the evaporator 36I, the superheat switch 24I opens the pulsed ejector 10H when the differential pressure between the bulb 30I pressure and the system pressure at the valve-ejector transition section 13I, which can be related to a superheat, rises above the differential pressure setting of the superheat switch 24I, permitting flow of the thermodynamic fluid from the refrigerant discharge side of the thermodynamic system through the pulsed ejector 10H to the refrigerant suction side of the thermodynamic system; producing an internal system pressure rise within the suction side of the system.

The rise in pressure within the suction side of the system is first noted at the valve-ejector transition section 13I as it is the closest location to the mass influx through the pulsed ejector valve element 12H. The rise in suction pressure is thus sensed simultaneously by the superheat switch 24I through the conduit 28H and sensed by the differential pressure switch 24J through the conduit 29J.

When the differential pressure between the valve-ejector transition section 13I and the valve-ejector transition section 13J sensed by the differential pressure switch 24J rises above the differential pressure setpoint of the differential pressure switch 24J, the electric contacts of the differential pressure switch 24J close and the solenoid coil 23K energizes opening the valve element 12I within the pulsed ejector 10I, allowing thermodynamic fluid flow through the pulsed ejector 10I.

Absorbent fluid flowing through pulsed ejector 10I into the absorber 93 raises the absorber pressure, as sensed by the differential pressure switch 24J through the conduit 28I at the valve-ejector transition conduit 13J.

When the differential pressure between the valve-ejector transition section 13I and the valve-ejector transition section 13J sensed by the differential pressure switch 24J drops below the differential pressure setpoint of differential pressure switch 24J, the electric contacts of the differential pressure switch 24J open and the solenoid coil 23K deenergizes, closing the valve element 12I within the pulsed ejector 10I, and stopping absorbent fluid flow through the pulsed ejector 10I.

As before, refrigerant flowing through the evaporator 36I to the outlet conduit 32I can lower the temperature as sensed by the thermostatic bulb 30I, lowering its internal bulb pressure.

When the differential pressure sensed by the superheat switch 24I drops below the differential pressure setpoint of the superheat switch 24I, the electric contacts of the superheat switch 24I open and the solenoid coil 23J deenergizes closing the valve element 12H within the pulsed ejector 10H, and stopping refrigerant fluid flow through the pulsed ejector 10H.

With the pulsed ejector 10H closed the pump 94 lowers the suction side pressure, and heat addition to the evaporator 36I can raise the temperature sensed at the evaporator outlet conduit 32I by the thermostatic bulb 30I, the differential pressure sensed by the superheat switch 24I rises above its differential pressure setpoint, resulting in the reopening of the pulsed ejector 10H.

With the pulsed ejector 10I closed the pump 94 lowers the suction side pressure within the absorber to maintain the natural flow from evaporator to absorber, resulting in a drop in the differential pressure sensed by the differential pressure switch 24J below its differential pressure setpoint, resulting in the reopening of the pulsed ejector 10I.

As the pulsed ejector 10H alternates between fully open and fully closed conditions, refrigerant fluid alternately flows and does not flow within the thermodynamic system.

As the pulsed ejector 10I alternates between fully open and fully closed conditions, absorbent fluid alternately flows and does not flow within the thermodynamic system.

The high velocity pulse mass flow flowing through absorbent fluid pulsed ejector valve element 12I flows through the valve-ejector transition section 13J, through the ejector body 11J which entails flow through the ejector nozzle 14J, the ejector converging section 18J, the ejector throat/bore section 19J, the ejector diverging diffuser section 20J, the ejector outlet section 21J through the conduit 35J into the absorber 93.

High velocity pulse flow through the ejector nozzle 14J transfers momentum to fluid within the ejector suction port 17J, which draws refrigerant from the evaporator 36I through the ejector suction conduit 32I. The high velocity absorbent fluid flow and the entrained refrigerant flow from the suction port 17J combine to flow through the ejector sections 18J, 19J and 20J which function to recover some of the combined flow velocity as a pressure rise. The pressure rise achieved lowers the pumping pressure rise requirement of the pump 94 as the pulsed ejector 10I helps the pump move refrigerant flow from the evaporator to the absorber.

The combined flow leaves the ejector body 11J through the ejector outlet section 21J, flowing through the conduit 35J to the absorber 93. The absorption process within the absorber 93 is generally exothermic as refrigerant vapor is absorbed by absorbent fluid. The high velocity pulsed flows and the inherent mixing within the ejector body 11J can be a means of improving absorber performance.

Absorbent fluid leaves the absorber 93 through the conduit 100 in a state of relatively higher absorbed refrigerant concentration than the fluid entering pulsed ejector 10I. The higher concentration absorbent fluid is returned to the pump 94 by the conduit 100 to continue the thermodynamic cycle.

The pulsed ejector absorption refrigeration system shown in FIG. 7 differs from that shown in FIG. 6 in the relative orientations of the ejectors relative to the heat exchangers and the absorber. The pulsed ejector absorption refrigeration system in FIG. 7 includes a condenser overfeed ejector 50E, a pulsed ejector 10J feeding the evaporator 34L and providing for recirculation of refrigerant, and a pulsed ejector 10K feeding the absorber 88A and providing for recirculation of absorbent fluid.

The evaporator outlet conduit 32J in FIG. 7 is analogous to the evaporator outlet conduit 32I in FIG. 6 except that the evaporator outlet conduit 32J transfers refrigerant directly to the absorber 88A, as opposed to the case in FIG. 6 where the evaporator outlet conduit 32I transfers refrigerant directly to the pulsed ejector 10I prior to entering the absorber 88.

In FIG. 7 the pulsed ejector suction conduit 41L, serves to recirculate absorbent fluid leaving the absorber 93A through the conduit 100A. In contrast in FIG. 6 the pulsed ejector suction conduit 32I, serves to assist the pump 94 in providing refrigerant flow from the evaporator 36I to the absorber 93.

The spirit of the difference between the embodiments of FIG. 6 and FIG. 7 with respect to the absorber pulsed ejector is the following:

The pulsed ejector 10I in FIG. 6 is in a series orientation with respect to the absorber 93 and the pump 94 so that the pulsed ejector provides a partial pressure rise to the refrigerant prior to entering the absorber. The pulsed ejector 10K in FIG. 7 is in a parallel orientation with respect to the absorber 93A and the pump 94A so that the pulsed ejector provides an increased flow rate within the absorber 93A by recirculating absorbent fluid through the absorber 93A.

The pulsed ejector 10K in FIG. 7 may have its ejector suction conduit 41L attached at any point within the absorber 93A, accomplishing recirculation of absorbent fluid. In FIG. 7 the component parts with the same numerical prefix as the component parts in FIG. 6 function in an analagous fashion, respectively. The aforementioned component parts of FIG. 6 and FIG. 7 differ in their alphanumeric suffixes, respectively.

In FIG. 7 the ejector suction conduit 41K is shown recirculating fluid from what can be construed to be the lower header 38J of a parallel, vertical, evaporator 36J. The ejector suction conduit 62D is shown recirculating fluid from what can be construed to be the upper header 39I of a parallel, vertical, condenser 44B.

The simple pulsed condenser overfeed ejector 50D communicates fluid flow through conduit 58D to the lower header 45B of the parallel, vertical condenser 44B, causing internal recirculation flow as previously described.

The pulsed ejector 10J communicates fluid flow through conduit 35K to the upper header 39H of the parallel, vertical evaporator 36J, causing internal recirculation flow as previously described.

Anticipating pulsed flow reversal phenomenon within the ejector suction recirculation conduit 41L in FIG. 7, a check valve may be placed within that conduit to prevent reversed flow from the ejector suction port 17L back to the pump 94A through the conduit 100B.

In the spirit of the invention, the relative communication of the ejectors to the heat exchangers in can be in any combination of 'top feed', 'bottom feed', 'parallel', and 'series'. In the event that the heat exchangers utilized in a particular embodiment do not have headers, the relative communication of the ejectors to the heat exchangers can be in any combination of 'parallel' and 'series' with respect to the means provided to increase the pressure of and provide flow to the thermodynamic fluid within the thermodynamic system.

The non-steady-state intermittent flow through the nozzling devices in the present invention is a substantially isentropic nozzling process. The flow process through the throttling valves in steady-state systems of the prior art is a substantially isenthalpic throttling process. In a throttling device there is a distinct means of flow restriction that results in fluid flow losses and a generation of entropy while providing a pressure drop to steady-state flow. The flow restriction results in a negligible velocity increase as fluid experiences a drop in pressure and temperature in what is modeled thermodynamically as a constant enthalpy Joule-Thomson throttling expansion process. The Joule-Thomson expansion process is the classical basis of steady-state refrigeration, heat pump and air-conditioning cycles.

The nozzling devices are either fully open or fully closed with no intermediate positions, with minimal flow restriction in the fully open condition. The absence of flow restriction results in a substantially isentropic nozzling flow process and a substantial fluid velocity increase as fluid experiences non-steady-state flow and a pressure drop. The pressure difference between the inlet and the outlet of a nozzling device occurs when fully closed. Inlet and outlet system pressures tend towards equalization when the nozzling devices are fully open. Slight flow losses and small departures from ideal isentropic flow through the nozzling devices are to be expected, but not to the extent to which throttling devices are designed to produce flow restrictions.

Both pressure and enthalpy are transferred to kinetic energy as fluid flows through a nozzling device. Flow increases to subsonic, sonic, and supersonic velocities depending on operating conditions and nozzle design as thermodynamic entropy remains substantially constant. The isentropic nozzling expansion process, with the corresponding drop in pressure, temperature, and enthalpy, and the increase in velocity is the basis of non-steady-state refrigeration, heat pump, and air-conditioning cycles.

The utilization of the high velocity pulsed flows as a means to transfer momentum and provide flow to thermodynamic fluid within the system is within the scope of this invention, including but not limited to the incorporation of ejectors and suitably designed heat exchangers that experience internal recirculation flows as aresult of the high velocity pulsed flows.

The scope of the invention is to be determined from the following claims rather than the foregoing description of certain preferred embodiments.

Claims

1. A thermodynamic system wherein a thermodynamic fluid circulates through at least one heat exchanger with means for providing flow and a pressure rise to the fluid and a nozzling device which includes a nozzle and a valve automatically fully opened and closed in a binary fashion for creating accelerated intermittent high velocity bursts of substantially unrestricted thermodynamic fluid flow through the valve and nozzle within the system comprising

a) a pulsed ejector having a pulsed ejector suction port into which the thermodynamic fluid is directed from the nozzle,
b) a pulsed ejector suction conduit connecting the pulsed ejector suction port to the thermodynamic fluid within the system.

2. A thermodynamic system according to claim 1 the thermodynamic fluid discharged from the pulsed ejector as combined nozzle flow and pulsed ejector suction port flow creates internal recirculation flow within the at least one heat exchanger.

3. A thermodynamic system according to claim 1 whereby the thermodynamic fluid reverses flow direction within the system due to the opening and closing of the nozzling device valve.

4. A thermodynamic system according to claim 1, including at least one simple pulsed ejector comprising

a) a simple nozzle in communication with the simple pulsed ejector suction port of the simple pulsed ejector body,
b) a simple pulsed ejector suction conduit connecting the simple pulsed ejector suction port to the thermodynamic fluid within the system,

whereby the thermodynamic fluid flow through the system creates suction flow within the simple pulsed ejector suction port.

5. A thermodynamic system according to claim 4

a) a simple pulsed ejector discharge section communicating combined simple nozzle flow and simple pulsed ejector suction port flow to the system

whereby the thermodynamic fluid discharged from the simple pulsed ejector as combined simple nozzle flow and simple pulsed ejector suction port flow creates internal recirculation flow within the at least one heat exchanger.

6. A thermodynamic system according to claim 4, whereby the thermodynamic fluid reverses flow direction within the system due to the opening and closing of the nozzling device valve.

7. A thermodynamic system according to claim 1, wherein

a) the at least one heat exchanger comprises a plurality of substantially vertical tubes interconnected substantially in parallel by lower and upper headers,
b) when the nozzling device valve is open thermodynamic fluid flow from the nozzling device and pulsed ejector moves the thermodynamic fluid through the heat exchanger from the lower to the upper headers so that thermodynamic fluid interacts with substantially all heat exchange surfaces of the vertical tubes,
c) the pulsed ejector suction conduit recirculates thermodynamic fluid leaving the upper header of the heat exchanger back to the pulsed ejector when the nozzling device valve is open.

8. A thermodynamic system according to claim 1, wherein

a) the at least one heat exchanger comprises a plurality of substantially vertical tubes interconnected substantially in parallel by lower and upper headers,
b) when the nozzling device valve is open thermodynamic fluid flow from the nozzling device and pulsed ejector moves the thermodynamic fluid through the heat exchanger from the upper

to the lower headers so that thermodynamic fluid interacts with substantially all heat exchange surfaces of the vertical tubes,

c) the pulsed ejector suction conduit recirculates thermodynamic fluid leaving the lower header of the heat exchanger back to the pulsed ejector when the nozzling device valve is open.

9. A thermodynamic system according to claim 2, wherein

a) the at least one heat exchanger comprises a plurality of substantially vertical tubes interconnected substantially in parallel by lower and upper headers,

b) when the nozzling device valve is open thermodynamic fluid flow through the simple pulsed ejector moves the thermodynamic fluid through the heat exchanger from the lower to the upper headers,

c) the simple pulsed ejector suction conduit recirculates thermodynamic fluid leaving the upper header of the heat exchanger back to the simple pulsed ejector when the nozzling device valve is open.

10. A thermodynamic system according to claim 2, wherein

a) the at least one heat exchanger comprises a plurality of substantially vertical tubes interconnected substantially in parallel by lower and upper headers,

b) when the nozzling device valve is open thermodynamic fluid flow through the simple pulsed ejector moves the thermodynamic fluid through the heat exchanger from the upper to the lower headers,

c) the pulsed ejector suction conduit recirculates thermodynamic fluid leaving the upper header of the heat exchanger back to the pulsed ejector when the nozzling device valve is open.

11. A thermodynamic system according to claim 1 which includes sensing means for sensing at least one of pressure and temperature of the thermodynamic fluid in said system to open fully or close the valve in response to a change in at least one of said pressure and temperature.

12. A thermodynamic system according to claim 11 wherein said sensing means comprises a thermostatic bulb sensing thermodynamic fluid temperature in the system and a pressure tap sensing thermodynamic fluid pressure in the system to infinitely vary a setpoint at which said nozzling device valve opens and closes.

13. A thermodynamic system according to claim 11 wherein momentary temperature and pressure operating setpoints are relational and the magnitude of the sensed temperature and the sensed pressure at each moment of system operation varies as the system self-regulates with changing environment conditions.

14. In a thermodynamic system according to claim 1, means for selectively operating the system alternatively as a heat pump system comprising

a) reversing valve means for reversing the direction of thermodynamic fluid flow from a refrigeration mode to a heat pump mode, and

b) the at least one heat exchanger including a condenser and an evaporator, the condenser in the refrigeration mode functioning as an evaporator in the heat pump mode and the evaporator in the refrigeration mode functioning as a condenser in the heat pump mode when said reversal of direction of thermodynamic fluid flow occurs.

15. A thermodynamic system according to claim 14 wherein operation of the system in the heat pump mode functions to rapidly melt frost accumulated on the evaporator during operation in the refrigeration mode.

16. A thermodynamic system according to claim 14 wherein the reversing valve means are provided at the discharge of the compressor and the outlet of the pulsed ejector for reversing the direction of thermodynamic fluid flow from the refrigeration mode to the heat pump mode while utilizing a single nozzling device and pulsed ejector.

17. In a thermodynamic system according to claim 14, means for selectively operating the system alternatively as a heat pump system comprising

a) said nozzling device and pulsed ejector being duplicated as first and second nozzling devices and pulsed ejectors,

b) said reversing valve means for reversing the direction of thermodynamic fluid flow from a refrigeration mode to a heat pump mode being without check valves,

c) the first nozzling device and pulsed ejector metering thermodynamic fluid flow in one direction and the second nozzling device and pulsed ejector metering thermodynamic fluid flow in the opposite direction, and

d) the condenser in the refrigeration mode functioning as an evaporator in the heat pump mode and the evaporator in the refrigeration mode functioning as a condenser in the heat pump mode when the first nozzling device and pulsed ejector cease operation and the second nozzling device and pulsed ejector meter flow in said opposite direction.

18. A thermodynamic system according to claim 1 wherein by-pass reservoir means are provided for withdrawing thermodynamic fluid from the system in response to a reduction in system superheat and returning thermodynamic fluid to the system in response to an increase in system superheat, whereby as system superheat rises and falls thermodynamic fluid alternately enters and leaves the reservoir means and alternately enters and leaves the system.

19. A thermodynamic system according to claim 1 wherein the system is an absorption system and an absorbent fluid and a refrigerant fluid each circulates in its own flow circuit, said nozzling device and pulsed ejector being duplicated as an absorber nozzling device and absorber pulsed ejector and a refrigerant nozzling device and refrigerant pulsed ejector, and said at least one heat exchanger includes at least one of a refrigerant evaporator, a refrigerant condenser, and an absorber.

20. A thermodynamic system according to claim 19 wherein refrigerant fluid passes from the refrigerant evaporator to the absorber pulsed ejector before entering the absorber, whereby the absorber pulsed ejector is in series with respect to the absorber.

21. A thermodynamic system according to claim 19 wherein the refrigerant fluid circulates directly from the refrigerant evaporator to the absorber without passing through the absorber pulsed ejector, whereby the absorber pulsed ejector is in parallel with respect to the absorber, recirculating thermodynamic fluid from within the system to the absorber.

22. A thermodynamic system according to claim 2 wherein the system is an absorption system and an absorbent fluid and a refrigerant fluid each circulates in its own flow circuit, said nozzling device and pulsed ejector being duplicated as an absorber nozzling device and absorber pulsed ejector and a refrigerant nozzling device and refrigerant pulsed ejector, and said at least one heat exchanger includes at least one of a refrigerant evaporator, a refrigerant condenser, and an absorber.

23. A thermodynamic system according to claim 22 wherein refrigerant fluid passes from the refrigerant evaporator to the absorber pulsed ejector before entering the absorber, whereby the absorber pulsed ejector is in series with respect to the absorber.

24. A thermodynamic system according to claim 22 wherein the refrigerant fluid circulates directly from the refrigerant evaporator to the absorber without passing through the absorber pulsed ejector, whereby the absorber pulsed ejector is in parallel with respect to the absorber, recirculating thermodynamic fluid from within the system to the absorber.

25. A thermodynamic system wherein a thermodynamic fluid circulates from an evaporator to a compressor to a condenser and thence back to the evaporator through a nozzling device which includes a nozzle and a valve automatically fully opened and closed in a binary fashion to create accelerated intermittent high velocity bursts of substantially unrestricted thermodynamic fluid flow through the valve and nozzle within the system comprising

a) a thermostatic bulb sensing thermodynamic fluid temperature in the system, and
b) a pressure tap sensing thermodynamic fluid pressure in the system to infinitely vary a setpoint at which said nozzling device valve opens and closes.

26. A thermodynamic system wherein a thermodynamic fluid circulates from an evaporator to a compressor to a condenser and thence back to the evaporator through a nozzling device which includes a nozzle and a valve automatically fully opened and closed in a binary fashion to create accelerated intermittent high velocity bursts of substantially unrestricted thermodynamic fluid flow through the valve and nozzle within the system comprising

a) the evaporator including a plurality of substantially vertical evaporator tubes interconnected substantially in parallel by lower and upper evaporator headers, and
b) an evaporator pulsed ejector having an evaporator pulsed ejector port into which the nozzle thermodynamic fluid flow is directed, an evaporator pulsed ejector suction port conduit for connecting the evaporator pulsed ejector suction port to the system, and an evaporator pulsed ejector discharge section for communicating nozzle flow and pulsed ejector suction port flow to the lower evaporator header, the combined flow moving thermodynamic fluid upwardly through the evaporator tubes from the lower to the upper evaporator headers, substantially wetting all the tube heat transfer surface area,
c) whereby when the nozzling device valve is open pulse thermodynamic fluid flow creates recirculation flow within the evaporator, and thermodynamic fluid leaving the evaporator is recirculated to the evaporator by the pulsed evaporator ejector suction conduit and the

pulsed evaporator ejector suction port,

d) whereby as the nozzling device valve opens and closes thermodynamic fluid flow reverses direction within the evaporator and within the thermodynamic system.

27. A thermodynamic system wherein a thermodynamic fluid circulates from an evaporator to a compressor to a condenser and thence back to the evaporator through a nozzling device which includes a nozzle and a valve automatically fully opened and closed in a binary fashion to create accelerated intermittent high velocity bursts of substantially unrestricted thermodynamic fluid flow through the valve and nozzle within the system comprising

a) the condenser including a plurality of substantially vertical condenser tubes interconnected in parallel by lower and upper condenser headers, and

b) a condenser ejector having a condenser ejector suction port, a condenser ejector suction conduit, and a condenser ejector discharge section for directing thermodynamic fluid vapor from the compressor upwardly through the condenser tubes from the lower to the upper condenser headers,

c) whereby when the nozzling device valve is open carrying the thermodynamic fluid upwardly through the condenser tubes from the lower to the upper condenser headers so that thermodynamic fluid liquid condenses on inner surfaces of the condenser tubes and flows downwardly in counterflow relation to thermodynamic fluid vapor carried upwardly through the condenser tubes, and thermodynamic fluid is recirculated from the upper condenser header to the lower condenser header by the condenser ejector suction conduit and the condenser ejector suction port,

d) whereby as the nozzling device valve opens and closes thermodynamic fluid flow reverses direction within the condenser and within the thermodynamic system.

28. A thermodynamic system according to claim 27 wherein the upper condenser header is a dead-end header.

29. In a thermodynamic process wherein a heat exchange fluid is circulated, a method of continual thermodynamic efficiency self-optimization in real time as energy is exchanged in the process with an external environment which comprises

a) directing the heat exchange fluid through a valve and nozzle,

b) sensing the pressure of the heat exchange fluid in the system,

c) sensing the temperature of the heat exchange fluid in the system,

d) converting the sensed temperature to an equivalent sensed pressure, and

e) automatically opening fully or closing the valve in a binary fashion in response to a change in the relation between the sensed pressure and the sensed temperature thus permitting substantially unrestricted bursts of fluid flow through the valve and permitting acceleration of the intermittent bursts of fluid flow by the nozzle.

30. A method according to claim 29 wherein the opening and closing of the valve functions in a mechanical feedback loop utilizing internal pressure information and internal temperature information to self-regulate said opening and closing of the valve and flow through the nozzle.

31. A method according to claim 30 wherein the opening and closing of the valve permits pulses of fluid flow through the thermodynamic system which creates recirculation flow within the thermodynamic system heat exchangers and causes the thermodynamic fluid to reverse flow direction within the thermodynamic system.

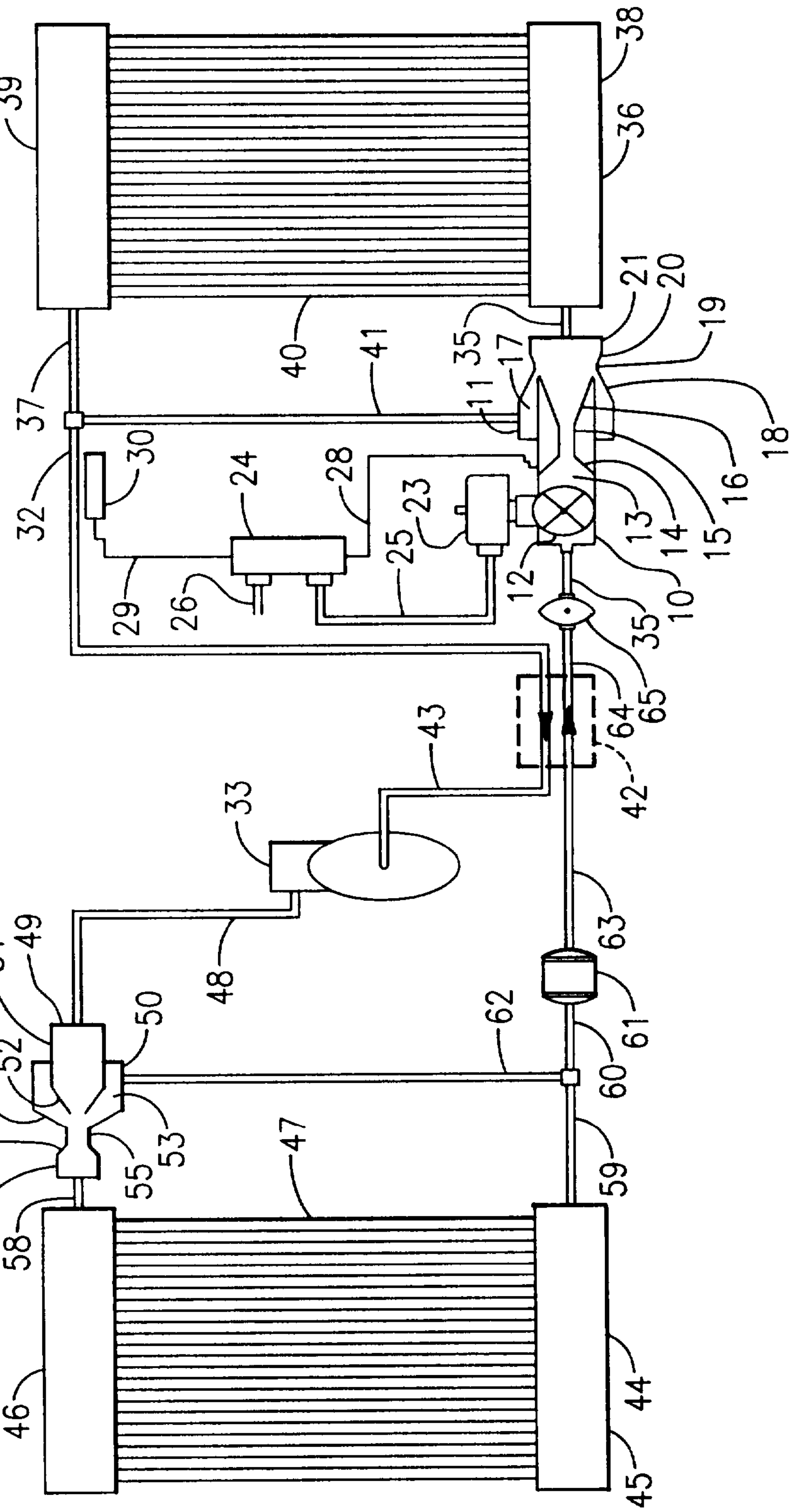
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FIG.1

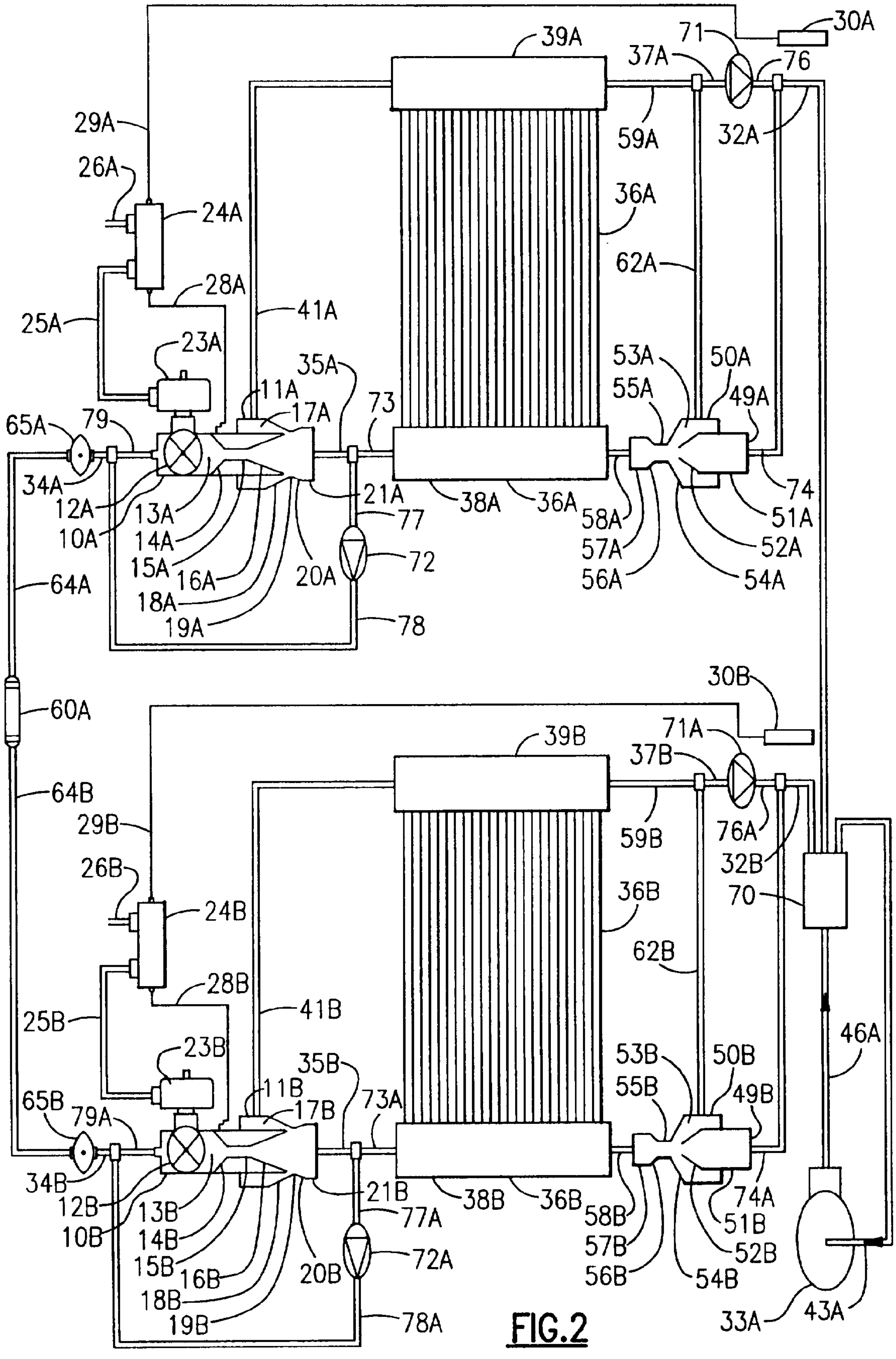
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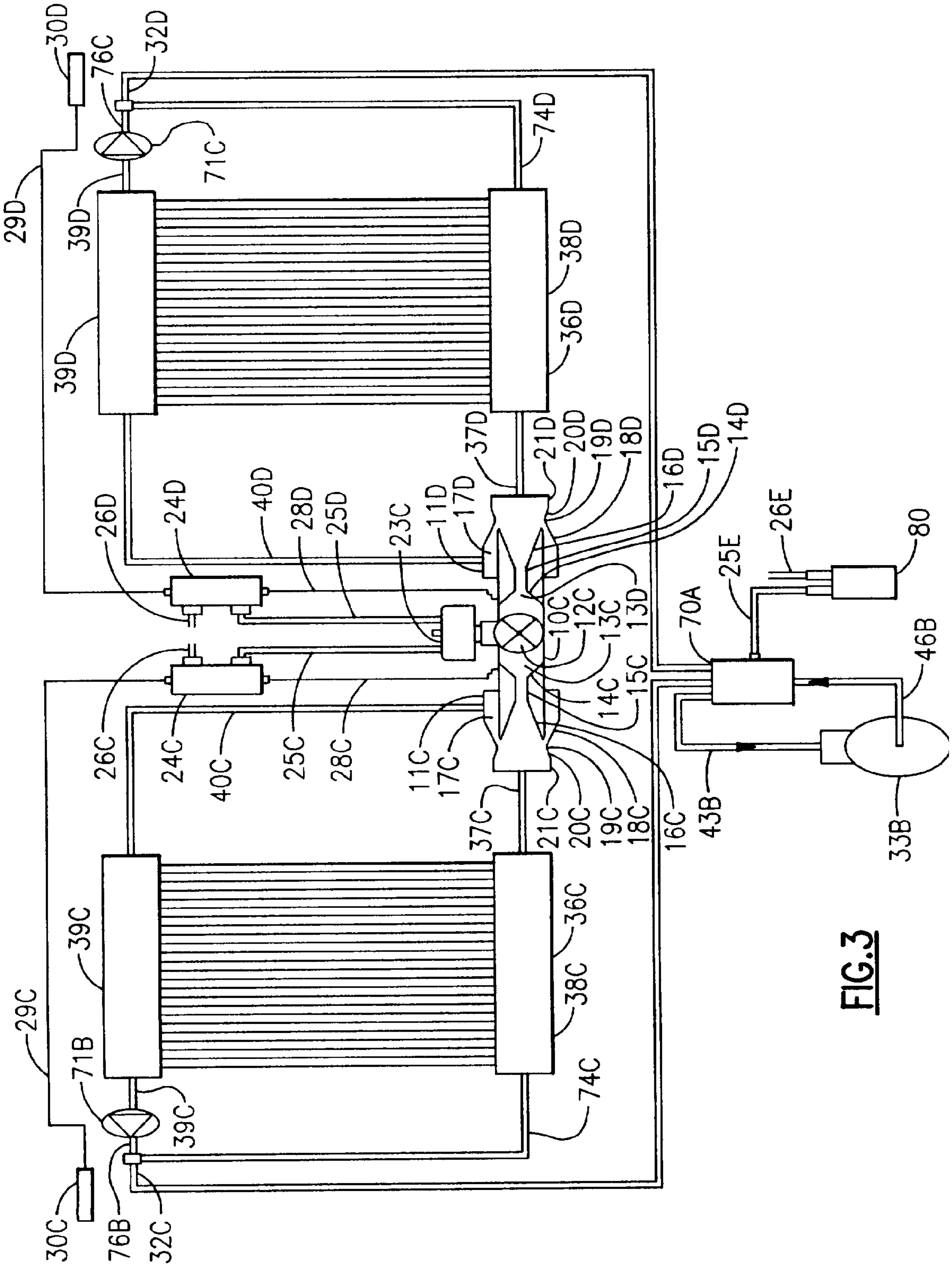
FIG.2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17D
10C
12C
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13D
70A
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26E
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46B
26C
24C
40C
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17C
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37C
21C
20C
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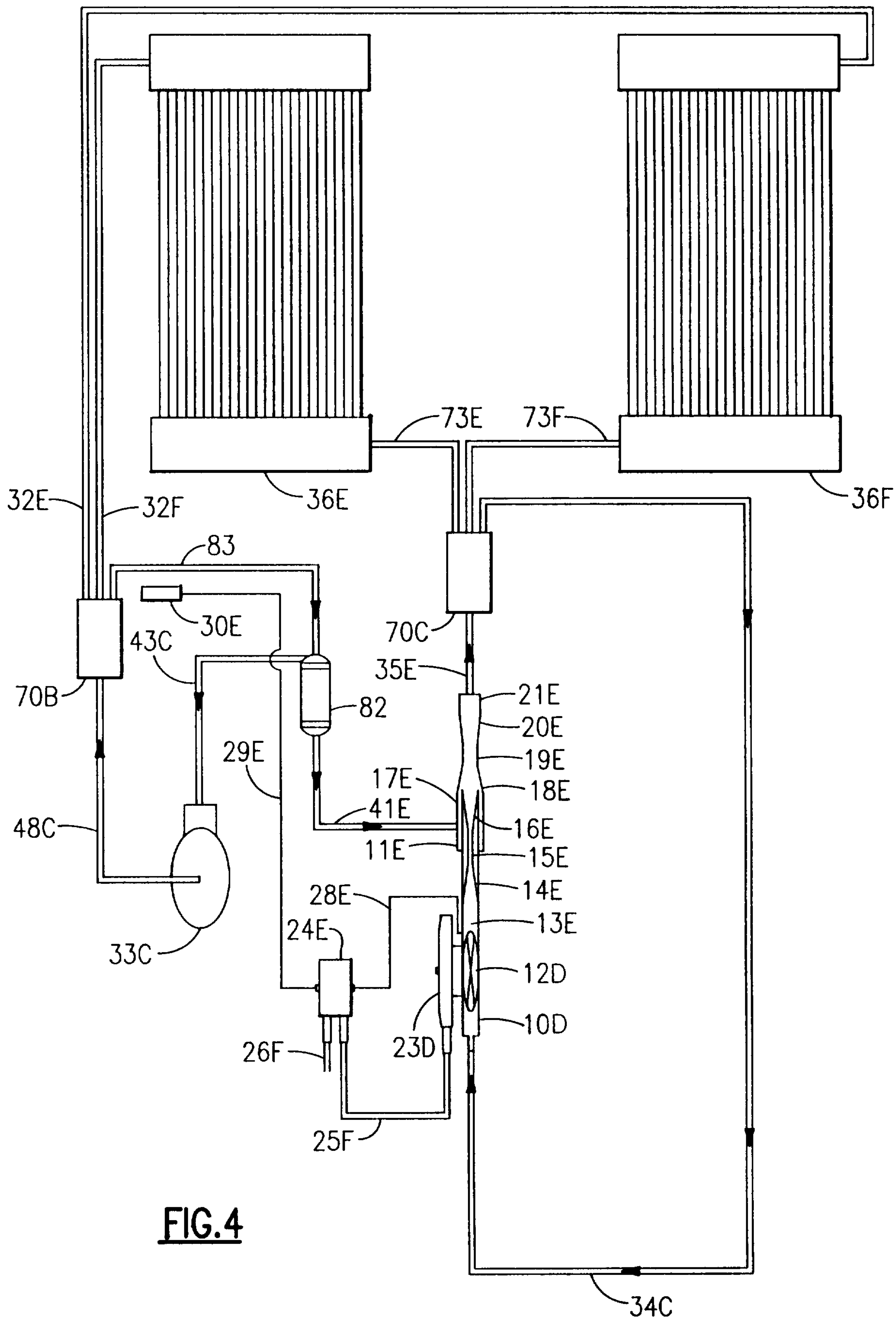
30C
76B
32C

FIG.3



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17E
16E
15E
14E
13E
12D
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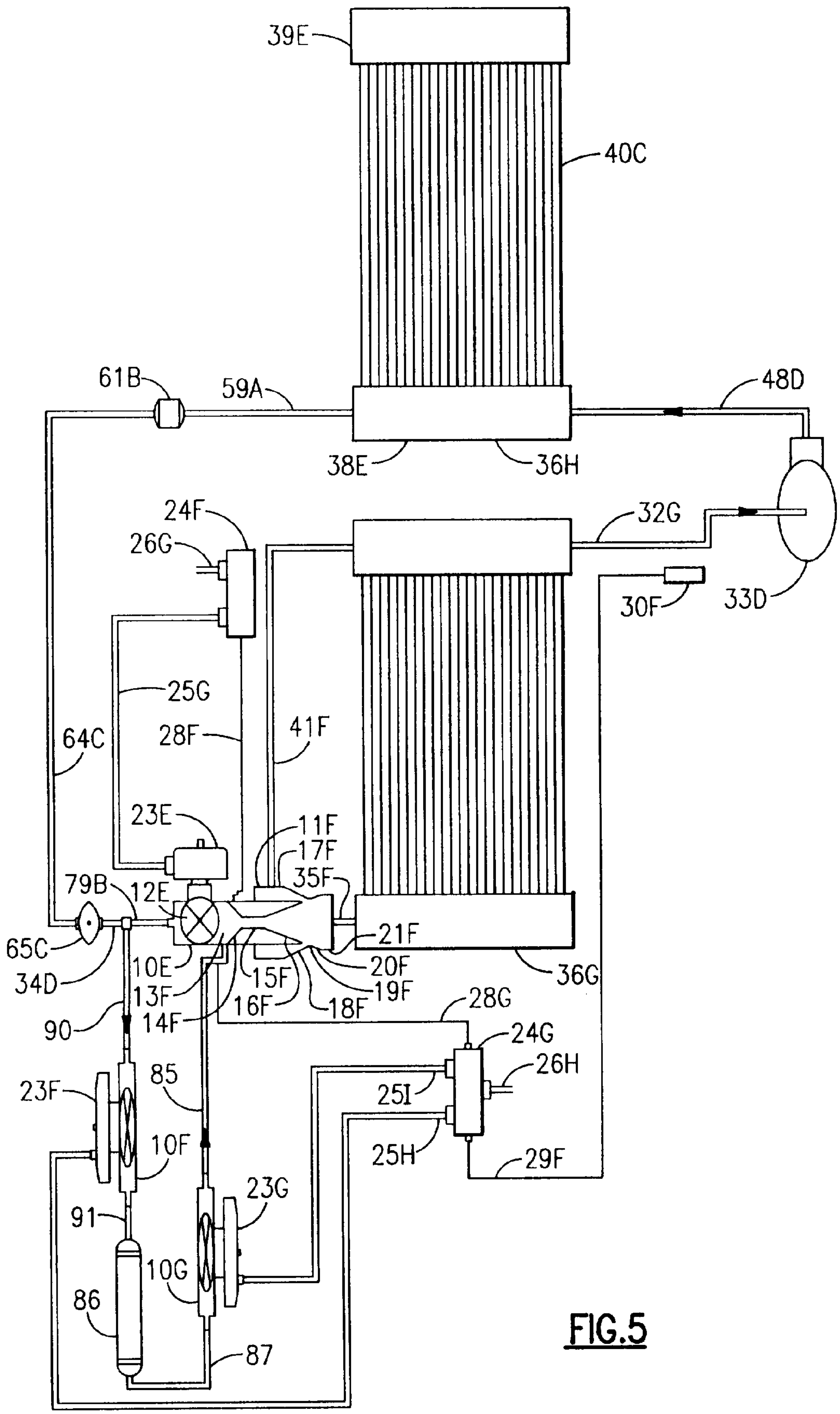
26F
23D
25F
34C

FIG.4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79B
12E
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65C
10E
20F
34D
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15F
19F
36G
16F
18F
28G
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23G
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10G
86
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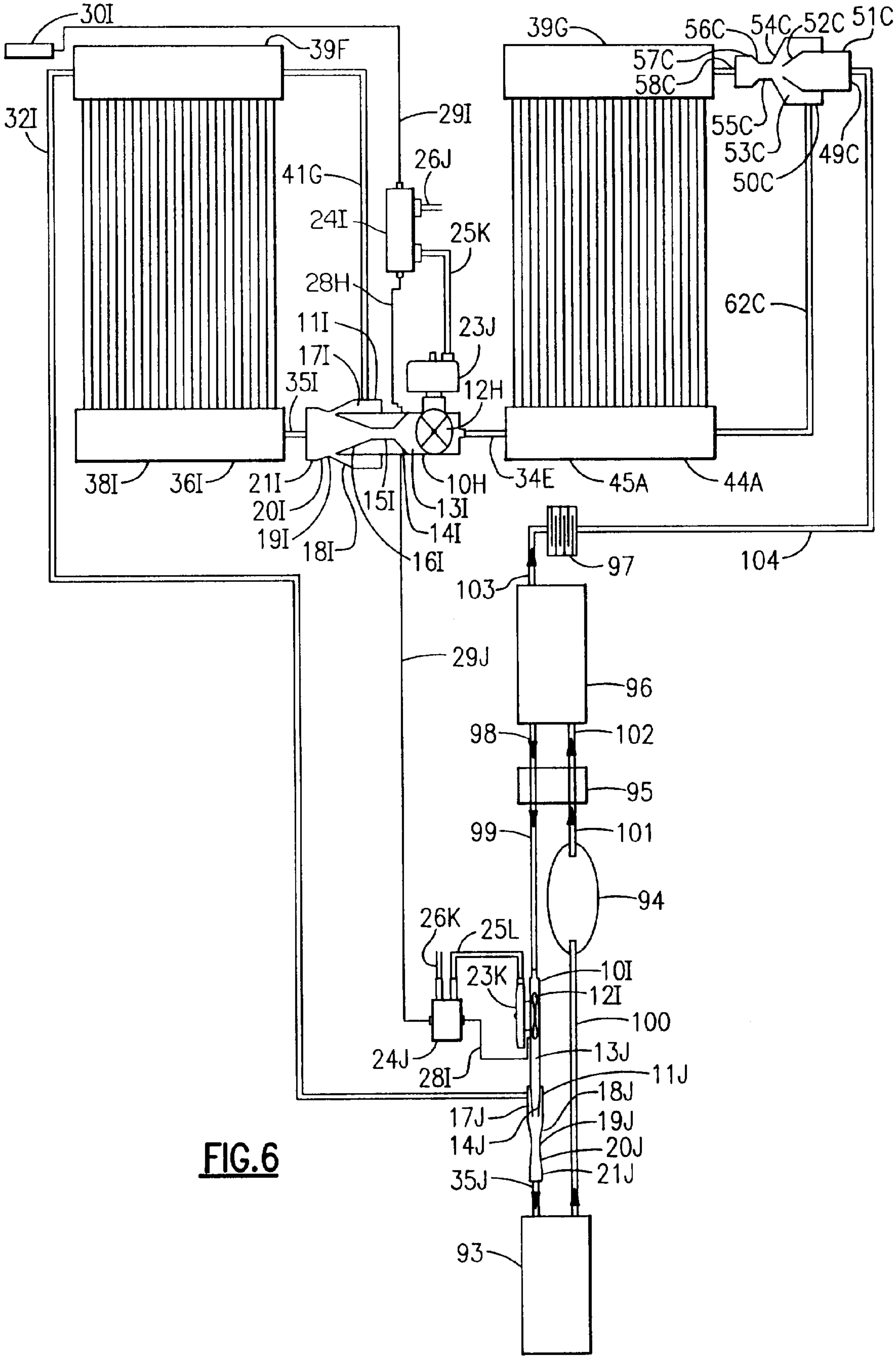
FIG.5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10H
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39G
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52C
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99
101
94
26K
25L
23K
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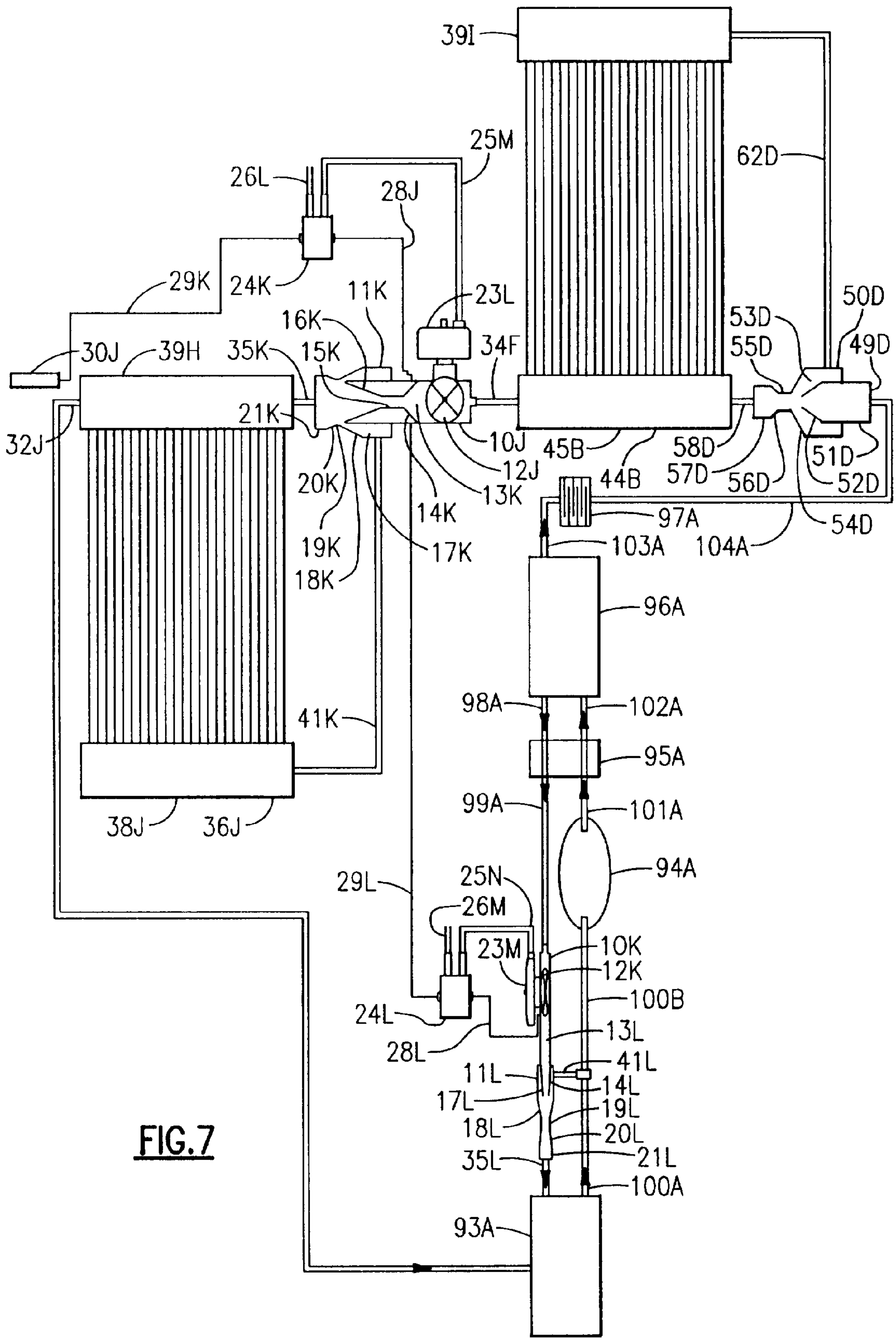
19J
14J
20J
35J
21J
93

FIG.6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32J
10J
45B
58D
51D
12J
20K
13K
44B
57D
56D
52D
14K
97A
54D
19K
17K
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104A
18K
96A
41K
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29L
25N
26M
23M
10K
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17L
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93A

FIG.7

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EUROPEAN SEARCH REPORT

Application Number
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DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X	WO-A-93 22605 (ALSENZ)	1,2,4,5, 11,12, 25,29	F25B41/00 F25B41/06
Y	* page 4, line 29 - page 14, line 15; figures 1-4 *	7-9,27	
Y	US-A-2 123 021 (PHILLIPS) * page 2, left column, line 1 - right column, line 35; figures 1,2 *	7,9	
Y	US-A-1 765 674 (INMAN) * page 1, line 56 - page 2, line 98; figures 1-6 *	8,27	
A	US-A-2 462 329 (MOJONNIER) * column 2, line 25 - column 5, line 68; figures 1-8 *	1,2,4,5, 7,9,26	
A	US-A-2 099 085 (SHRODE) * page 3, left column, line 59 - page 5, right column, line 8; figures 1-5 *	1,2,4,5, 7,9,26	TECHNICAL FIELDS SEARCHED (Int.Cl.6) F25B
A	DE-C-501 093 (KUPRIANOFF) * page 2, line 23 - line 96; figures 1-3 *	1,2,4,5, 8,10,26	
D,A	WO-A-94 21975 (CALMAC MANUFACTURING) * page 7, paragraph 2 - page 35, paragraph 2; figures 1-9 *	1,11,12, 14,17, 19,21, 25-27,29	

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The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	9 January 1996	Boets, A

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D : document cited in the application
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& : member of the same patent family, corresponding document

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EUROPEAN SEARCH REPORT

Application Number
EP 95 30 6791

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
A	US-A-4 621 502 (IBRAHIM) * column 5, line 36 - column 15, line 35; figures 1-7 *	1,11,12, 25,29	
D,E	US-A-5 452 585 (PINCUS) * column 3, line 6 - column 6, line 45; figures 1,2 *	25,29,30	
A	US-A-1 958 087 (HOFFMAN)		
A	US-A-3 667 244 (HOCK)		
A	US-A-3 103 106 (TIPTON)		

TECHNICAL FIELDS SEARCHED (Int.Cl.6)

The present search report has been drawn up for all claims

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