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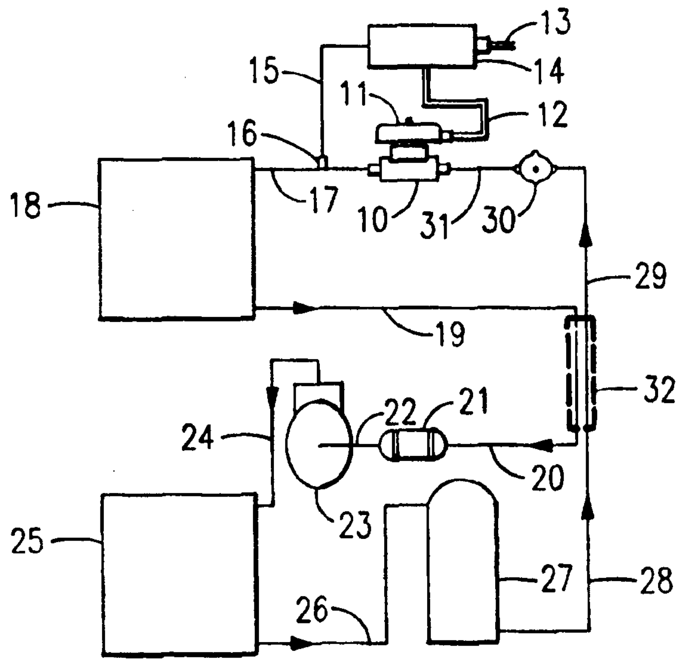
(54) Title: NON-STEADY-STATE SELF-REGULATING INTERMITTENT FLOW THERMODYNAMIC SYSTEM

(57) Abstract

There is disclosed a thermodynamic system comprising a compressor or pump (23), at least one heat exchanger, a conduit recirculating a heat exchange fluid through the system, at least one nozzling device (10) including a valve (47) and a nozzle (48, 49, 50), the valve (47) having only fully open and closed binary positions with no intermediate positions and causing minimal restriction to fluid flow when open, the nozzle (48, 49, 50) being configured to accelerate fluid flow to a maximum attainable velocity with minimum restriction to fluid flow, and means (14, 15, 16) sensing the pressure of the heat exchange fluid in said conduit to open fully or close the valve (47) in response to a change in pressure in the conduit to impart an intermittent operation to the valve (47) and permit intermittent substantially unrestricted acceleration of bursts of fluid flow through the nozzling device (10).



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NON-STEADY-STATE SELF-REGULATING INTERMITTENT FLOW THERMODYNAMIC SYSTEM

The most common refrigeration cycle used world-wide consists of a refrigerant vapor compressor a condenser changing 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. A great deal of research and development has gone into improving the compressor, the condenser, and the evaporator, whereas research on the expansion device has been commercially unsuccessful in recovering the energy lost during the expansion.

The prior art of refrigeration, heat pump and air-conditioning systems utilize metering systems that incorporate throttling devices that provide throttling of fluid flow through a substantial flow restriction. The most common metering systems produce steady-state throttling of the refrigerant as it flows. Thermodynamic processes are controlled by the pressure-flow characteristics of the throttling devices.

This invention relates to non-steady-state self-regulating intermittent flow refrigeration, heat pump, and air-conditioning systems. The refrigerant is transferred through the system in pulses, or bursts that recover the energy of expansion as improved heat transfer, mechanical compression work, and fluid flow work.

Non-steady-state metering systems incorporate nozzling devices that provide intermittent substantially unrestricted high velocity nozzling of the bursts of fluid flow. Nozzling devices include a valve and a nozzle. The valve is actuated based on internal system pressure, fully opening and closing in a binary fashion with no intermediate positions to provide intermittent substantially unrestricted acceleration of bursts of fluid flow through the nozzling device. The nozzle increases the velocity of fluid flow with minimal restriction.

Thermodynamic processes are self-regulated by the non-steady-state metering system as part of a mechanical feedback loop that provides for continual system self-optimization in real time as environment conditions change.

The thermodynamic model for the transfer of fluid through a nozzling device is described as follows :

(i) an isentropic nozzling expansion process in which a substantial increase in fluid velocity occurs,

(ii) fluid enthalpy drops, being converted to kinetic energy,

(iii) pressure and temperature drops,

(iv) entropy remains constant, indicating no loss of the recoverable work.

Transfer of fluid through a throttling valve in a stead-state-system is modelled as follows :

(i) a Joule-Thomson isenthalpic throttling expansion process in which a neglibible fluid velocity increase occurs,

(ii) fluid enthalpy remains constant, the potential energy associated with the pressure drop is converted to heat in the friction and flow restricting throttling process,

(iii) pressure and temperature drops,

(iv) entropy increases, indicating the loss of the recoverable work as heat.

By replacing isenthalpic, entropy generating throttling flow processes with isentropic nozzling flow processes, non-steady-state thermodynamic cycles are more efficient than steady-state thermodynamic cycles.

Non-stead-state metering systems can replace all current steady-state flow, pressure, and temperature throttling valve based regulation devices, and can be utilized in the less common thermodynamic systems. For example, the steady-state throttling expansion valves and flow regulating valves in absorption refrigeration systems can be replaced by the intermittent flow nozzling devices. The pulsed high velocity flows will improve the heat transfer within the heat exchangers, increase the cooling capacity, and reduce the pumping power requirements.

The invention relates to novel refrigeration, heat pump, and air-conditioning systems in which the thermodynamic fluid is internally transferred in an intermittent fashion. Rate and metering of flow is self-regulated by the thermodynamic system in a fashion modelled after a heart and its pressure regulation of a circulatory system. A heart will beat faster or slower to maintain blood pressure and flow. A nozzling device, as part of the non-steady-state metering system will open and close faster or slower as the thermodynamic system exchanges energy with its environment. The thermodynamic system opeates as a mechanical feedback loop, continuously self-optimising in real time as it seeks a minimum entropy generating equilibrium state.

The non-steady-state intermittent flow metering system includes a pressure switch that regulates the opening and closing of a mechanically-actuated nozzling device. The valve component of a nozzling device fully opens to provide substantially unrestricted fluid flow and minimal parasitic pressure drop, and closes to prevent fluid flow and enable the compressor or pump to create a pressure difference across the ports of the nozzling device. The nozzle element accelerates the fluid flow to the maximum attainable velocity with the minimum restriction to fluid flow as the fluid drops in pressure, temperature, and enthalpy while its entropy remains substantially constant. A nozzle can consist of straight, coverging, and diverging sections. Nozzles produce subsonic, sonic or supersonic fluid velocities at their outlets. The valve and nozzle elements of a nozzling device can be linked in series, or integrally composed.

A pressure tap into the thermodynamic system transfers pressure information to the pressure switch which regulates the opening and closing of the valve based on a pressure setpoint. The valve and nozzle provide intermittent nozzling of fluid flow to an outlet conduit that is sensed by the pressure tap, as part of the overall mechanical feedback loop of the entire system.

For an understanding of the present invention, reference should be made to the detailed description which follows and to the accompanying drawings, in which:

Figure 1 shows a schematic for an intermittent flow refrigeration or air-conditioning system;

Figures 2 and 3 show schematics for an intermittent flow heat pump system;

Figure 4 shows a schematic for an intermittent flow absorption refrigeration system;

Figure 5 shows a schematic for an intermittent flow refrigeration or air-conditioning system with variations on the intermittent flow metering unit;

Figure 6 shows a thermodynamic temperature-entropy diagram comparing a simplified non-steady-state intermittent flow thermodynamic cycle to a steady-state thermodynamic cycle;

Figure 7 shows a schematic of a nozzling device in which the valve precedes the nozzle;

Figure 8 shows a schematic of a nozzling device in which the nozzle precedes the valve;

Figure 9 shows a schematic of a nozzling device in which the valve is integrally composed with the nozzle.

In the refrigeration of air-conditioning system shown in Figure 1, actuation of nozzling device 10 is provided by a solenoid. Nozzling device 10 is actuated by solenoid coil 11 which fully opens the valve element when energized and fully closes the valve element when de-energized. Pressure switch 14 regulates the operation of solenoid coil 11. Electrical conduit 12 transfers power between the electric contacts of pressure switch 14 and solenoid coil 11. Electrical conduit 13 supplies power to solenoid coil 11 through the contacts of switch 14 and conduit 12. Power from conduit 13 fully opens nozzling device 10 when the contacts of switch 14 complete an electrical circuit between 13,12 and 11. When the circuit between 13,12 and 11 is broken by the opening of the contacts of switch 14, solenoid coil 11 is de-energized and nozzling device 10 returns to its normally closed conditions.

Conduit 15 transfers pressure information from downsteam of nozzling device 10 to pressure switch 14. Pressure information from within conduit 17 is transferred to conduit 15 by pressure tap 16. As compressor 23 lowers the pressure in the suction side of the system, pressure switch 14 opens nozzling device 10 when the pressure drops below the switch setting, permitting fluid to flow from within upstream conduit 31 through nozzling device 10 to downstream conduit 17. As the high velocity burst of fluid enters downstream conduit 17 it produces a pressure rise within the suction side of the system. When the pressure within conduit 17 is above the pressure switch setting the contacts of pressure switch 14 open and solenoid coil 11 de-energizes closing nozzling device 10 and stopping fluid flow through nozzling device 10. With nozzling device 10 closed compressor 23 lowers the suction side pressure until it is below the pressure switch setting, resulting in the re-opening of nozzling devide 10. As 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 evaporator heat exchanger 18 through conduit 17 and out through conduit 19 to counter-flow heat exchanger 32. Fluid flows out of counter-flow heat exchanger 32 through conduit 20 to filter-drier 21, and through conduit 22 to compressor 23. Counter-flow heat exchanger 32 serves to further lower the temperature of the refrigerant leaving heat exchanger 25 and entering nozzling device 10 by exchanging heat with the lower temperature refrigerant leaving heat exchanger 18. Counter-flow heat exchanger 32 may not be used in all applications.

Compressor 23 transfers mechanical energy to the fluid, increasing the pressure and temperature of the fluid and discharging it through conduit 24 to heat exchanger 25. Fluid flows out of heat exchanger 25 through conduit 26 to liquid reservoir 27 and out of liquid reservoir 27 through conduit 28 to counter-flow heat exchanger 32. The fluid that enters counter-flow heat exchanger 32 through conduit 28 in counter-flow heat relationship with fluid flowing from heat exchanger 18 to compressor 23 emerges through conduit 29 and returns to nozzling device 10 to complete a thermodynamic cycle. Sight glass 30 and connecting conduit 31 may be provided upstream of nozzling device 10 to indicate the quality of the refrigerant in the system. Sight glass 30 and liquid reservoir 27 are not required in all applications.

In the heat pump system shown in its cooling mode in Figure 2, the nozzling devices are actuated by solenoids.

System components 10,11,12,13,14,15,16,17,18,23 and 31 communicate and operate in an identical fashion to identical system components 10,11,12,13,14,15,16,17, 18,23 and 31 as referred to and described in Figure 1.

The high velocity burst of fluid flows into evaporator heat exchanger 18 through conduit 17 and out through conduit 19 to four-way reversing valve 34. Fluid flows out of reversing valve 34 through conduit 20 to filter-drier 21, and through conduit 22 to compressor 23.

Compressor 23 discharges the refrigerant through conduit 24 to reversing valve 34. Refrigerant flows out of reversing valve 34 through conduit 33 to condenser heat exchanger 25.

The refrigerant flows from heat exchanger 25 through conduit 17A through nozzling device 10A which is held fully open to allow continual unrestricted flow to nozzling device 10 to complete the thermodynamic cycle. Nozzling device 10A is held fully open by solenoid coil 11A. Solenoid coil 11A is electrically energized by power from electrical conduit 13A. Fluid flowing through nozzling device 10A enters conduit 31A and flows through sight glass 30 and conduit 31 to the inlet of nozzling device 10. Sight glass 30 is not required in all applications.

In the heat pump system shown in its heating mode in Figure 3, the nozzling devices are actuated by solenoids. Reversing valve 34 is in the heating mode, reversing the direction of refrigerant flow between heat exchangers 18 and 25 from the direction of flow indicated in Figure 2.

System components 10A,11A,12A,13A,14A,15A,16A, 17A,18 and 23 and 31A communicate and operate in an identical fashion to identical system components 10,11, 12,13,14,15,16,17,18,23 and 31 as referred to and described in Figure 2.

The high velocity burst of fluid flows into evaporator heat exchanger 25 through conduit 17A and out through conduit 33 to reversing valve 34. The fluid flows out of reversing valve 34 through conduit 20 to filter-drier 21, and through conduit 22 to compressor 23.

Compressor 23 discharges the refrigerant through conduit 24 to reversing valve 34. Refrigerant flows out of reversing valve 34 through conduit 19 to condenser heat exchanger 18.

The refrigerant flows from heat exchanger 18 through conduit 17 through nozzling devide 10 which is held fully open to allow for continual unrestricted flow to nozzling device 10A to complete the thermodynamic cycle. Nozzling device 10 is held fully open by solenoid coil 11. Solenoid coil 11 is energized by power from electrical conduit 13. Fluid flowing through nozzling device 10 enters conduit 31 and flows through sight glass 30 and conduit 31A to the inlet of nozzling device 10A. Sight glass 30 is not required in all applications.

In the absorption refrigeration system shown in Figure 4, the nozzling devices are actuated by solenoids.

System components 10,11,12,13,14,15,16,17,18 and 31 communicate and operate in an identical fashion to identical system components 10,11,12,13,14,15,16,17,18 and 31 as referred to and described in Figure 1 with pump 37 functioning to lower the pressure in the suction side of the thermodynamic system as compared to the compressor 23 in Figure 1.

The high velocity burst of fluid from nozzling device 10 flows into evaporator heat exchanger 18 through conduit 17 and out through conduit 19 to absorber 35. Vapor from evaporator 18 is absorbed by the liquid absorbent fluid within absorber 35 in an exothermic process, releasing heat energy to the external environment. Liquid absorbent fluid is pumped out of absorber 35 through conduit 36 by pump 37. Pump 37 raises the pressure of the liquid absorbent fluid and discharges it through conduit 38 to counter-flow heat exchanger 32A. Pressurized liquid absorbent fluid that enters counter-flow heat exchanger 32A through conduit 38 leaves through conduit 39 and enters vapor generator 40. Heat energy from a higher temperature ambient environment is transferred to vapor generator 40 so that refrigerant vapor is released from the absorbent fluid in an endothermic process. The high pressure refrigerant vapor leaves vapor generator 40 through conduit 41 and flows to rectifier 42 which functions as a desiccant to remove any water in the liquid or vapor phase from the refrigerant vapor. Dry refrigerant vapor leaves rectifier 42 through conduit 33 and enters condenser heat exchanger 25. Liquid refrigerant leaves condenser 25 through conduit 31 and flows to the inlet of nozzling device 10 to complete a thermodynamic cycle. In absorption refrigeration applications in which water is the refrigerant, rectifier 42 is not required.

High pressure liquid absorbent fluid from vapor generator 40 leaves through conduit 43 and flows through counter-flow heat exchanger 32A, where it transfers heat energy to absorbent fluid flowing from pump 37 to vapor generator 40, preheating the absorbent fluid before it enters vapor generator 40. Liquid absorbent fluid entering counter-flow heat exchanger 32A through conduit 43 leaves counter-flow heat exchanger 32A through conduit 31A and flows to the inlet of nozzling device 10A. Nozzling device 10A regulates absorbent fluid flow back to absorber 35. Absorbent fluid continually cycles through its system loop in an intermittent fashion as pressure switch 14A response to internal pressure information, alternately opening and closing nozzling device 10A.

System components 10A,11A,12A,13A,14A,15A,16A, 17A and 31A communicate and operate in an identical fashion to identical system components 10,11,12,13,14,15, 16,17 and 31 as referred to and described in Figure 4.

With respect to Figure 5, replacement of steady-state throttling valve based pressure, temperature, and flow regulation devices with non-steady-state metering devices is accomplished by utilizing the appropriate pressure switch in conjunction with a nozzling device, and by the appropriate placement of pressure taps within the thermodynamic system. Some non-steady-state metering system configurations are as follows :

(1) The normally closed nozzling device opens on a decrease in downstream pressure below the pressure switch setting.

(2) The normally closed nozzling device opens on an increase in downstream pressure above the pressure switch setting.

(3) The normally closed nozzling device opens on a decrease in upstream pressure below the pressure switch setting.

(4) The normally closed nozzling device opens on an increase in upstream pressure above the pressure switch setting.

(5) The normally closed nozzling device opens on a decrease in differential pressure between the upstream and downstream pressures.

(6) The normally closed nozzling device opens on an increase in differential pressure between the upstream and downstream pressures.

(7) The normally open nozzling device closes on a decrease in downstream pressure below the pressure switch setting.

(8) The normally open nozzling device closes on an increase in downstream pressure above the pressure switch setting.

(9) The normally open nozzling device closes on a decrease in upstream pressure below the pressure switch setting.

(10) The normally open nozzling device closes on an increase in upstream pressure above the pressure switch setting.

(11) The normally open nozzling device closes on a decrease in differential pressure between the upstream and downstream pressures.

(12) The normally open nozzling device closes on an increase in differential pressure between the upstream and downstream pressures.

Figure 5 shows a non-steady-state vapor-compression refrigeration or air-conditioning system utilizing some of the variations of non-steady-state metering devices for temperature, pressure, and flow regulation. The variations consist of: dual evaporators, each with evaporator pressure regulating devices on their respective downstream sides, a crankcase pressure regulating device on the inlet of the compressor on its suction side, a condenser pressure regulating device on the outlet of the condenser, and a differential pressure regulating device that bypasses high pressure fluid from the compressor outlet directly to the downstream side of the condenser pressure regulating device.

In the vapor-compression refrigeration or air-conditioning system shown in Figure 5, mechanical actuation of the nozzling devices is provided by solenoids. Nozzling device 10 regulates the flow of a refrigerant to evaporator heat exchanger 18. Nozzling device 10A regulates the flow of a refrigerant to evaporator heat exchanger 18A. Nozzling devices 10B and 10C regulate the pressures in heat exchangers 18 and 18A respectively, allowing for different operating pressures within each heat exchanger. Refrigerant flows from the outlet of nozzling devices 10B and 10C through filter-drier 21 to nozzling device 10D which regulates the pressure of the refrigerant flowing to compressor 23. Compressed refrigerant flows to condenser heat exchanger 25. Nozzling device 10E regulates the flow of refrigerant from and the pressure within heat exchanger 25. Nozzling device 10F bypasses refrigerant from the outlet of compressor 23 to the downstream side of nozzling device 10E, functioning as a differential pressure bypass of heat exchanger 25. Counter-flow heat exchanger 32 provides for heat exchange between the fluid flowing from the outlet of heat exchangers 18 and 18A to compressor 23 and the fluid that flows from the outlet of heat exchanger 25 to nozzling devices 10 and 10A. Refrigerant continually cycles through the system in an intermittent fashion as the pressure switches 14,14A,14B,14C,14D,14E and 14F respond to internal pressure information, alternately opening and closing the nozzling devices 10,10A,10B,10C,10D,10E and 10F respectively.

System components 10,11,12,13,14,15,16,17,18,23 and 31 communicate and operate in an identical fashion to identical system components 10,11,12,13,14,15,16,17, 18,23 and 31 as referred to and described in Figure 1.

System components 10A,11A,12A,13A,14A,15A,16A, 17A,18A, 23 and 31A communicate and operate in an identical fashion to identical system components 10,11, 12,13,14,15,16,17,18,23 and 31 as referred to and described in Figure 1.

System components 10B,11B,12B,13B,14B,15B,16B, 17B,18,23 and 31B communicate and operate in an identical fashion to identical system components 10,11, 12,13,14,15,16,71,18,23 and 31 as referred to and described in Figure 1 except that nozzling device 10B opens on an increase in pressure within upstream conduit 31B above the pressure switch 14B pressure setting, functioning as an evaporator pressure regulating device.

Conduit 15B transfers pressure information from upstream of nozzling device 10B to pressure switch 14B. Pressure information from within conduit 31B is transferred to conduit 15B by pressure tap 16B. As nozzling device 10 opens allowing refrigerant to enter heat exchanger 18 and raise its pressure, pressure switch 14B opens nozzling device 10B when the pressure rises above the switch setting, permitting flow of the fluid from within upstream conduit 31B through nozzling device 10B to downstream conduit 17B. As the high velocity burst of fluid leaves upstream conduit 31B it produces a pressure drop within heat exchanger 18. When the pressure within heat exchanger 18 is below the pressure switch setting the electric contacts of pressure switch 14B open and solenoid coil 11B de-energises closing nozzling device 10B and stopping fluid flow through nozzling device 10B. The pressure at which nozzling device 10B is set to close should be below the pressure at which nozzling device 10 is set to open so that nozzling device 10 can open before nozzling device 10B closes. With nozzling device 10B closed heat transfer into heat exchanger 18 and refrigerant flowing from the opening of nozzling device 10 raises the pressure within heat exchanger 18 until it is above the pressure switch setting of pressure switch 14B, resulting in the reopening of the nozzling device 10B. As nozzling device 10B alternates between fully open and fully closed conditions, fluid alternately flows and does not flow out of heat exchanger 18.

System components 10C,11C,12C,13C,14C,15C,16C, 17C,18A,23 and 31C communicate and operate in an identical fashion to identical system components 10B,11B, 12B,13B,14B,15B,16B,17B,18, 23 and 31B as referred to and described in Figure 4.

The high velocity bursts of fluid flowing out of heat exchangers 18 and 18A through conduits 17B and 17C converge into a common conduit 19 and flow to counter-flow heat exchanger 32. The fluid flows out of counter-flow heat exchanger 32 through conduit 20 to filter-drier 21, and through conduit 31D to the inlet of nozzling device 10D.

System components 10D,11D,12D,13D,14D,15D,16D, 17D,23 and 31D communicate and operate in an identical fashion to identical system components 10,11,12,13,14,15, 16,17,23 and 31 as referred to and described in Figure 1 except that nozzling device 10D is normally open when solenoid coil 11D is de-energized, and nozzling device 10D closes when solenoid coil 11D is energized, functioning as a compressor crankcase pressure regulating device.

Electric power from conduit 13D fully closes the nozzling device 10D when the contacts of switch 14D complete an electrical circuit between 13D,12D and 11D. When the circuit between 13D,12D and 11D is broken by the opening of the contacts of switch 14D, the solenoid coil 11D is de-energized and nozzling device 10D returns to its normally open condition. Pressure switch 14D closes nozzling device 10D when the pressure rises above the switch setting, stopping flow of the fluid from within upstream conduit 31D through nozzling device 10D to downstream conduit 17D. As fluid ceases to flow from upstream conduit 31D compressor 23 lowers the pressure within conduit 17D. When the pressure within conduit 17D is below the pressure switch setting the contacts of pressure switch 14D open and solenoid coil 11D deenergizes, opening nozzling device 10D and allowing unrestricted fluid flow through nozzling device 10D. As nozzling device 10D alternates between fully open and fully closed conditions, fluid alternately flows and does not flow into compressor 23.

Compressor 23 discharges refrigerant through conduits 24 and 33 to condenser heat exchanger 25. Fluid in the liquid state flows out of heat exchanger 25 through conduit 31E to the inlet of nozzling device 10E.

System components 10E,11E,12E,13E,14E,15E,16E, 17E, 23 and 31E communicate and operate in an identical fashion to identical system components 10B,11B,12B,13B, 14B,15B,16B,17B,23 and 31B as referred to and described in Figure 5 except that nozzling device 10E functions as a condenser pressure regulating device.

With nozzling device 10E closed refrigerant flowing into heat exchanger 25 from the compressor 23 raises the pressure within heat exchanger 25 until it is above the pressure switch setting of pressure switch 14E, resulting in the reopening of nozzling device 10E. As nozzling device 10E alternates between fully open and fully closed conditions, fluid alternately flows and does not flow out of heat exchanger 25.

High pressure discharge of fluid from compressor 23 through conduit 24 can bypass heat exchanger 25 by flowing through conduit 31F to the inlet of nozzling device 10F. Nozzling device 10F is actuated by solenoid coil 11F which fully opens the valve element when electrically energized and fully closes the valve element when de-energized. Differential pressure switch 14F regulates the operation of solenoid coil 11F. Electrical conduit 12F transfers power between the contacts of pressure switch 14F and the solenoid coil 11F. Electrical conduit 13F supplies power to the solenoid coil 11F through the contacts of switch 14F and conduit 12F. Power from conduit 13F fully opens the nozzling device 10F when the contacts of switch 14F complete an electrical circuit between 13F,12F and 11F. When the circuit between 13F,12F and 11F is broken by the opening of the contacts of switch 14F, solenoid coil 11F is de-energized and nozzling device 10F returns to its normally closed condition.

Conduit 15F transfers pressure information from downstream of nozzling device 10F to differential pressure switch 14F. Conduit 15G transfers pressure information from upstream of nozzling device 10F to differential pressure switch 14F. Upstream pressure information from within conduit 31F is transferred to conduit 15G by pressure tap 16G. Downstream pressure from within conduit 17F is transferred to conduit 15F by pressure tap 16F. Differential pressure switch 14F opens nozzling device 10F on a rise in differential upstream to downstream pressure above the switch setting, permitting flow of fluid from within the upstream conduit 31F through nozzling device 10F to downstream conduit 17F. As the high velocity burst of fluid leaves upstream conduit 31F it tends to equalize the pressure within downstream conduit 17F. When the difference in pressure between upstream conduit 31F and downstream conduit 17F is below the differential pressure switch setting the contacts of pressure switch 14F open and solenoid coil 11F deenergizes closing nozzling device 10F and stopping fluid flow through nozzling device 10F. With nozzling device 10F closed refrigerant flowing into upstream conduit 31F from compressor 23 raises the pressure within conduit 31F until the differential between the pressure within conduit 31F and conduit 17F is above the differential pressure switch setting of pressure switch 14F, resulting in the reopening of nozzling device 10F. As nozzling device 10F alternates between fully open and fully closed conditions, fluid alternately flows and does not flow to bypass heat exchanger 25.

The high velocity bursts of fluid flowing out of nozzling devices 10E and 10F through conduits 17E and 17F converge into a common conduit 26 and flow to liquid receiver 27. Fluid flows out of liquid receiver 27 through conduit 28 to counter-flow heat exchanger 32. The fluid that enters counter-flow heat exchanger 32 through conduit 28 in counter-flow heat relationship with the fluid flowing from heat exchangers 18 and 18A to compressor 23 emerges through conduit 29. Refrigerant from conduit 29 bifurcates into two paths, one through conduit 44 to sight glass 30 to conduit 31 to the inlet of nozzling device 10 and the other through conduit 45 to sight glass 30A to conduit 31A to the inlet of nozzling device 10A. Counter-flow heat exchanger 32 serves to further lower the temperature of the refrigerant leaving heat exchanger 25 and entering nozzling devices 10 and 10A by exchanging heat with the lower temperature refrigerant leaving heat exchangers 18 and 18A. Refrigerant flowing to the inlets of nozzling devices 10 and 10A complete a thermodynamic cycle.

With respect to Figure 6, the non-steady-state intermittent flow through the nozzling devices in the present invention is an isentropic nozzling process.

The flow process through the throttling valves in steady-state systems of the prior art is an 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 modelled 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 an 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.

An isentropic thermodynamic process is more thermodynamically efficient than a non-isentropic, entropy generating thermodynamic process. Non-steady-state intermittent flow thermodynamic cycles have fundamentally higher efficiencies than steady-state thermodynamic cycles.

Figure 6 shows a simple thermodynamic temperature-entropy diagram comparing a non-steady-state intermittent flow thermodynamic cycle denoted by process steps [10'10"]-[11',11"]-[12'12"]-[13',13"]-[10',10"], with each bracket representing pressure ranges, to a steady-state thermodynamic cycle, denoted by process steps 10-11-12-13-10. The abscissa, denoted by 15, represents entropy. The ordinate axis denoted by 14 represents temperature. The two cycles are demarcated by a vapor dome. The non-steady-state work, heat and mass transfer processes are represented in a simplified fashion due to their far-from-equilibrium nature.

The non-steady-state thermodynamic cycle process step [10′,10"]-[11′,11"] represents an isentropic nozzling flow process with a corresponding drop in temperature, pressure, and enthalpy from the pressure range [10′,10"] to the pressure range [11′,11"]. Range [10′-10"] represents the nozzle inlet and condenser outlet isentropic pressure, temperature and enthalpy transient drop when the nozzling device is fully open, and the corresponding rise when the nozzling device is fully closed and the compressor transfers mass to the discharge side of the system. Range [11′-11"] represents the nozzle outlet and evaporator inlet isentropic pressure, temperature and enthalpy transient rise when the nozzling device is fully open, and the corresponding drop when the nozzling device is fully closed and the compressor removes mass from the suction side of the system.

The non-steady-state thermodynamic cycle process step [11′,11"]-[12′,12"] represents a non-isothermal, non-isobaric phase change accompanying the non-steady-state mass and heat transfer within the evaporator heat exchanger. Fluid changes state from the range of mixed liquid-vapor states represented by [11′-11"] to the range of states represented by [12′-12"]. Ranges [11′-11"] and [12′-12"] represent an evaporator inlet and an evaporator outlet isentropic pressure rise respectively as a high velocity mass transfer occurs when the nozzling device fully opens, and an isentropic pressure drop respectively when the nozzling device fully closes and the compressor removes mass from the evaporator. Range [12′-12"] also represents the compressor inlet pressures. The evaporator outlet pressure range [12′-12"] can be substantially less than the evaporator inlet pressure range [11′-11"] as the compressor continually removes mass from the evaporator outlet while intermittent high velocity mass flow enters the evaporator inlet at a rate faster than the compressor can react to remove it. When the nozzling device is fully closed, the compressor lowers the pressure in the evaporator until it reaches the setpoint at which the nozzling device is set to open by actuation from the pressure switch.

The non-steady-state thermodynamic cycle process step [12′,12"]-[13′,13"] represents an isentropic compression process with a corresponding increase in temperature, pressure and enthalpy from the range of states [12′-12"] to the range of states [13′-13"] which are in the superheated vapor region. Range [13′-13"] represents the compressor outlet and the condenser heat exchanger inlet. There is a transient isentropic temperature, pressure and enthalpy rise from 13′ to 13" as the nozzling device is fully closed due to the action of the compressor. When the nozzling device opens there is a transient isentropic temperature, pressure, and enthalpy drop from 13" to 13′ as an intermittent high velocity mass transfer occurs from the high pressure to low pressure side of the system faster than the compressor can maintain pressure and flow. The non-steady-state compressor power and energy use is represented by a mathetmatical integration of the non-steady-state isentropic pressure rise from range [12′,12"] to range [13′,13"].

The non-steady-state thermodynamic cycle process step [13′,13"]-[10′,10"] represents a non-isobaric state change from superheated vapor to saturated vapor states followed by a non-isobaric, non-isothermal phase change from saturated vapor to saturated liquid states followed by a non-isobaric state change from saturated liquid to subcooled liquid states. Ranges [10′-10"] and [13′-13"] represent a condenser outlet and a condenser inlet isentropic pressure drop respectively as a high velocity mass transfer occurs when the nozzling device fully opens, and an isentropic pressure rise respectively as the compressor replaces the mass within the condenser. The condenser inlet pressure range [13'-13"] can be substantially less than the condenser outlet pressure range [10'-10"] as the compressor continually inputs mass to the condenser inlet while intermittent high velocity mass flow leaves the condenser outlet at a rate faster than the compressor can react to replace it.

The rate at which the nozzling device opens and closes is self-determined by the non-steady-state thermodynamic system as the compressor lowers the suction side pressure and raises the high side pressure to provide for an accommodate the amount of and rate of heat energy transferred by the cooling and heating heat exchangers. The mechanical feedback system continuously self-optimizes in real time as the thermodynamic system seeks a minimum entropy generating equilibrium with its external and internal environment.

The steady-state thermodynamic cycle process step 10-11 represents an isenthalpic expansion process with a corresponding decrease in temperature and pressure and increase in entropy.

The steady-state thermodynamic cycle process step 11-12 represents an isothermal, isobaric heat absorption evaporation phase change from mixed vapor-liquid phase to saturated vapor phase followed by an isobaric heat absorption process from saturated vapor state to superheated vapor state with a corresponding increase in temperature.

The steady-state thermodynamic cycle process step 12-13 represents an isentropic compression process with a corresponding increase in temperature, pressure and enthalpy.

The steady-state thermodynamic cycle process step 13-10 represents an isobaric heat rejection state change from superheated vapor to saturated vapor followed by an isobaric, isothermal heat rejection condensation phase change from saturated vapor to saturated liquid followed by an isobaric heat rejection state change from saturated liquid to subcooled liquid.

The non-steady-state nozzling expansion process [10′,10"]-[11′-11"] enables the recovery of the energy available to do work in the pressure difference between the condenser and the evaporator. The steady-state throttling expansion process 10-11 dissipates the available energy as an internal heat generation within the throttling restriction, lowering the available cooling capacity of the refrigerant within the evaporator.

The non-steady-state evaporator heat absorption process [11′,11″]-[12′,12″] has more effective heat transfer and a higher heat transfer rate than the steady-state process 11-12 due to the higher kinetic energy, lower enthalpy, and lower entropy at states [11′-11″] than the corresponding fluid at state 11.

The non-steady-state compression process [12′,12″]-[13′,13″] requires less energy than the steady-state process 12-13 is that due to the intermittent compressor pressure and flow work requirements and recovery of the expansion flow work. While the steady-state compressor continually maintains a pressure rise from state 12 and state 13, the non-steady-state compressor is able to cycle the high and low side system pressure between the ranges [13′-13″] and [12′-12″] respectively.

The non-steady-state condenser heat release process [13′,13″]-[10′,10″] has more effective heat transfer than the steady-state process 13-10 due to the intermittent high velocity fluid flows within the heat exchanger. This results in the increased sub-cooling of state 10′ when compared to state 10.

Herein lies the basis for the increased efficiency of the non-steady-state thermodynamic cycle when compared to the steady-state thermodynamic cycle; by replacing an isenthalpic throttling process with an isentropic nozzling process, a non-steady-state thermodynamic cycle requires lower energy and power use and provides improved heat transfer and increased heat transfer rate.

In the nozzling devices depicted in Figure 7, Figure 8 and Figure 9, the mechanical valve element 47 is a schematic representation of the valve element referred to within the nozzling devices of the previously described thermodynamic systems. Straight conduit section 49 and diverging conduit section 50 are simple schematic representations of the straight and diverging sections of a straight-diverging nozzle. Nozzling device inlet 46 and nozzling device outlet 51 function as transition elements for connecting to inlet and outlet conduits respectively.

With respect to Figure 7, formation of valve inlet 46, mechanical valve element 47 and valve outlet 48 could be a complete and separate unit. Valve inlet 46 functions as the inlet to the nozzling device and as a transition element for connection to an inlet conduit. Valve outlet 48 functions as a transition element for connection with the straight nozzle section 49, functioning as the nozzle inlet as well. Straight nozzle section 49 is integrally formed with diverging nozzle section 50 to produce a complete straight-diverging nozzle. Nozzle outlet 51 functions as the outlet to the nozzling device and as a transition element for connection to an outlet conduit. The nozzle and the valve are attached in series with respect to fluid flow, with the valve preceding the nozzle.

With respect to Figure 8, the formation of valve inlet 48, mechanical valve element 47 and valve outlet 51 could be a complete and separate unit. Valve outlet 51 functions as the outlet to the nozzling device and a transition element for connection to an outlet conduit. Valve inlet 48 functions as a transition element for connection with the diverging nozzle section 50, functioning as the nozzle outlet as well. Straight nozzle section 49 is integrally formed with diverging nozzle section 50 to produce a complete straight-diverging nozzle. Nozzle inlet 46 functions as the inlet to the nozzling device and as a transition element for connection to an inlet conduit. The nozzle and the valve are attached in series with respect to fluid flow, with the nozzle preceding the valve.

With respect to Figure 9, the nozzle and valve inlet and outlet elements are congruent within the body of the nozzling device. Nozzling device inlet 46 and nozzling device outlet 51 serve as transition elements for connection to inlet and outlet conduits respectively. The inlet to mechanical valve element 47 is simultaneously a valve inlet and the straight nozzle section 49. The outlet to mechanical valve element 47 is simultaneously a valve outlet and the divering nozzle section 50. Transition element 48 incorporates the mechanical valve element 47 as a transition between the straight and diverging sections of the nozzle.

Nozzle elements can take the form of straight conduit sections, converging conduit sections, and diverging conduit sections. The nozzling devices depicted in Figure 7, Figure 8 and Figure 9 can include the following nozzle inlet-outlet combinations: straight-straight, straight-converging, straight-diverging, converging-straight, converging-converging, and converging-diverging. Nozzles optimize the acceleration of fluid flow, to attain the highest velocity possible with minimal or neglible pressure drop and flow restriction. The pressure drop across the nozzling device results from the closed valve condition with the compressor or pump operating, and not from flow restriction within the nozzling device.

The invention has been shown in preferred forms and by way of example and modifications and variations are possible within the spirit of the invention.

The invention, therefore, is not intended to be limited to any specified form or embodiment, except insofar as such limitations are expressly set forth in the claims.

CLAIMS

1. A thermodynamic system comprising a compressor or pump, at least one heat exchanger, a conduit recirculating a heat exchange fluid through the system, at least one nozzling device including a valve and a nozzle, the valve having only fully open and closed binary positions with no intermediate positions and causing minimal restriction to fuid flow when open, the nozzle being configured to accelerate fluid flow to a maximum attainable velocity with minimum restriction to fluid flow, and means sensing the pressure of the heat exchange fluid in said conduit to open fully or close the valve in response to a change in pressure in the conduit to impart an intermittent operation to the valve and permit intermittent substantially unrestricted acceleration of bursts of fluid flow through the nozzling device.

2. A thermodynamic system as set forth in claim 1, wherein the nozzling device comprises a mechanical valve element and an associated nozzle composed of elements which include at least one of straight, converging and diverging sections that provide for the acceleration of fluid flow with minimal restriction.

3. A thermodynamic system as set forth in claim 1, including a solenoid for moving the valve between the

fully open and closed positions and a pressure controlled switch responsive to the sensing means to operate the solenoid to fully open and close the valve in response to a change in pressure in the conduit.

4. A thermodymamic system as set forth in claim 1, in which the sensing means senses the pressure in the conduit in at least one of the following locations:

(i) downstream of the nozzling device,

(ii) upstream of the nozzling device, and

(iii) both upstream and downstream of the nozling device.

5. A thermodynamic system as set forth in claim 1, in which the system includes at least two heat exchangers, one receiving heat from the heat exchange fluid and the other supplying heat to the heat exchange fluid, and in which the compressor is connected in the system by the conduit intermediate the two heat exchangers, the heat exchanger communicating with the discharge of the compressor being the source of the heat exchange fluid supplied to the nozzling device.

6. A thermodynamic system as set forth in claim 1, in which the system as a whole functions in a mechanical feedback loop utilizing internal pressure information to regulate the opening and closing of the nozzling device,

providing for continual thermodynamic efficiency self-optimization in real time as the system exchanges energy with its external environment.

7. A thermodynamic system as set forth in claim 1, including at least one nozzling device intermediate two heat exchangers, the sensing means being in communication with the conduit adjacent each nozzling device for controlling the operation of the nozzling device, and at least one reversing valve for changing the direction of flow of the heat exchange fluid through the heat exchangers.

8. A thermodynamic system as set forth in claim 1, including nozzling devices upstream and downstream respectively of a heat exchanger regulating the intermittent substantially unrestricted acceleration of heat exchange fluid flow entering and leaving the heat exchanger.

9. A thermodynamic system as set forth in claim 1, including a plurality of heat exchangers arranged in parallel, and respective nozzling devices for regulating the intermittent substantially unrestricted acceleration of heat exchange fluid flow to each heat exchanger.

10. A thermodynamic system as set forth in claim 1, including at least two of said nozzling devices arranged

in parallel for regulating the intermittent substantially unrestricted acceleration of fluid flow from the heat exchanger and bypassing the heat exchanger.

11. A thermodynamic system as set forth in claim 1, including a nozzling device upstream of the compressor or pump for regulating the intermittent substantially unrestricted acceleration of heat exchange fluid flow the compressor or pump.

12. A thermodynamic system as set forth in claim 2, wherein the mechanical valve element is joined in series with the nozzle.

13. A thermodynamic system as set forth in claim 2, wherein the mechanical valve element is integrally formed with the nozzle.

14. A thermodynamic system comprising a compressor or pump, at least one heat exchanger, a conduit recirculating a heat exchange fluid through the system, at least one nozzling device including a mechanical valve element and an associated nozzle, the valve element having only fully open and closed binary positions with no intermediate positions and causing minimal restriction to fluid flow when open, a solenoid for moving the valve element between its fully open and

closed positions, the nozzle being configured to accelerate fluid flow to a maximum attainable velocity with minimum restriction to fluid flow, means for sensing the pressure of the heat exchange fluid in said conduit, a pressure controlled switch responsive to a change in the sensed pressure to intermittently operate the solenoid to fully open and close the valve element.

15. In a thermodynamic process wherein a heat exchange fluid is circulated, a method for 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, and

(c) automatically opening fully or closing the valve in a binary fashion in response to a change in the sensed pressure thus permitting substantially unrestricted bursts of fluid flow through the valve and permitting acceleration of the intermittent bursts of fluid flow by the nozzle, whereby maximum attainable velocity with minimum restriction is achieved in the fluid flow through the nozzle.

16. A method according to claim 15, wherein the opening and closing of the valve functions in a

mechanical feedback loop utilizing internal pressure information to self-regulate said opening and closing of the valve and flow through the nozzle.

17. A thermodynamic system comprising a compressor, at least one heat exchanger, a conduit recirculating a heat exchange fluid through the system, at least one nozzling device through which flow is substantially isentropic, a means sensing at least one thermodynamic property in association with the system, the sensing means self-regulating the actuation of at least one nozzling device based on a setpoint as the system exchanges energy with its environment.

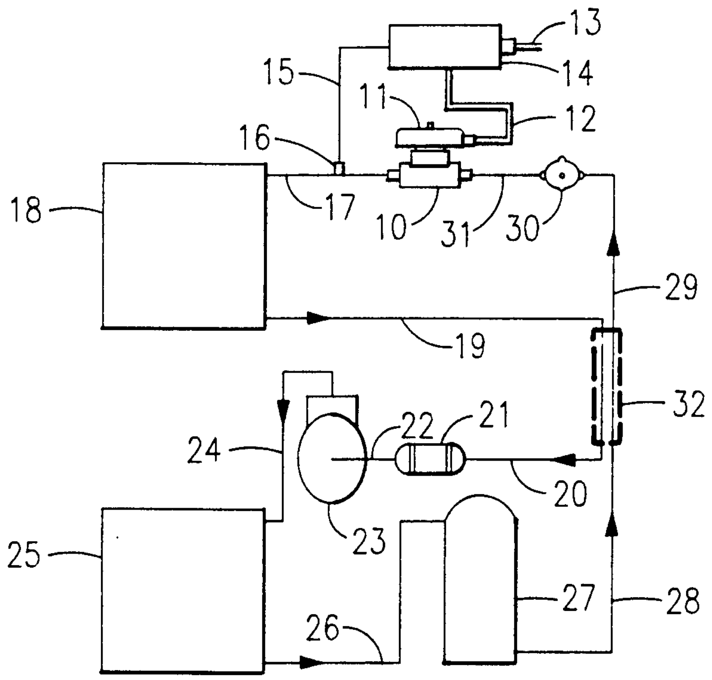
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FIG.1

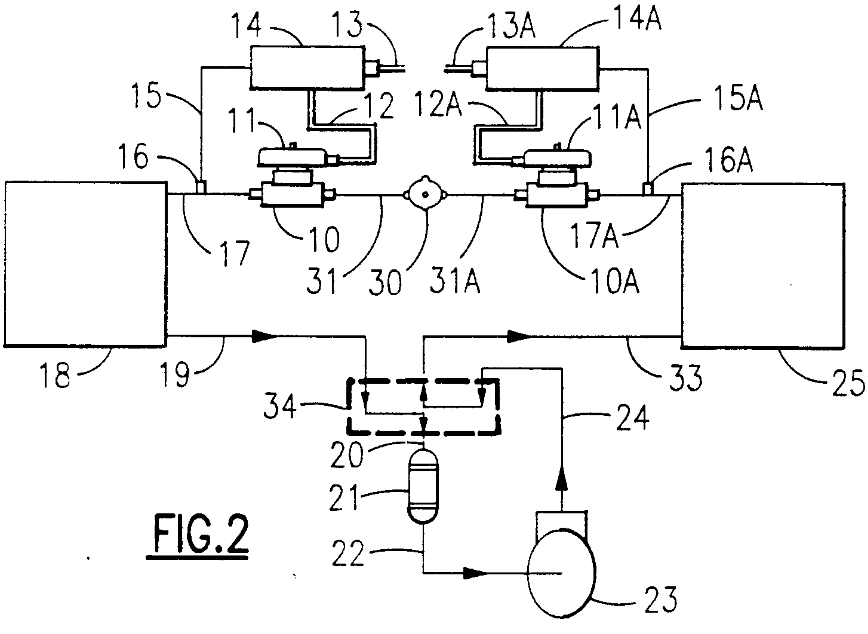
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FIG.2

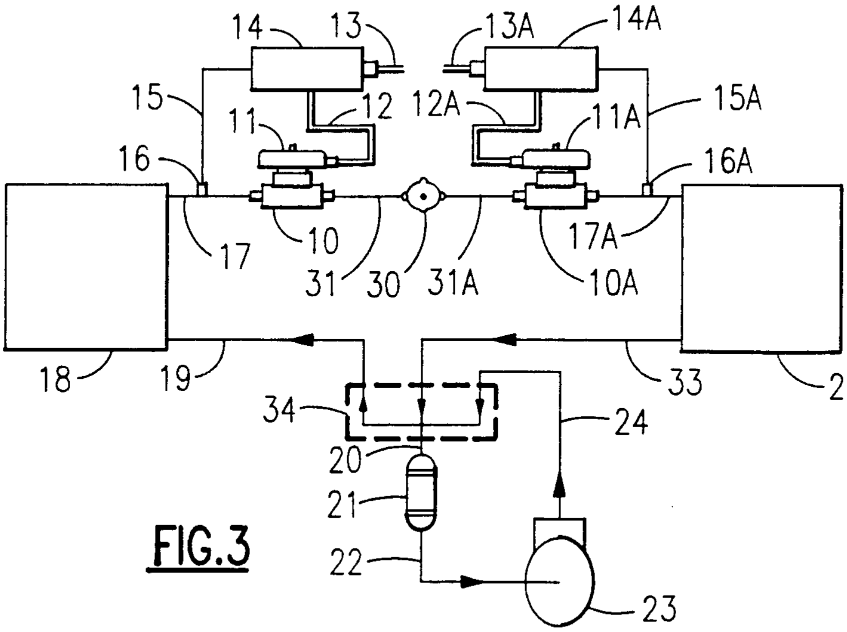
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FIG.3

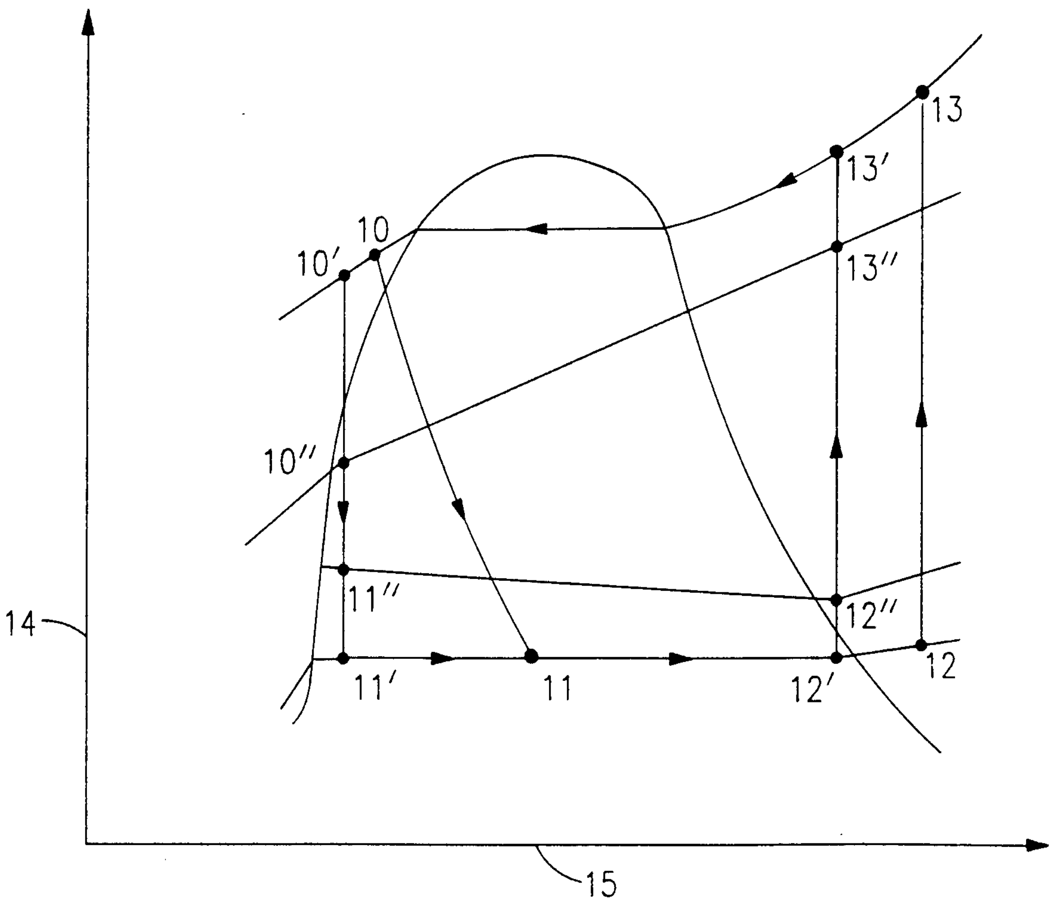
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FIG.6

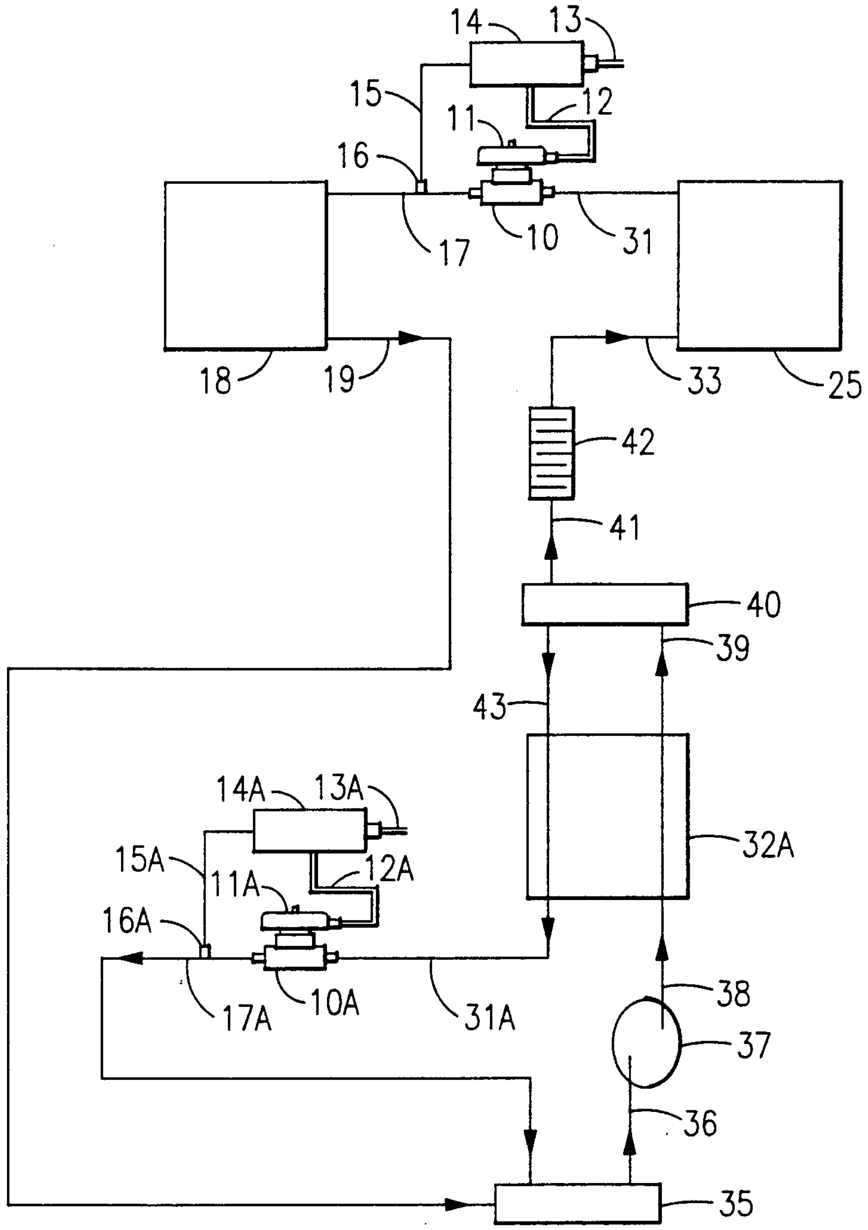
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FIG.4

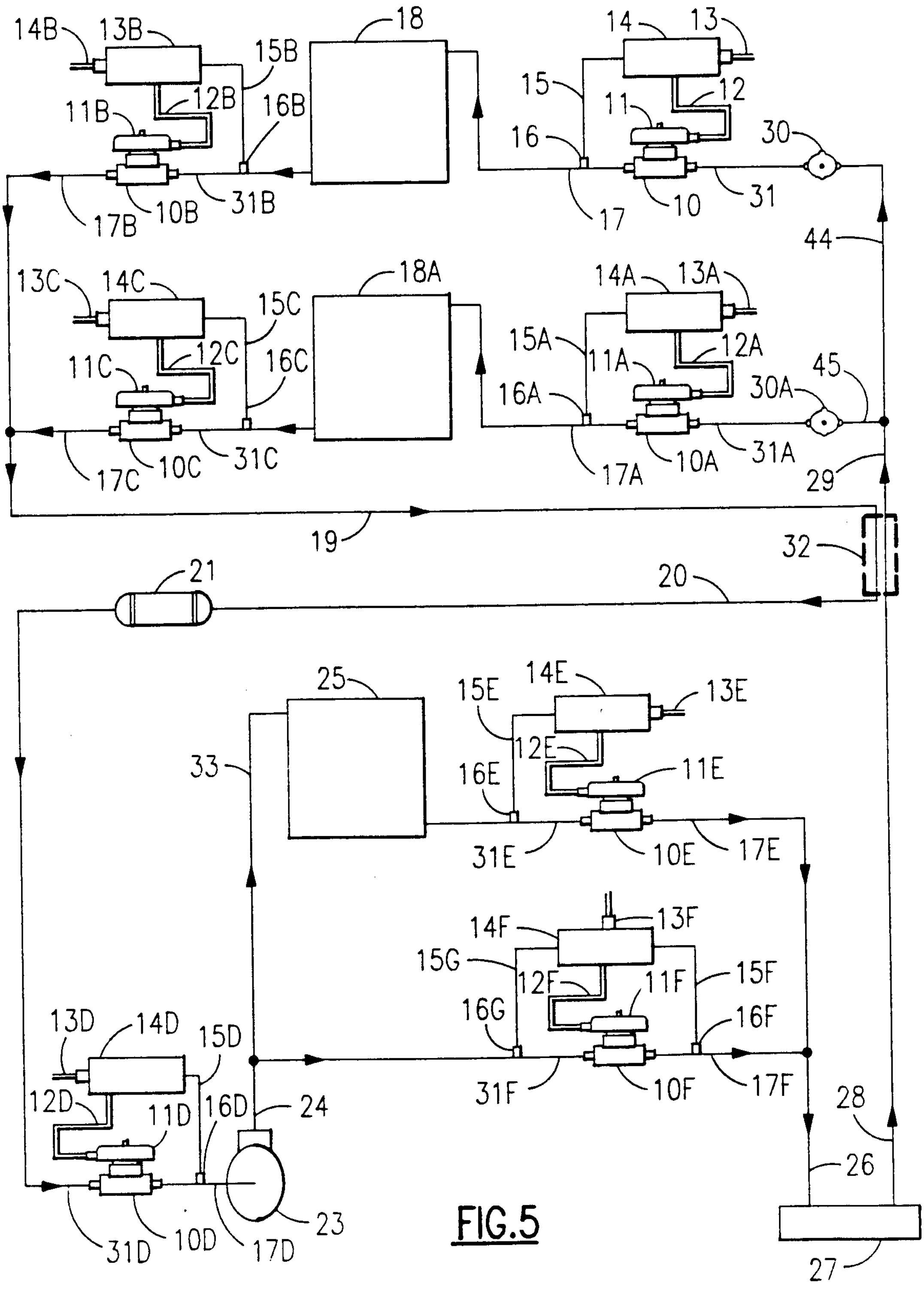
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FIG.5

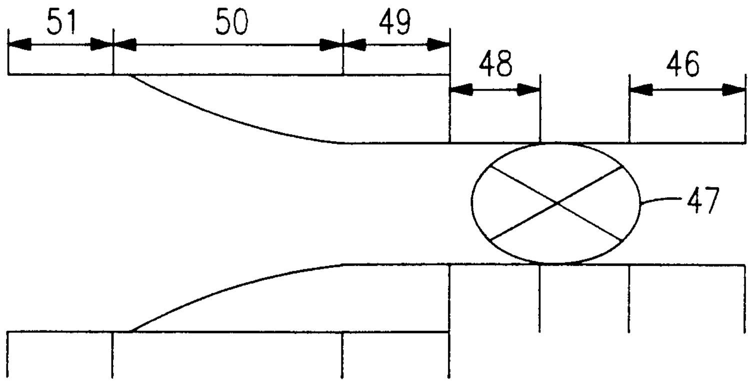

FIG.7

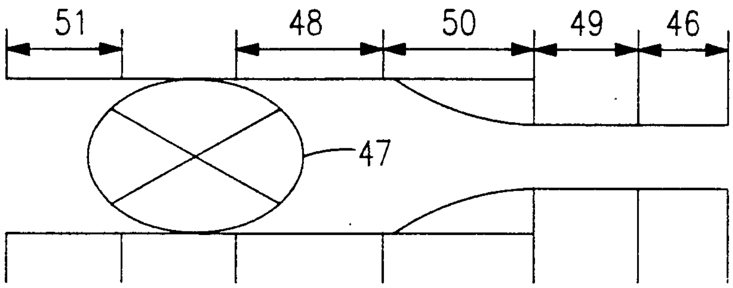

FIG.8

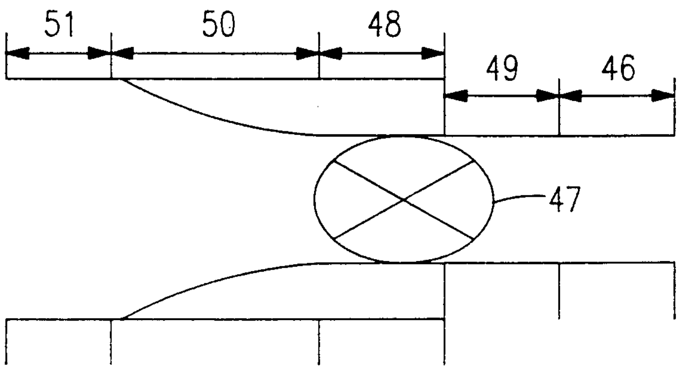

FIG.9

INTERNATIONAL SEARCH REPORT

International application No.
PCT/GB 94/00360

A. CLASSIFICATION OF SUBJECT MATTER
IPC 5 F25B41/06

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 5 F25B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US,A,4 459 819 (HARGRAVES) 17 July 1984 see column 4, line 37 - column 14, line 4; figures 1-9	1-3,5,7, 14,15,17
A	US,A,2 095 834 (RODMAN) 12 October 1937 see page 2, left column, line 73 - page 3, right column, line 41; figures 1-3	1-3,5, 14,15,17
A	US,A,3 023 591 (TILNEY) 6 March 1962 see column 2, line 13 - column 6, line 48; figures 1-5	1,2,5, 13-15,17

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

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"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

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Date of the actual completion of the international search	Date of mailing of the international search report
8 June 1994	16.06.94

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+ 31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+ 31-70) 340-3016	Boets, A

Form PCT/ISA/210 (second sheet) (July 1992)

INTERNATIONAL SEARCH REPORT

International application No.
PCT/GB 94/00360

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US,A,3 531 947 (DRURY) 6 October 1970 see column 2, line 19 - column 4, line 22; figures 1-3 ---	1,2,5, 14,15,17
A	DE,U,92 17 405 (KM-SCHMÖLE) 18 February 1993 see page 6, paragraph 6 - page 8, last paragraph; figures 1-6 ---	1,2,14, 15,17
A	DE,C,307 359 (LANGER) 17 August 1918 ---	
A	DE,C,293 379 (GERLACH) 2 August 1916 ---	
A	FR,A,2 222 616 (BRUCH) 18 October 1974 -----	

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No.
PCT/GB 94/00360

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US-A-4459819	17-07-84	NONE	
US-A-2095834		NONE	
US-A-3023591		NONE	
US-A-3531947	06-10-70	NONE	
DE-U-9217405	18-02-93	EP-A- 0563718	06-10-93
DE-C-307359		NONE	
DE-C-293379		NONE	
FR-A-2222616	18-10-74	NONE	