Jan. 17, 1967 R. C. SCHLICHTIG 3,298,196

DYNAMIC PUMP TYPE REFRIGERATION SYSTEM

Filed May 3, 1965 2 Sheets-Sheet 1

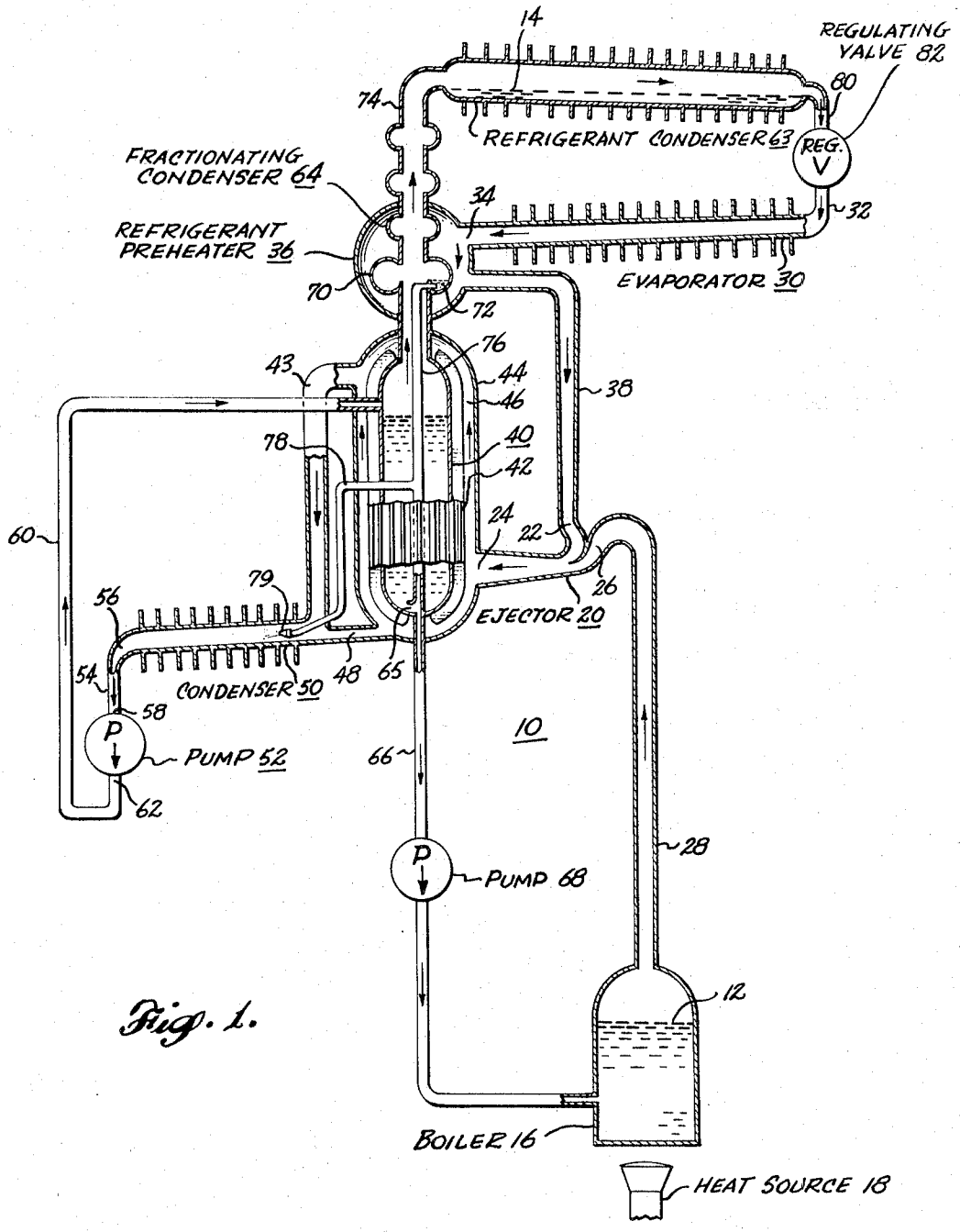


Fig. 1.

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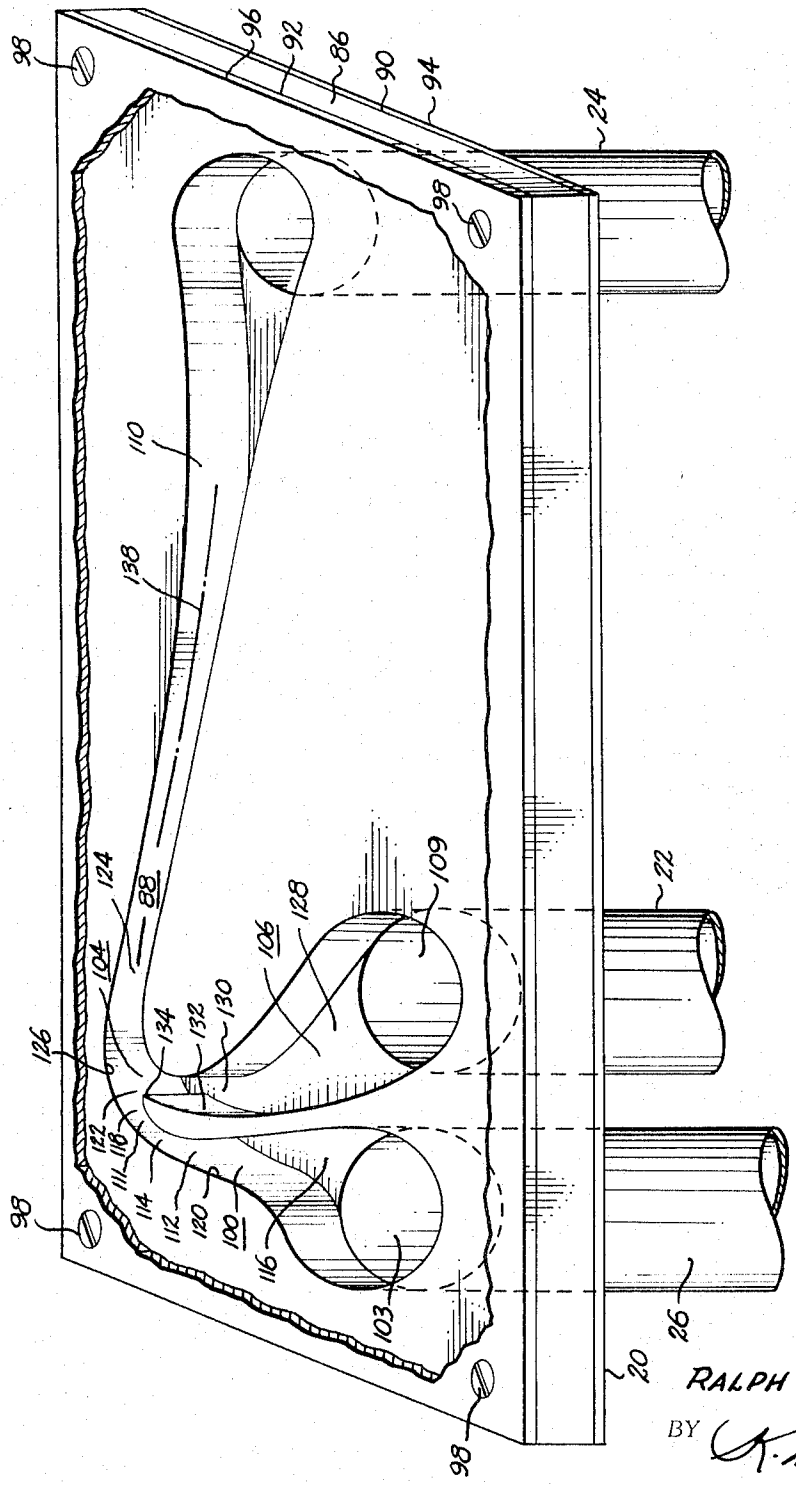


Fig. 2.

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United States Patent Office 3,298,196 Patented Jan. 17, 1967

3,298,196

DYNAMIC PUMP TYPE REFRIGERATION SYSTEM

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Filed May 3, 1965, Ser. No. 452,648
7 Claims. (Cl. 62—500)

This invention relates to heat transfer apparatus, and more particularly to a dynamic pump type refrigeration system employing heat as a source of power and which is suitable for either air conditioning or refrigeration.

This is a continuation-in-part application of United States patent application Serial Number 253,571, filed by applicant on January 24, 1963, and entitled "Ejector Type Refrigeration System," now Patent No. 3,199,310.

The refrigeration system shown and described in the present application has incorporated therein many of the features of the refrigeration system shown in FIG. 1 of the aforementioned patent application Serial Number 253,571, however, in addition thereto it has several added features such as those which greatly increase the coefficient of performance of the refrigeration system. For instance, in accordance with the invention of the present application the back pressure on the output of the dynamic pump, which preferably is a centrifugal ejector, is substantially reduced by effectively condensing a mixture of refrigerant and power fluid, which are mutually soluble, discharge from the output of the dynamic pump. By effectively condensing such mixture of refrigerant and power fluid the back pressure on the dynamic pump is lowered to thereby further increase the coefficient of performance of the refrigeration system.

In the refrigeration system shown in FIG. 1 of the aforementioned patent application Serial Number 253,571 the superheat of the mixture of vaporized power fluid and refrigerant vapor discharge from the ejector is transferred to the liquid within an auxiliary boiler to thus evaporate refrigerant dissolved in the power fluid disposed within the auxiliary boiler. Thus a large portion of the excess energy of the power fluid discharged from the ejector is recovered. In accordance with the present invention not only is the superheat of the mixture of vaporized power fluid and refrigerant vapor effectively utilized, but in addition under given operating conditions the heat of vaporation of part of the power fluid discharged from the ejector is also utilized in evaporating refrigerant dissolved in the power fluid disposed within the auxiliary boiler. Thus, the coefficient of performance of the refrigeration system is further increased.

An object of this invention is to provide for increasing the coefficient of performance of a dynamic pump type refrigeration system.

Another object of this invention is to provide for reducing the back pressure on the dynamic pump of a binary fluid refrigeration system by condensing a mixture of mutually soluble fluids, namely the refrigerant and the higher boiling point power fluid, discharged from the dynamic pump, to thereby increase the weight flow ratio of refrigerant to power fluid delivered by the dynamic pump and thus increase the coefficient of performance of the refrigeration system.

A further object of this invention is to provide a thermally powered dynamic pump type refrigeration system which utilizes mutually soluble binary fluids and which has a high coefficient of performance and which can effectively operate from a low boiler supply pressure by condensing the binary fluids discharged from the dynamic pump so as to reduce the back pressure on the dynamic pump and by employing a centrifugal type ejector as the dynamic pump, the characteristics of which are such that it will operate with high efficiency even with a small pressure difference between the pressure at the discharge of the ejector and the pressure at the primary of the ejector and thus at the boiler supply.

Another object of this invention is to provide a dynamic pump type refrigeration system which achieves a lower pressure in the evaporator and thus a lower temperature at the evaporator.

A further object of this invention is to provide a dynamic pump type binary fluid refrigeration system which utilizes a refrigerant and a power fluid whose vapor pressures are above atmospheric pressure so as to prevent air contamination of the refrigeration system.

Still another object of this invention is to provide for increasing the coefficient of performance of a dynamic pump type, binary fluid, refrigeration system by utilizing a suitable dynamic pump and a power fluid having a higher molecular weight than the refrigerant so that, according to the law of conservation of angular momentum and the Coriolis effect, on partial mixing of the vapors of the power fluid and the refrigerant in the dynamic pump, the free energy represented by the separated vapors is recovered as kinetic energy on partial mixing of the vapors, to thus increase the efficiency of the dynamic pump.

A further object of this invention is to provide a thermally powered ejector type refrigeration system which has a high coefficient of performance over a relatively wide range of boiler pressure by properly incorporating in the refrigeration system a centrifugal type ejector so that centrifugal pressure builds up in the power vapor within the accelerator compartment of the centrifugal type ejector to thereby render the ejector less sensitive to changes in the ejector primary pressure and thus less sensitive to changes in the boiler pressure.

Still another object of this invention is to provide a thermally powered refrigeration system which has incorporated therein at least one dynamic pump and in which the superheat of the mixture of vaporized power fluid and refrigerant discharged from the ejector and the heat of vaporization of part of such vaporized power fluid is effectively utilized to increase the coefficient of performance of the refrigeration system by disposing an auxiliary boiler in heat exchange relationship with such mixture so that such superheat and heat of vaporization can be effectively utilized in evaporating the refrigerant dissolved in the power fluid disposed in the auxiliary boiler to thereby separate the liquid refrigerant from the liquid power fluid.

A further object of this invention is to provide a thermally powered binary fluid refrigeration system which utilizes a power fluid having a higher molecular weight than the refrigerant so as to minimize the loss of kinetic energy on mixing the vapors of the power fluid and refrigerant in a dynamic pump by retaining the greater part of the momentum in the power fluid.

Another object of this invention is to provide for minimizing the back pressure on the dynamic pump of a binary fluid refrigeration system by utilizing a refrigerant such as a fluorocarbon which has a slow rise of pressure versus temeprature to thus maintain a high coefficient of performance for the refrigeration system.

Other objects of this invention will become apparent from the following description when taken in conjunction with the accompanying drawings in which:

FIG. 1 is a flow diagram of a dynamic pump type refrigeration system illustrating the teachings of this invention; and

FIG. 2 is a detailed and partly cut away illustration of the ejector shown in FIG. 1.

Referring to FIG. 1 there is shown a two fluid dynamic pump type refrigeration system 10 illustrating the teachings of this invention and in which the power fluid 12 has a greater molecular weight and boiling point than the refrigerant fluid 14 and the two fluids 12 and 14 are separated by condensation and distillation. The re-

frigeration system **10** is such that it can employ a power liquid and a refrigerant liquid that are mutually miscible to various degrees of completeness.

It is preferable that the refrigerant liquid **14** has the following characteristics: low molecular weight, a boiling point below room temperature, nontoxic, nonflammable, slow rise of pressure versus temperature, chemically stable, low freezing point, low specific heat, low cost, easily separable by distillation, and low viscosity in either the vapor or liquid state. On the other hand, the power liquid **12** should preferably have the following characteristics: high molecular weight, a boiling point higher than that for the refrigerant, nontoxic, nonflammable, a high critical temperature, chemically stable, low freezing point, low specific heat, low cost, and low viscosity in both the vapor and liquid state. The advantages to be obtained when the power fluid **12** and refrigerant fluid **14** have many of the aforementioned characteristics will be set forth hereinafter.

The following is a partial list of suitable combinations of refrigerant and power fluid that can be utilized as the binary fluids in the refrigeration system shown in FIG. 1.

Power Fluid	Refrigerant
1. $CBrF_2$-$CBrF_2$	$CHCl_2F$.
Molecular Weight, 259.9	Molecular Weight, 102.9.
Boiling Point, 117.5° F.	Boiling Point, 48.1° F.
2. CCl_2F-$CClF_2$	$CHCl_2F$.
Molecular Weight, 187.4	Molecular Weight, 102.9.
Boiling Point, 117.6° F.	Boiling Point, 48.1° F.
3. $CClF_2$-$CClF_2$	CCl_2F_2.
Molecular Weight, 170.9	Molecular Weight, 120.9.
Boiling Point, 38.4° F.	Boiling Point, −21.6° F.

The refrigeration system **10** includes a high temperature or main boiler **16** for vaporizing the power fluid **12** which is disposed within the boiler **16** in a liquid state. In practice, the main boiler **16** can be heated from any suitable source **18**. The refrigeration system **10** also includes a dynamic pump, preferably a centrifugal type ejector such as the ejector **20**, which will be described in greater detail hereinafter. The ejector **20** is similar to the ejector shown in FIG. 4 of United States patent application Serial No. 234,816, filed November 1, 1962, by the applicant herein, and entitled "Ejectors," now Patent No. 3,215,088.

In general, the ejector **20** comprises a refrigerant inlet **22**, a discharge portion **24**, and a power inlet **26** for receiving the vaporized power fluid from the main boiler **16** through a conduit **28**. A conventional finned evaporator **30**, having an inlet **32** and an outlet **34**, is provided for evaporating the refrigerant fluid **14** which enters the evaporator **30** in a liquid state. In order that the ejector **20** can effectively pump the refrigerant vapor from the evaporator **30** the outlet **34**, of the evaporator **30**, is in communication with the refrigerant inlet **22**, of the ejector **20**, through a refrigerant preheater **36** and a conduit **38**. The function of the refrigerant preheater will be described hereinafter.

Since the vaporized power fluid flows into the power inlet **26**, of the ejector **20**, and the refrigerant vapor flows into the refrigerant inlet **22**, of the ejector **20**, a mixture of refrigerant vapor and vaporized power fluid is discharged from the discharge portion **24** of the ejector **20**. A cylindrical shaped auxiliary boiler **40** with heat exchange fins and having an outer heat exchange surface **42** is so disposed that the mixture of refrigerant vapor and vaporized power fluid discharged from the discharge portion **24**, of the ejector **20**, flows over the heat exchange surface **42**. In order to direct the mixture of vaporized power fluid and refrigerant vapor upward and over the heat exchange surface **42**, and out through an outlet conduit **43** which is connected to a conventional condenser **50** which is cooled to near ambient temperature, a cylindrical shaped jacket **44** is disposed around the auxiliary boiler **40** so as to form a chamber **46** between the jacket **44** and the auxiliary boiler **40**. A liquid by-pass conduit **48** is interconnected between the jacket **46** and the condenser **50** to conduct any condensed power fluid from the chamber **46** to the condenser **50** as will be explained more fully hereinafter.

The condenser **50** is in communication with the chamber **46** through the conduit **43** so as to receive and effectively condense a mixture of mutually soluble refrigerant and power fluid from the chamber **46**, thereby reducing the back pressure on the discharge portion **24** of the ejector **20**, to thus increase the weight flow ratio of refrigerant to power fluid delivered by the ejector **20** and thereby increase the coefficient of performance of the refrigeration system **10**.

For the purpose of pumping the mixture of condensed refrigerant and condensed power fluid from the condenser **50** into the auxiliary boiler **40** at a pressure higher than the pressure in the condenser **50**, a liquid pump **52** is provided. In particular, a conduit **54** is interconnected between the outlet **56**, of the condenser **50**, and the inlet **58** of the liquid pump **52**, and a conduit **60** is interconnected between the outlet **62**, of the liquid pump **52**, and the auxiliary boiler **40**. In the embodiment shown, the pressure within the upper portion of the auxiliary boiler **40**, where the mixture of refrigerant and power fluid is in a vapor state, must be high enough so as to effectively condense the refrigerant vapor in a conventional refrigerant condenser **63** which is cooled to near ambient temperature.

Due to heat exchange from the heat exchange surface **42** most of the refrigerant dissolved in the power fluid in the auxiliary boiler **40** is reevaporated and flows upward into a heat exchanger **64**, specifically a fractionating condenser, carrying with it a small fraction of power fluid vapor, thus leaving a residue of condensed power fluid in the auxiliary boiler **40**. A small portion of the residue of power fluid in the lower portion of the auxiliary boiler **40** is evaporated and flows upward, however, it is condensed on the inside surface of the auxiliary boiler **40** before it reaches the liquid level in the auxiliary boiler **40**. On condensing such power fluid gives up heat which causes evaporation of refrigerant which is in close proximity to such condensing action. Of course, such condensing on the inside surface of the auxiliary boiler **40** depends upon the temperature at the heat exchange surface **42**.

The residue of liquor power fluid in the auxiliary boiler **40** which is not evaporated so as to flow to the fractionating condenser **64** is returned to the main boiler **16** through an opening **65** in the bottom of the auxiliary boiler **40** and through a conduit **66**. Depending upon the relative elevation of the auxiliary boiler **40** with respect to the main boiler **16** a liquid pump **68** may or may not be required to pump the condensed power fluid from the auxiliary boiler **40** to the main boiler **16**.

The fractionating condenser **64** is provided with separating surfaces **70** for condensing the evaporated power fluid received from the auxiliary boiler **40**. In order to collect such condensed power fluid a collector **72** is provided. The refrigerant vapor in a substantially pure form flows from the fractionating condenser **64** through a conduit **74** and into the refrigerant condenser **63** where it is effectively condensed. However, as will be explained more fully hereinafter by a proper choice of refrigerant and power fluid it is advantageous that a small fraction of power fluid be allowed to be carried along with the refrigerant vapor and into the refrigerant condenser **63**. In order to return the greater part of condensed power fluid and any small fraction of dissolved refrigerant from the collector **72** to the main boiler **16**, a conduit **76** is provided. Here again the liquid pump **68** may or may not be necessary depending upon the relative elevation of the collector **72** with respect to the main boiler **16** and the relative elevation of the bottom of the auxiliary boiler **40** with respect to the main boiler **16**.

In order to further reduce the back pressure on the discharge portion 24, of the ejector 20, a small part of the condensed power fluid and any small fraction of dissolved refrigerant flowing from the collector 72 and through the conduit 70 is bled off from the conduit 70 through a conduit 78 and finally into the condenser 50 where it is sprayed onto the inside surface of the condenser 50 by a spray nozzle 79. In other words, since the power fluid has a higher boiling point than the refrigerant the greater the portion of power fluid in the condenser 50 the lower will be the pressure in the condenser 50 and thus the lower the back pressure on the discharge portion 24 of the ejector 20.

As hereinbefore mentioned the refrigerant vapor flows from the evaporator 30 through the refrigerant preheater 36. In so doing the refrigerant vapor flows around the fractionating condenser 64 and through heat exchange obtains heat from the fractionating condenser 64 to thereby preheat the refrigerant vapor before it enters the refrigerant inlet 22 of the ejector 20.

A conduit 80 is interconnected between the refrigerant condenser 63 and the evaporator 30 for returning the condensed refrigerant to the evaporator 30. For the purpose of regulating the flow of condensed refrigerant from the refrigerant condenser 63 to the evaporator 30, a conventional thermal regulating valve 82 is disposed in the conduit 80.

The ejector 20, which is similar to the ejector of FIG. 4 of the aforementioned patent application Serial No. 234,816, will now be described. In general, the ejector 20 includes a median plate 86 having an opening 88 of predetermined shape which extends from a face 90, of the median plate 86, through to the opposite face 92 of the median plate 86; a cover plate 94 disposed against the face 90 of the median plate 86; a cover plate 96 all the surfaces of which are continuous except for a plurality of openings for a plurality of screws 98, the cover plate 96 being disposed against the face 92 of the median plate 86; and the plurality of screws 98 for maintaining the cover plates 94 and 96 in fixed relationship with respect to the median plate 86, so that the combination of the median plate 86 and the cover plates 94 and 96 define a system of interconnected passageways including a curved converging-diverging power stream passageway 100 for receiving power vapor through the power inlet connection 26 having an opening 103, a curved accelerator compartment 104, a curved converging secondary passageway 106 for receiving refrigerant vapor through the refrigerant inlet 22 having an opening 109, and a discharge passageway 110. The power inlet 26 is disposed in an opening in the cover plate 94 and is suitably secured to the cover plate 94. The refrigerant inlet 22 is disposed in another opening in the cover plate 94 and is suitably secured to the cover plate 94.

The curved converging-diverging power stream passageway 100 has a diverging discharge end 111 and a converging receiving end 112 for receiving the power vapor. Thus, the converging-diverging power stream passageway 100 has a restricted throat 114 and is so shaped as to accelerate the power vapor within the curved converging-diverging power stream passageway 100 and create a kinetic energy gradient in the accelerated power vapor in direction transverse to the flow direction of the accelerated power vapor and a pressure gradient in the accelerated power vapor in a direction transverse to the flow direction of the accelerated power vapor, so that a portion of the accelerated power vapor has the greatest total unit pressure energy and unit kinetic energy. The power stream passageway 100 is curved in direction from its receiving end 116 to its discharge end 118 and is bounded in part by a curved outer boundary surface 120 which curves in direction from the receiving end 116, of the power stream passageway 100, to the discharge end 118, whereby centrifugal force is set up within the accelerated power vapor within the power

stream passageway 100 to thereby effect a kinetic energy gradient in the accelerated power vapor in direction transverse to the flow direction of the accelerated power vapor and toward the curved outer boundary surface 120 and a pressure gradient in the accelerated power vapor in direction transverse to the flow direction of the accelerated power vapor and toward the curved outer boundary surface 120, so that the portion of the accelerated power vapor flowing along the curved outer boundary surface 120 of the power stream passageway 100 has the greatest total unit pressure energy and unit kinetic energy.

The curved accelerator compartment 104 has a receiving end 122, which is in communication with the discharge end 118 of the power stream passageway 100, and a discharge end 124. The curved accelerator compartment 104 is curved in direction from its receiving end 122 to its discharge end 124 and is bounded in part by a curved outer boundary surface 126 which likewise curves in direction from the receiving end 122, of the curved accelerator compartment 104, to the discharge end 124, the curved outer boundary surface 126 forming a continuous curve with the curved outer boundary surface 120 of the converging-diverging power stream passageway 100. The curvature of the curved outer boundary surface 126 has the same algebraic sign as the curvature of the curved outer boundary surface 120. The curved converging-diverging power stream passageway 100 is so positioned as to direct the accelerated power vapor through the receiving end 122, of the curved accelerator compartment 104, in such a way that the portion of the accelerated power vapor of greatest total unit pressure energy and unit kinetic energy flows continuously along the curved outer boundary surface 126 of the curved accelerator compartment 104, whereby centrifugal force maintains within the accelerated power vapor within the curved accelerator compartment 104 a pressure gradient and a kinetic energy gradient in direction toward the curved outer boundary surface 126 such that the total unit pressure energy and unit kinetic energy of the accelerated power vapor is greatest along the curved outer boundary surface 126. Thus the region of lowest pressure within the accelerated power vapor within the curved accelerator compartment 104 is at a region which is farthest away from the curved outer boundary surface 126.

The curved converging secondary passageway 106 has a receiving end 128 in communication with the refrigerant inlet 22 and a discharge end 130 in communication with the receiving end 122 of the curved accelerator compartment 104. In practice the curved converging secondary passageway 106 is so shaped as to accelerate the refrigerant vapor within the curved converging secondary passageway 106 and create a kinetic energy gradient in the accelerated refrigerant vapor in direction transverse to the flow direction of the accelerated refrigerant vapor and create a pressure gradient in the accelerated refrigerant vapor in direction transverse to the flow direction of the accelerated refrigerant vapor, so that a portion of the accelerated refrigerant vapor has the greatest total unit pressure energy and unit kinetic energy. In particular, in order to accelerate the refrigerant vapor within the secondary passageway 106, the secondary passageway 106 converges from its receiving end 128 to its discharge end 130. The secondary passageway 106 curves in direction from its receiving end 128 to its discharge end 130 and is bounded in part by a curved boundary surface 132 which likewise curves in direction from the receiving end 128 to a point of confluence 134 between the discharge end 130, of the secondary passageway 106, and the discharge end 118, of the converging-diverging power stream passageway 100, so as to create a kinetic energy gradient in the accelerated refrigerant vapor in direction transverse to the flow direction of the accelerated refrigerant vapor and toward the curved boundary sur-

face 132 and create a pressure gradient in the accelerated refrigerant vapor in direction transverse to the flow direction of the accelerated refrigerant vapor and toward the curved boundary surface 132, so that the portion of the accelerated refrigerant vapor flowing along the curved boundary surface 132 of the curved converging secondary passageway 106 has the greatest total unit pressure energy and unit kinetic energy. As shown, the curvature of the curved outer boundary surface 132 of the curved converging secondary passageway 106 has the same algebraic sign as the curvature of the curved outer boundary surface 120 of the converging-diverging power stream passageway 100. The curved converging secondary passageway 106 is so positioned as to direct the accelerated refrigerant vapor into the curved accelerator compartment 104 in such a way that the portion of the accelerated refrigerant vapor of greatest total unit pressure energy and unit kinetic energy flows in contact with the accelerated power vapor within the curved accelerator compartment 104, whereby the accelerated refrigerant vapor is further accelerated.

Referring in particular to the discharge passageway 110, the discharge passageway 110 is in communication with the discharge end 124, of the curved accelerator compartment 104, for receiving from the curved accelerator compartment 104 the accelerated power vapor and the accelerated refrigerant vapor.

In the passageway 110 the power vapor and refrigerant vapor is decelerated so that the kinetic energy of the mixture of power vapor and refrigerant vapor in the discharge passageway 110 is converted to pressure energy. From the discharge passageway 110 the mixture of power vapor and refrigerant vapor flows into the discharge portion 24.

The longitudinal axis 138, of the discharge passageway 110, is curved and its curvature goes through zero. Since the curvature of the longitudinal axis 138 goes through zero the flow path of the mixed power vapor and refrigerant vapor is changed so that the total unit pressure energy and unit kinetic energy of the mixed power vapor and refrigerant vapor within the discharge passageway 110 approaches a uniform value at all points in direction transverse to the flow direction of the mixed power vapor and refrigerant vapor.

The operation of the refrigeration system 10 will now be described. Sufficient heat is applied to the main boiler 16 by means of the heat source 18 to effect a continuous vaporization of the power fluid 12 disposed within the main boiler 16, to thereby produce a power vapor at the system's highest vapor pressure, P_h. This pressure must be sufficiently high so as to effectively operate the ejector 20. The vaporized power fluid is directed to the power inlet 26, of the ejector 20, by means of the conduit 28. The ejector 20, acting as a pump, effects a greatly reduced pressure in the conduit 38 and in the refrigerant preheater 36 so that refrigerant vapor is withdrawn from the evaporator 30 at a pressure P_1 which is the saturation pressure of the refrigerant fluid 14 at the temperature prevailing within the evaporator 30. Evaporation at this reduced pressure P_1 keeps the evaporator 30 cold. Referring to the ejector 20 as shown in FIG. 2 the vaporized power fluid received from the main boiler 16 through the power inlet 26 enters the receiving end 116, of the power stream passageway 100, at a pressure considerably higher than the pressure existing at the discharge portion 24 of the ejector 20. The power vapor is accelerated to a high velocity as it flows through the curved power stream passageway 100 where centrifugal force creates a pressure gradient and a kinetic energy gradient in the accelerated power vapor in direction transverse to the flow direction of the accelerated power vapor so that the portion of the accelerated power vapor closest to the curved outer boundary surface 120 has the greatest total unit pressure energy and unit kinetic energy. On entering the curved accelerator compartment 104 the portion of the power vapor of greatest total unit pressure energy and unit kinetic energy flows continuously along the curved outer boundary surface 126, of the curved accelerator compartment 104, where centrifugal force maintains a pressure gradient in the accelerated power vapor within the curved accelerator compartment 104 in direction toward the curved outer boundary surface 126, of the curved accelerator compartment 104, with the greatest pressure at the curved outer boundary surface 126. A kinetic energy gradient is produced in the accelerated power vapor within the curved accelerator compartment 104 in direction toward the curved outer boundary surface 126 of the curved accelerator compartment 104, with the greatest kinetic energy at the curved outer boundary surface 126. Refrigerant vapor flows from the outlet 34, of the evaporator 30, through the refrigerant preheater 36 where it comes in heat exchange relationship with the fractionating condenser 64, to thus preheat the refrigerant vapor, and from there the preheated refrigerant vapor flows through the conduit 38 to the refrigerant inlet 22 and through the secondary passageway 106 to the receiving end 122 of the curved accelerator compartment 104. The accelerated refrigerant vapor is directed by the secondary passageway 106 into the curved accelerator compartment 104 in such a way that the portion of the accelerated refrigerant vapor of greatest unit pressure energy and unit kinetic energy flows into contact with that portion of the accelerated power vapor within the curved accelerator compartment 104 of least total unit pressure energy and unit kinetic energy, where the refrigerant vapor is accelerated by contact with the power vapor and is diffused into the power vapor.

Since the power vapor flows along the curved outer boundary surface 126, of the curved accelerator compartment 104, with the refrigerant vapor adjacent the power vapor, the refrigerant vapor is diffused into the power vapor thereby forcing the higher molecular weight power vapor to expand radially inward with respect to the curved outer boundary surface 126 in accordance with the Coriolis effect to thereby increase the kinetic energy of the diffused mixture of refrigerant vapor and power vapor. In other words, the free energy represented by the separated vapors of refrigerant and power vapor is recovered as kinetic energy to thus increase the efficiency of the ejector 20 and thereby increase the coefficient of performance of the refrigeration system 10.

Since centrifugal pressure builds up in the power vapor within the accelerator compartment 104 the ejector 20 is less sensitive to changes in the primary pressure at the power inlet 26 and thus less sensitive to changes in the pressure P_h at the main boiler 16. Therefore, normally it is not necessary to provide means for regulating the pressure at the power inlet 26 of the ejector 20.

On leaving the accelerator compartment 104 the power vapor and refrigerant vapor enter the discharge passageway 110 where the vapors are decelerated so that the kinetic energy of the vapors flowing in the discharge passageway 110 is converted to pressure energy. The mixture of refrigerant vapor and vaporized power fluid is then discharged from the discharge portion 24 of the ejector 20 at a pressure P_2 which is sufficient to effectively condense the refrigerant and power fluid in the condenser 50 at near ambient temperatures.

The mixture of refrigerant vapor and vaporized power fluid discharged from the discharge portion 24, of the ejector 20, flows upward through the chamber 46 and over the heat exchange surface 42, of the auxiliary boiler 40, where a small portion of such vaporized power fluid is condensed in the chamber 46 provided the pressure difference between the pressure within the auxiliary boiler 40 and the pressure within the chamber 46 isn't too great. This small portion of condensed power fluid then flows by gravity through the conduit 48 into the condenser 50. In practice, the condensed power fluid in the conduit 48 prevents the flow of refrigerant vapor and vaporized power

fluid from the chamber 46 through the conduit 48 and into the condenser 50. Substantially no refrigerant vapor is condensed in the chamber 46. The refrigerant vapor and the uncondensed portion of the vaporized power fluid flows as a mixture from the chamber 46 through the conduit 43 and into the condenser 50 where such mixture of mutually soluble refrigerant and power fluid is effectively condensed. By so condensing the mixture of mutually soluble refrigerant and power fluid in the condenser 50, the back pressure on the discharge portion 24, of the ejector 20 is reduced, to thus increase the flow ratio of refrigerant to power fluid delivered by the ejector 20, and thereby increase the coefficient of performance of the refrigeration system 10. Since the ejector 20 can operate with high efficiency even with a small pressure difference between the pressure at the discharge portion 24 and the pressure at the power inlet 26 and since the back pressure on the ejector has been reduced as hereinbefore described the refrigeration system 10 can effectively operate from a low boiler supply pressure P_h.

Since the refrigeration system 10 utilizes a refrigerant and a power fluid whose vapor pressures are above atmospheric pressure there is substantially no air contamination of the refrigeration system 10. Also, the refrigerant used in the refrigeration system 10 is a fluorocarbon which has a slow rise of pressure versus temperature which minimizes the back pressure on the discharge portion 24, of the ejector 20, to thus maintain a high coefficient of performance for the refrigeration system 10.

After condensing the mixture of refrigerant and power fluid in the condenser 50 the liquid pump 52 pumps the mixture including condensed refrigerant dissolved in condensed power fluid through the conduit 60 into the auxiliary boiler 40 at a higher pressure than the pressure within the condenser 50. The pressure, P_3, within the auxiliary boiler 40 is at a higher pressure than the pressure in the condenser 50 and also at a higher pressure than P_2 at the discharge portion 24, of the ejector 20. The pressure P_3 must be sufficiently high so as to effectively condense the refrigerant vapor within the refrigerant condenser 63. Since a mixture of refrigerant vapor and vaporized power fluid discharged from the ejector 20 flows over the heat exchange surface 42, of the auxiliary boiler 40, heat is transferred to the condensed power fluid and refrigerant in the auxiliary boiler 40 to thereby reevaporate the condensed refrigerant which flows to the fractionating condenser 64 carrying with it a small fraction of power fluid, to thus separate the condensed refrigerant from the condensed power fluid and thereby leave a residue of condensed power fluid in the auxiliary boiler 40. A small portion of the residue of condensed power fluid which evaporates within the auxiliary boiler 40 but does not flow to the fractionating condenser 64, but rather flows upward through the liquid in the auxiliary boiler 40 into the upper portion of the liquid in the auxiliary boiler 40, condenses on the inner surface of the auxiliary boiler 40 and in so doing gives up heat which effects an evaporation of some of the refrigerant in the auxiliary boiler 40 which refrigerant vapor then flows to the fractionating condenser 64. In operation, on entering the auxiliary boiler 40 a mixture of condensed refrigerant and power fluid discharged from the conduit 60 floats on top of the liquid already in the auxiliary boiler 40 and gradually works its way downward in the auxiliary boiler 40 as more condensed refrigerant and condensed power fluid enters the auxiliary boiler 40 and as the condensed refrigerant underneath such mixture is being reevaporated.

By flowing the mixture of refrigerant vapor and vaporized power fluid, discharged from the ejector 20, over the heat exchange surface 42, of the auxiliary boiler 40, the coefficient of performance of the refrigeration system 10 is increased, since not only is the superheat carried by the mixture of vaporized power fluid and refrigerant vapor discharged from the discharge portion 24, of the ejector 20, effectively utilized, but also the heat of vaporization of part of such vaporized power fluid is effectively utilized.

The residue of condensed power fluid in the auxiliary boiler 40 is returned to the main boiler 16 by way of the conduit 66. On the other hand, the evaporated refrigerant and any small fraction of entrained power fluid flowing into the fractionating condenser 64 flows into contact with the separating surfaces 70 and in so doing the power fluid is condensed and flows into the collector 72. The evaporated refrigerant flows through the fractionating condenser 64 and the conduit 74 with the desired degree of purity and into the refrigerant condenser 63 where it is effectively condensed. If any of the evaporated refrigerant on flowing into contact with separating surfaces 70, of the fractionating condenser 64, condenses it will almost immediately be reevaporated due to the heat given off by the latent heat of vaporization of the power fluid when it condenses upon flowing into contact with the separating surfaces 70. The condensed refrigerant in the refrigerant condenser 63 is then returned to the evaporator 30 through the conduit 80 and the regulating valve 82.

In practice it is desirbale to utilize a combination of refrigerant and power fluid such that the mutual solubility is good enough so that nearly complete separation can take place by distillation at ambient temperatures but such that the mutual solubility becomes restricted at the low temperatures existing in the evaporator 30 so that the combined vapor pressure of the refrigerant and power fluid is greater than the vapor pressure would be for the pure refrigerant. This desirable characteristic is a deviation from Rault's law. Thus, since the pressure in the evaporator 30 is greater than it would be with pure refrigerant, with no fraction of power fluid, the load on the ejector 20 is reduced, to thereby increase the coefficient of performance of the refrigeration system 10. Examples of pairs of refrigerant and power fluid having the aforementioned desirable characteristic are pairs 1 and 2 previously listed in column 3.

Most of the condensed power fluid, and any small fraction of refrigerant dissolved in such power fluid, which is disposed in the collector 72 of the fractionating condenser 64 is returned to the main boiler 16 through the conduit 76 and the conduit 66. A portion of the condensed power fluid, and any small fraction of refrigerant dissolved in the power fluid, flowing from the collector 72 and downward through the conduit 76 flows into the conduit 78 where it flows into the condenser 50 by way of the spray nozzle 79, to thereby increase the proportion of power fluid to refrigerant in the condenser 50. Since the power fluid has a higher boiling point than the refrigerant a larger proportion of power fluid decreases the pressure in the condenser 50 and thus the back pressure on the discharge portion 24 of the ejector 20.

The above described cycle of operation is then repeated.

Although a centrifugal type ejector such as the ejector 20 is preferred, it is to be understood that another type of dynamic pump such as a turbine (not shown) could be substituted for the ejector 20 and some of the aforementioned advantages obtained by using the ejector 20 would also be obtained when using a turbine (not shown).

The apparatus embodying the teachings of this invention has several advantages. For instance, apparatus constructed in accordance with the teachings of this invention has a higher coefficient of performance than other thermally powered refrigeration systems incorporating dynamic pumps. The reason for such higher coefficient of performance has heretofore been discussed. In addition, apparatus constructed in accordance with the teachings of this invention is capable of operating from a relatively low boiler pressure. Further, apparatus constructed in accordance with the teachings of this invention utilizes a refrigerant and a power fluid whose vapor

pressures are above atmospheric pressure so that there is no air contamination of the refrigeration system. Further, apparatus constructed in accordance with the teachings of this invention does not require a regulation of the pressure of the vaporized power fluid flowing to the power inlet of the dynamic pump incorporated in the refrigeration sytsem. Also, apparatus constructed in accordance with the teachings of this invention can utilize a refrigerant which has a very low freezing point.

Since certain changes may be made in the above described apparatus and different embodiments of the invention may be made without departing from the spirit and the scope thereof, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

I claim as my invention:

1. In a heat transfer system for using as working mediums a power fluid and a refrigerant, the combination comprising, pump means having power inlet means, refrigerant inlet means and discharge means; means for vaporizing the power fluid and for directing the vaporized power fluid to the power inlet means of said pump means; an evaporator for evaporating the refrigerant to a vapor and for effecting in cooperation with said pump means a flow of the refrigerant vapor to the refrigerant inlet means of said pump means so that a mixture of refrigerant vapor and vaporized power fluid is discharged from the discharge means of said pump means; first condenser means operatively associated with the discharge means of said pump means for receiving at least a part of the mixture of refrigerant vapor and vaporized power fluid discharged from the discharge means of said pump means and for effectively condensing such received mixture to thereby reduce the back pressure on the discharge means of said pump means; second condensing means; means operatively associated with said first condenser means and with said second condensing means for receiving a mixture of condensed power fluid and condensed refrigerant from said first condenser means and for separating condensed refrigerant of such received condensed mixture from condensed power fluid of such received condensed mixture by reevaporating condensed refrigerant of such received condensed mixture thus leaving a residue of condensed power fluid and for directing at least a part of the reevaporated condensed refrigerant to said second condensing means for condensing; and means for returning at least a part of the residue of condensed power fluid to said vaporizing means and for returning condensed refrigerant from said second condensing means to said evaporator.

2. In a heat transfer system for using as working mediums a power fluid and a refrigerant, the combination comprising, first pump means having power inlet means, refrigerant inlet means and discharge means; means for vaporizing the power fluid and for directing the vaporized power fluid to the power inlet means of said first pump means; an evaporator for evaporating the refrigerant to a vapor and for effecting in cooperation with said first pump means a flow of the refrigerant vapor to the refrigerant inlet means of said first pump means so that a mixture of refrigerant vapor and vaporized power fluid is discharged from the discharge means of said first pump means; first condenser means operatively associated with the discharge means of said first pump means for receiving at least a part of the mixture of refrigerant vapor and vaporized power fluid discharged from the discharge means of said first pump means and for effectively condensing such received mixture to thereby reduce the back pressure on the dischrage means of said first pump means; second pump means for pumping a mixture of condensed refrigerant and condensed power fluid from said first condenser means; second condensing means; means operatively associated with said second pump means and with said second condensing means for receiving the pumped mixture of condensed refrigerant and condensed power fluid at a pressure higher than the pressure in said first condenser means and for separating condensed refrigerant of such received pumped mixture from condensed power fluid of such received pumped mixture by reevaporating condensed refrigerant of such received pumped mixture thus leaving a residue of condensed power fluid and for directing at least a a part of the reevaporated condensed refrigerant to said second condensing means for condensing; and means for returning at least a part of the residue of condensed power fluid to said vaporizing means and for returning condensed refrigerant from said second condensing means to said evaporator.

3. In a heat transfer system for using as working mediums a power fluid and a refrigerant, the combination comprising, first pump means having power inlet means, refrigerant inlet means and discharge means; a main boiler for vaporizing the power fluid and for directing the vaporized power fluid to the power inlet means of said first pump means; an evaporator for evaporating the refrigerant to a vapor and for effecting in cooperation with said first pump means a flow of the refrigerant vapor the the refrigerant inlet means of said first pump means so that a mixture of refrigerant vapor and vaporized power fluid is discharged from the discharge means of said first pump means; an auxiliary boiler having a heat exchange surface; guide means for effecting a flow of the mixture of refrigerant vapor and vaporized power fluid discharged from the discharge means of said first pump means over the heat exchange surface of said auxiliary boiler; first condensing means connected to receive a mixture of refrigerant vapor and vaporized power fluid after the latter mixture has flowed over the heat exchange surface of said auxiliary boiler, to thereby effectively condense such received mixture of refrigerant vapor and vaporized power fluid in said first condensing means to thus reduce the back pressure on the discharge means of said first pump means; second pump means for pumping a mixture of condensed refrigerant and condensed power fluid from said first condenser means into said auxiliary boiler at a pressure higher than the pressure in said first condensing means where, in said auxiliary boiler, condensed refrigerant of such condensed mixture is separated from condensed power fluid of such condensed mixture by reevaporating condensed refrigerant of such condensed mixture due to heat exchange from the heat exchange surface of said auxiliary boiler, thus leaving a residue of condensed power fluid in said auxiliary boiler; second condensing means; means for directing at least a part of the reevaporated condensed refrigerant from said auxiliary boiler to said second condensing means for condensing; and means for returning at least a part of the reside of condensed power fluid from said auxiliary boiler to said means boiler and for returning condensed refrigerant from said second condensing means to said evaporator.

4. The heat transfer system in accordance with claim 3 in which means is provided for directing into said first condensing means condensed power fluid which is condensed at the heat exchange surface of said auxiliary boiler due to loss of heat from the mixture of refrigerant vapor and vaporized power fluid which is discharged from the discharge means of said first pump means and which flows over the heat exchange surface of said auxiliary boiler.

5. In a heat transfer system for using as working mediums a power fluid comprising a fluid of a given molecular weight and a given boiling point and a refrigerant comprising a fluid of relatively lower molecular weight and relatively lower boiling point in which the refrigerant fluid is at least partially miscible with the power fluid, the combination comprising, first pump means having power inlet means, refrigerant inlet means and discharge means; a main boiler for vaporizing the power

fluid and for directing the vaporized power fluid to the power inlet means of said first pump means; an evaporator for evaporating the refrigerant to a vapor and for effecting in cooperation with said first pump means a flow of the refrigerant vapor to the refrigerant inlet means of said first pump means so that a mixture of refrigerant vapor and vaporized power fluid is discharged from the discharge means of said first pump means, and said first pump means being so constructed that the refrigerant vapor and the vaporized power fluid flow in a portion of said first pump means in such a way that the vaporized power fluid flows along a curved surface with the refrigerant vapor adjacent the vaporized power fluid so that the refrigerant vapor is diffused into the vaporized power fluid causing the vaporized power fluid to expand radially inward with respect to the curved surface thereby increasing the kinetic energy of the diffused mixture of vaporized power fluid and refrigerant vapor to thereby effect an improved efficiency for said first pump means; an auxiliary boiler having a heat exchange surface; guide means for effecting a flow of the mixture of refrigerant vapor and vaporized power fluid discharged from the discharge means of said first pump means over the heat exchange surface of said auxiliary boiler; first condensing means connected to receive a mixture of refrigerant vapor and vaporized power fluid after the latter mixture has flowed over the heat exchange surface of said auxiliary boiler, to thereby effectively condense such received mixture of refrigerant vapor and vaporized power fluid in said first condensing means to thus reduce the back pressure on the discharge means of said first pump means; second pump means for pumping a mixture of condensed refrigerant and condensed power fluid from said first condenser means into said auxiliary boiler at a pressure higher than the pressure in said first condensing means where, in said auxiliary, boiler condensed refrigerant of such condensed mixture is separated from condensed power fluid of such condensed mixture by evaporating condensed refrigerant of such condensed mixture due to heat exchange from the heat exchange surface of said auxiliary boiler, thus leaving a residue of condensed power fluid in said auxiliary boiler; a heat exchanger connected to said auxiliary boiler for receiving the evaporated refrigerant and the evaporated power fluid from said auxiliary boiler so as to condense at least a portion of the evaporated power fluid within said heat exchanger; second condensing means connected to said heat exchanger for receiving evaporated refrigerant from said heat exchanger so as to condense the evaporated refrigerant; and means for returning at least a part of the residue of condensed power fluid from said auxiliary boiler and at least a part of the condensed power fluid from said heat exchanger to said main boiler and for returning condensed refrigerant from said second condensing means to said evaporator.

6. The heat transfer system in accordance with claim 5 including a refrigerant preheater disposed in heat transfer relationship with said heat exchanger, and conduit means for interconnecting said refrigerant preheater with said evaporator and with the refrigerant inlet means of said first pump means so that refrigerant vapor flows from said evaporator, through said refrigerant preheater and into the refrigerant inlet means of said first pump means.

7. The heat transfer system in accordance with claim 5 including means for effecting a flow of part of the condensed power fluid within said heat exchanger to said first condensing means, to thereby further reduce the back pressure on the discharge means of said first pump means.

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