April 4, 1967 J. H. ANDERSON 3,312,063

HEAT TRANSFER AT EQUALIZED PRESSURE

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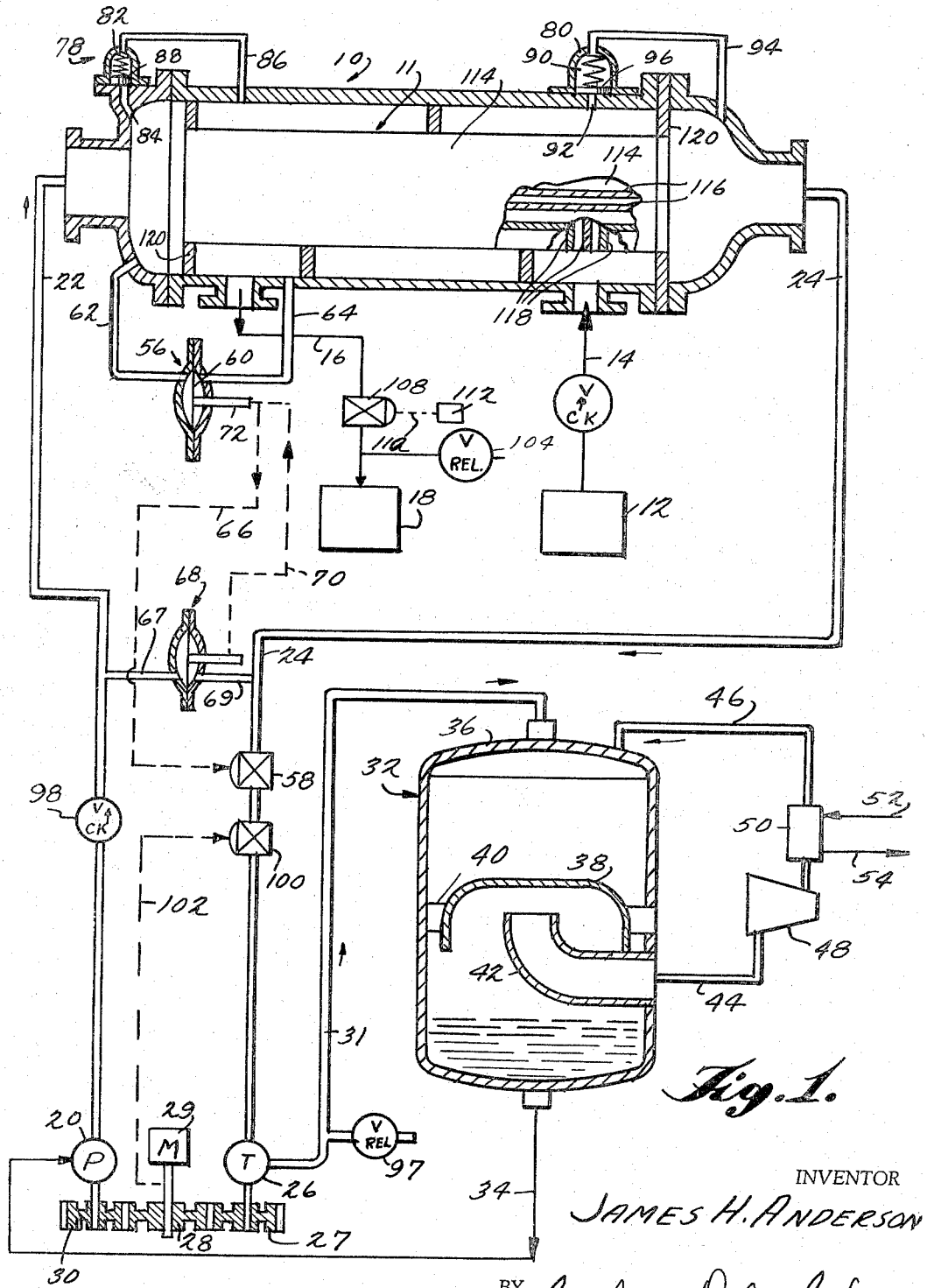


Fig. 1.

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Fig. 2.

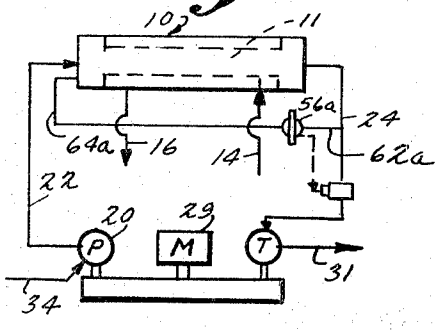


Fig. 4.

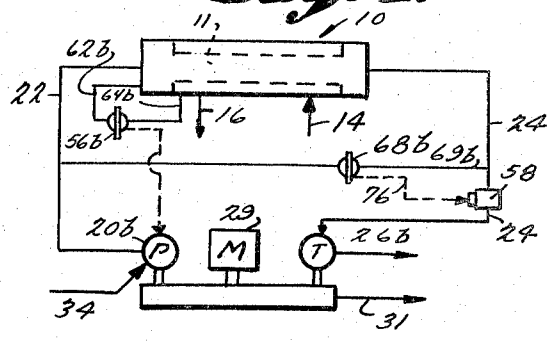


Fig. 3.

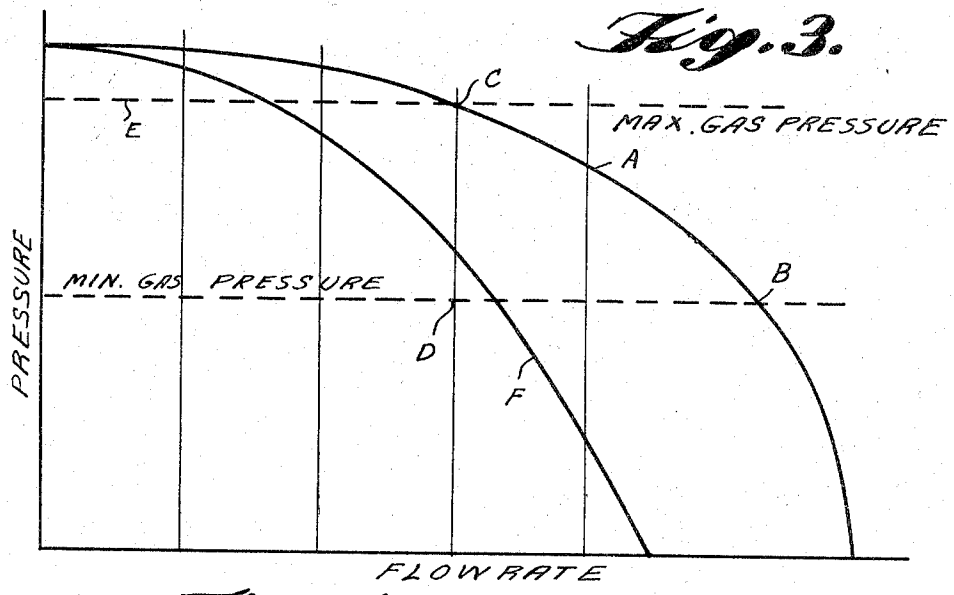
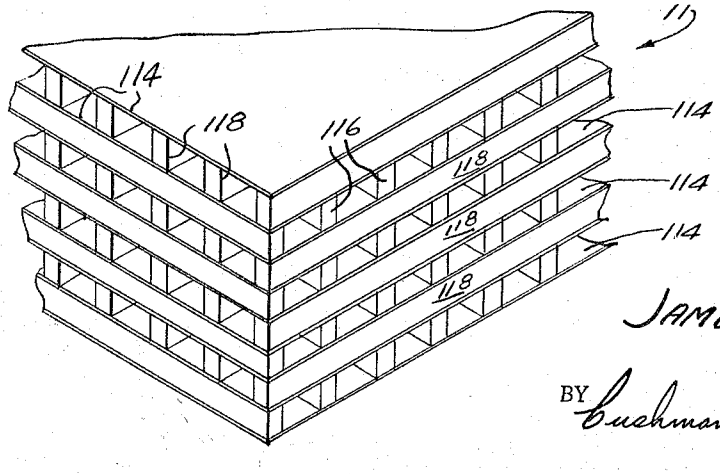


Fig. 5.



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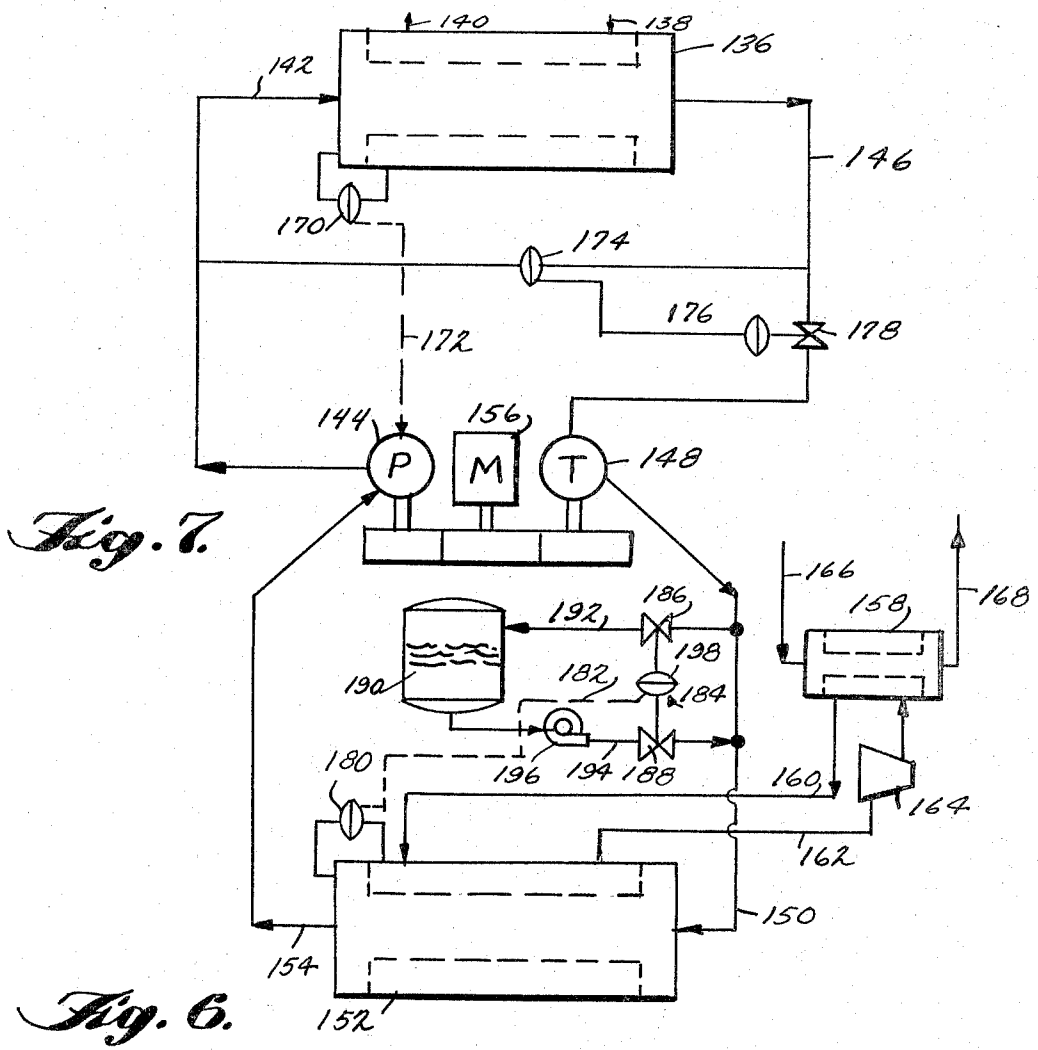
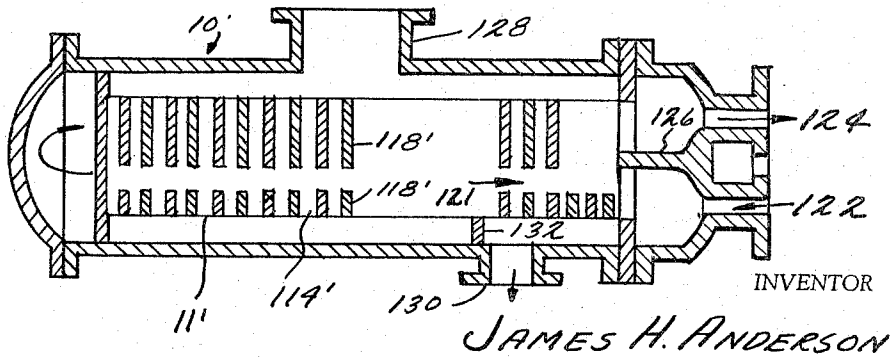


Fig. 7.

Fig. 6.



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United States Patent Office 3,312,063
Patented Apr. 4, 1967

3,312,063

HEAT TRANSFER AT EQUALIZED PRESSURE

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Filed July 22, 1965, Ser. No. 474,017
13 Claims. (Cl. 60—59)

This invention relates to heat transfer systems and in particular to the economic transfer of heat between a stream of high pressure fluid and a stream of fluid at a lower pressure and having a different heat content. The transfer of heat effects a change in the heat content of the fluids as by changing the temperature or the phase of the fluids or both.

In heat transfer systems which employ heat exchangers of the type having internal partition walls which separate fluid streams of different heat contents, the cost and operating difficulties of the system are increased when there is a large pressure differential between the two fluids. The partition walls in a heat exchanger must be constructed with sufficient strength to withstand the presure differential between the fluids, and this requires either high strength, costly materials or thick, heavy partitions. In the latter case the amount of material employed adds considerably to the cost of the equipment and, in addition, the increased wall thickness reduces heat transfer between the fluids. Operation with a pressure differential across the partitions increases the possibility of leaks and the quantity of leakage flow and, accordingly, the construction of the exchanger must take this difficulty into account and assure that all joints are extremely tight.

It is the primary object of the present invention to provide a heat transfer system for fluid streams at different pressures and heat contents which overcomes the disadvantages of costly equipment and operating difficulties usually associated with a pressure differential between fluid streams being passed in heat exchange relationship.

It is a further object of the invention to provide a heat transfer system for fluid streams at different pressures and heat contents which increases the pressure of the low pressure fluid to about that of the high pressure fluild whereby thin-walled low-cost heat exchangers may be employed and in which a substantial amount of the energy required for the pressurizing of the low pressure fluid is recovered for reuse within the system.

It is another object to provide a heat transfer system of the above kind in which the energy required for pressurizing the low pressure fluid is recovered by expanding the same in a turbine which is employed to operate a pump for the pressurizing step.

Broadly, the invention contemplates a pump which raises the pressure of a first fluid stream, available at a low pressure, to the pressure of a second high pressure fluid stream whose heat content is desired to be changed and a heat exchanger effecting the desired change in heat content. It will frequently be desirable to include a power-producing turbine in which the first fluid is expanded after having passed through the heat exchanger. The work from the turbine is employed to help drive the pump, and the expanded first fluid is discarded, recycled, or otherwise utilized depending on the nature of the fluid and the particular system. The second fluid, whose heat content has been changed to a desired value, is then ready for further processing in whatever manner is desired. The principles of the invention may thus be employed to reduce construction and operating costs in any process in which it may be necessary to cool, heat, condense or evaporate a high pressure fluid stream with another fluid, such as water, which is normally available at a relatively low pressure. As already pointed out the primary economic advantages of the invention are that a thin-walled, low-cost heat exchanger may be employed and that a substantial amount of the work of pressurizing the low pressure fluid can be reclaimed.

In the description which follows there are described two systems in which a stream of relatively high pressure natural gas is cooled with a recycled cooling medium which is brought up to the pressure of the natural gas by means of a pump and which is expanded to produce useful work after having absorbed heat from the natural gas. Other processes which may advantageously employ the pressure-equalized heat transfer system of this invention are:

(1) Condensing of refrigerant gases such as propane, propylene, R–13B1 ($CBrF_3$), R–22 ($CHClF_2$), R–12 (CCl_2F), ammonia with water which has been pressurized by condensing pressure of the refrigerant. A process of this kind is described in detail in my application Ser. No. 547,690, filed April 5, 1966 entitled, Sea Water Power Plant.

(2) Evaporating refrigerant liquids such as R–22, R–12, R–13B1 at temperatures of about 40° F. with pressurized water which may then be employed for air conditioning purposes.

(3) Evaporating liquified gases at warm water temperatures, such as 80° F. to 90° F., for expansion through a turbine to produce power. A system of this kind is described in detail in my application Ser. No. 414,239, filed Nov. 27, 1964, entitled, Method and Apparatus for Evaporating Liquified Gases, now Patent No. 3,266,261.

(4) Evaporating liquid methane with another fluid at the methane pressure.

(5) Cooling gases such as nitrogen, oxygen and hydrogen at high presure to provide better heat transfer and better liquefaction cycles.

The invention will be further understood from the following detailed description in conjunction with the drawings in which:

FIGURE 1 is a schematic view of a system for cooling relatively high pressure natural gas with a propane refrigerant which is initially at relatively low pressure;

FIGURE 2 is a fragmentary schematic view showing a second manner of controlling the pump of FIGURE 1;

FIGURE 3 is a graph illustrating the pressure-flow characteristics of the centrifugal pump of FIGURES 1 and 2;

FIGURE 4 is a fragmentary schematic view showing a third manner of controlling the pump of FIGURE 1;

FIGURE 5 is a perspective view of the core of the heat exchanger of FIGURE 1;

FIGURE 6 is a vertical sectional view of another heat exchanger adapted for condensing and subcooling a liquid; and

FIGURE 7 is a schematic view of a system for cooling relatively high pressure natural gas with an evaporating refrigerant through an intermediate heat exchange fluid whose pressure is controlled at the cooling and evaporating stages so as to be about equal to the pressures of the natural gas and the evaporating refrigerant at these stages.

Referring to FIGURE 1 there is shown, schematically, and in the manner of a flow sheet, a system for cooling a stream of high pressure natural gas with a stream of pressurized liquid propane refrigerant in a heat exchanger 10 having a core 11. For purposes of illustration, the natural gas is assumed to be obtained from a source thereof, such as a tank and pump assembly 12, at a pressure of about 600 p.s.i.a. and a temperature of about 50° F. The gas enters the heat exchanger 10 through a line 14 and leaves through a line 16 at about 0° F. for transmittal to another storage tank 18.

According to the principles of the invention the cold refrigerant, liquid propane in this case, for cooling the natural gas is pressurized as by a centrifugal pump **20** to about the same pressure as the natural gas in the heat exchanger **10** and is subsequently expanded for the purpose of obtaining mechanical work therefrom. The pressurized liquid propane passes from the outlet of the pump **20**, through a line **22**, to the heat exchanger **10** where it passes in countercurrent relationship to the stream of warmer natural gas. From the heat exchanger **10** the warmed propane is conducted through a line **24** to a turbine **26** where its pressure energy is converted to mechanical energy in the form of rotation of the turbine shaft. The latter carries a gear **27** which meshes with a larger gear **28** on the shaft of an electric motor **29**. The gear **28** also meshes with a smaller gear **30** carried by the pump shaft so that the pump **20** is driven by both motor and turbine at a rotational speed that can be the same or different than either of the latter two elements, depending on design conditions. From the turbine **26** the propane, which may now be a mixture of liquid and vapor, is conveyed through a line **31** to the top of a cooler **32**. As shown, the cooler **32** is a conventional evaporative type, but other means may be provided for cooling the propane. Cooled liquid propane is withdrawn from the bottom of the cooler **32** through a line **34** which connects with the inlet of the propane pump **20**, the latter being located below the liquid level in the cooler so as to have sufficient vertical head to cause liquid to flow into the pump without cavitation.

The propane cooler **32**, which is of conventional construction, includes a vertical tank **36** containing a downwardly concave dome **38** supported therein by a spider **40**. A large diameter vapor outlet pipe **42** extends from the underside of the dome **38** to the exterior of the tank **36**. Part of the warm propane entering the top of the tank **36** through the line **30** evaporates to thereby cool the remaining liquid propane which collects in the bottom of the tank **36**. Propane vapor is withdrawn through the vapor outlet pipe **42** and is passed through a line **44** to a refrigeration-liquefaction system from which it is returned to the top of the tank **36** in liquid form through a line **46**. As is conventional the refrigeration-liquefaction system includes a motor driven compressor **48** which compresses the vapor and a condenser **50** which condenses the compressed vapor. As shown, the condenser **50** is a heat exchanger which is supplied with a continuous stream of cooling fluid through lines **52** and **54**. If the pressure in lines **52** and **54** is controlled so as to be nearly equal to the propane pressure in the heat exchanger **50** and the line **46**, the core of the heat exchanger **50** may be of the simple and economical construction utilized in the exchanger **10** and described in detail hereinafter.

The pressure in the propane cooler **32** is determined by the saturation temperature of the propane therein. In the illustrated embodiment the temperature in the cooler **32** is maintained at 30° F. by means of the refrigeration system **48**, **50** and the resulting pressure is 66.3 p.s.i.a. Accordingly, liquid propane is delivered to the inlet of the pump **20** at about 66.3 p.s.i.a.

Maintenance of the propane pressure at the outlet of the pump **20** at or near 600 p.s.i.a., which is the pressure of the natural gas in this illustration, may be achieved in several ways. In FIGURE 1 the left end or propane inlet end, of the heat exchanger **10** for cooling the natural gas is provided with a differential pressure responsive controller **56** which controls a conventional diaphragm valve **58** in the propane inlet line **24** to the turbine **26** in accordance with the difference in pressure between the natural gas and propane sides of the heat exchanger **10**. As shown, the device **56** is of a conventional type having a diaphragm **60** capable of transmitting a pneumatic signal upon movement thereof. One side of the diaphragm **60** is exposed to propane pressure by way of a conduit **62** leading from the propane side of the heat exchanger **10**,

and the other side is exposed to natural gas pressure on the gas side of the heat exchanger by way of a conduit **64**. The signal from the pressure responsive element passes by way of a line **66** to the control mechanism of the valve **58** which under normal operation is in an open position. An increase in propane pressure, or a decrease in natural gas pressure will cause the valve **58** to open wider, and a change in differential pressure in the opposite direction causes the valve **58** to partially close. In either case the action of the valve **58** tends to maintain the pressures of the propane and natural gas equal in the heat exchanger **10**.

Ordinarily, it will be desirable to control the propane flow more closely and independently of the natural gas pressure level. This may be done by means of resetting the controller **56** so that it will control the valve **58** to maintain a higher pressure differential between the propane and natural gas sides of the exchanger **10** at high propane flow than at low propane flow. In FIGURE 1 there is shown a second differential pressure controller **68** connected by means of lines **67** and **69**, between the propane inlet and outlet lines **22** and **24**. The signal from this controller **68** is responsive to the pressure drop across the exchanger and is thereby approximately proportional to the square of the propane flow rate. A line **70** carries the signal to the controller **56** where it resets the signal generating portion **72** of the latter. As indicated above, the controller **56** will normally be set to maintain a fixed pressure differential between propane and natural gas, but by means of the additional controller **68** the setting will be adjusted to produce greater differentials with increasing propane flow. While the controller **68** has been shown as a differential pressure device, it can be a differential temperature device inasmuch as the temperature increase of the propane stream across the heat exchanger is inversely proportional to propane flow rate assuming a constant heat transfer rate.

Alternatively the valve **58** can be controlled in accordance with the pressure differential between fluid streams at the propane outlet end of the heat exchanger **10**. This arrangement is shown schematically in FIGURE 2 wherein a pressure responsive device **56***a* is connected between the propane outlet line **24** and the natural gas side of the heat exchanger by means of conduits **62***a* and **64***a*, respectively, and wherein the signal of the controller **56***a* is conducted to the valve **58** by a line **66***a*. The advantage of this arrangement may be understood by referring to FIGURE 3 in which curve A is a typical characteristic curve for discharge pressure versus flow for a centrifugal pump at constant suction pressure and constant speed. If the pressure at the discharge of the propane pump **20** is controlled exactly at the pressure of the natural gas, then propane flow would vary from point B to point C as the natural gas pressure varied from a minimum (dotted line D) to a maximum (dotted line E). Curve F shows the propane pressure at the propane outlet of the heat exchanger **10**. This pressure is equal to the discharge pressure of the pump **20** less friction loss in the line **22** and heat exchanger. Since friction loss varies approximately as the square of the flow, curve F becomes steeper than curve A. Therefore, connecting the propane side of the pressure differential device to the propane outlet side of the heat exchanger will cause less changes in propane flow for a given change in natural gas pressure. However, the ratio of propane flow between maximum and minimum control pressure point is greater on outlet propane pressure curve F than on inlet propane pressure curve A. Therefore, depending on the effect on flow that is preferred, the controller **56** may be connected to the inlet line **22**, or the outlet line **24** from the exchanger.

Since it is plain that propane flow through the cooler is affected by the pressure controller and valve **58**, and this may be undesirable, it is obviously desirable in at least some cases to control pressure and flow independent-

ly. In order to do this another alternative in controlling pressure differential is illustrated in FIGURE 4 wherein a variable speed pump 20*b* is employed to pressurize the propane and wherein the speed of the pump 20*b* is controlled by the pressure diffential between the propane inlet and natural gas outlet of the heat exchanger 10. As shown, a differential pressure controller 56*b*, connected to the heat exchanger 10 as in FIGURE 1 by lines 62*b* and 64*b*, is connected by a control line 66*b* to the pump 20*b* in a manner to increase or decrease the pump speed upon a decrease or increase, respectively, in the propane pressure over the natural gas pressure. Where a constant speed motor is used it may be desirable to control pressure developed by the pump by the use of variable prerotation vanes at the impeller inlet, as is well known in centrifugal compressor and pump engineering. Valve 58, now used to control quantity rate of flow instead of pressure, is controlled through a line 76 from a differential pressure controller 68*b*, which is connected across the propane lines 22 and 24 by means of lines 67*b* and 69*b*. 67*b* and 69*b* could also be temperature sensing lines, making 68*b* a temperature rise controller. Alternatively, 68*b* could be a temperature controller, operating to control valve 58*b* in response to temperature at natural gas outlet 16. Valve 58*b* could also be a nozzle control on the turbine inlet instead of the throttle valve as shown.

Referring again to FIGURE 1 it will be seen that the heat exchanger 10 is provided with two equalizing valves 78 and 80 for bleeding fluid from one side of the partition walls to the other side in the event of a large pressure differential. Such a differential could be caused by stopping of the pump due to a power failure in which case the pressure in the propane side of the heat exchanger would drop far below the pressure in the natural gas side. A large differential in the opposite direction would occur in the event of a large leak in the natural gas piping. As shown, the valve 78 includes a chamber 82 located at the left end of the heat exchanger 10 and communicating with the propane side and the natural gas side thereof by means of conduits 84 and 86, respectively. Communication between the conduit 84 and the chamber 82 is normally prevented by a spring-biased valve member 88 contained within the latter. The valve 80 is similarly, but oppositely constructed with a chamber 90, conduits 92 and 94 and a spring-biased valve member 96 normally blocking passage of natural gas into the chamber 90. As will be apparent, a decrease in propane pressure (or an increase in natural gas pressure) sufficient to open the valve 80 bleeds natural gas into the propane side of the heat exchanger 10, and a decrease in natural gas pressure (or an increase in propane pressure) sufficient to open the valve 82 bleeds propane into the natural gas side of the heat exchanger 10.

As just mentioned, pump failure will cause high pressure natural gas to flow into the propane system. The latter system, as is conventional in refrigeration practice, includes a safety relief valve 97 in the line 31 which vents to atmosphere in the event of overpressure. To prevent loss of natural gas through such a vent arrangement, valves 98 and 100 are provided in the propane system to seal off the heat exchanger from the pump 20 and from the cooler 32. Valve 98 is a check valve located in the propane pipe 22 leading to the heat exchanger 10 from the pump 20 and prevents reverse flow into the latter. The valve 100 is an automatic shut-off valve in the propane pipe 24 leading from the heat exchanger to the turbine 26. The valve 100 is normally open but is arranged, through a control line 102, to close upon stopping of the pump 20. Some other variable, such as the opening of the equalizing valve 80 may be employed to activate the valve 100, if desired. Alternatively, the valve 58 and the controller 56 could be so arranged that the former would completely close upon a predetermined maximum increase of natural gas pressure over propane pressure at the propane inlet end of the heat exchanger 10.

Similarly, the natural gas system may be provided with a safety relief valve 104 and means for preventing loss of propane therethrough in the event of loss of natural gas pressure. In order to achieve this, the natural gas inlet pipe 14 may be provided with a check valve 106 which prevents reverse flow therethrough, and the gas outlet pipe 16 may be provided with an automatic shut-off valve 108 located upstream of the relief valve 104. The shut-off valve 108 may be controlled by a control line 110 which receives its signal from any suitable measuring device 112 which is responsive to a condition resulting from loss of natural gas pressure.

Referring now more specifically to the heat exchanger 10, and to FIGURE 5, it will be seen that the heat exchanger core 11 is economically constructed from a plurality of equi-sized, thin rectangular metal sheets 114 disposed in coextensive relationship and separated from each adjacent sheet 114 by a space. Every other space is divided into a plurality of parallel flow channels extending longitudinally through the core 11 from one end to the other by means of a plurality of elongated, parallel blocks 116 which engage the surface of adjacent sheets 114. The alternate spaces between sheets 114 are divided into a plurality of parallel flow channels extending transversely of the core 11 by means of spaced parallel blocks 118. The blocks 116 and 118 serve to maintain the spacing between the sheets 114 and may be constructed of metal, plastic or other material suitable for use with the fluids passed through the core 11. While the sheets 114 and blocks 116 and 118 are shown as having flat surfaces, they may be wave shaped to increase turbulence. If desired, the use of spacer blocks 116 and 118 as separate elements may be avoided by crimping the sheets 114 with V-shaped or U-shaped grooves and ridges to produce the desired spacing of the sheets 114. In any event, the structures which form the spacers need not be tightly sealed to the sheets 114 which they separate, and the only joints required are along the edges of the bundle of sheets 114. The bundle may be held together by mechanical clamps, or by soldering or brazing. As shown in FIGURE 1, the core 11 is mounted within the heat exchanger 10 with the sheets in vertical positions and is held in place by means of end sheets 120 which are sealed thereto in a manner to prevent passage of one fluid into the other.

Referring to FIGURE 6, there is shown a modified heat exchanger 10′ which is adapted to condense a gas and to subcool the liquid. In this construction the core 11′ is a double-pass type in which cold refrigerant flows from right to left in the lower longitudinal channels between the vertical sheets 114′ and from left to right in the upper longitudinal channels while gas to be condensed flows downwardly in the transverse channels defined by the sheets 114′ and the transverse spacer blocks 118′. The latter are horizontally slotted at 121 at locations below their midpoints. At the right end of the exchanger housing is a refrigerant inlet conduit 122 and a refrigerant outlet conduit 124 located above the latter and separated therefrom within the exchanger housing by a horizontal plate 126 which extends into engagement with the right end of the core 11′ so as to prevent communication between the upper and lower longitudinal channels at that end. A gas inlet 128 is provided in the top wall and a condensed gas outlet 130 is provided in the bottom wall near the refrigerant inlet conduit 122. A vertical baffle 132, transverse to the direction of refrigerant flow, is located near the condensed gas outlet 130 on the side opposite the refrigerant inlet 122 so that gas which is condensed to the left of the baffle 132 collects in the bottom of the exchanger and then overflows through the slots 121 in the spacer blocks 118′. By this arrangement the liquid must pass through the coldest channels before

being discharged from the exchanger through the outlet 130. In operation it will usually be desired to adjust the refrigerant temperature and/or the incoming gas pressure so that the condensed gas is cooled to below the saturation temperature corresponding to the incoming gas pressure.

The operation of the gas-cooling system of FIGURE 1 has been described in general terms and has as its practical end result merely the cooling of natural gas in the heat exchanger 10 at about 600 p.s.i.a. from a temperature of about 50° F. to a lower temperature, say 35° F. The details of the invention, as applied to this particular system, are concerned with effecting the cooling at as low a cost as possible. In order to make possible an economical heat exchanger, the refrigerant, in this case liquid propane, is delivered to the heat exchanger at about the same pressure as the natural gas so that the heat exchanger core 11 may be constructed with very thin, low-cost partition sheets 114. The propane at about 30° F. is delivered to the heat exchanger 10 through the pipe 22 after having been pressurized to about 600 p.s.i.a. by the pump 20. There, the propane passes through the longitudinal channels in the core 11 in countercurrent relationship to the stream of warm natural gas, the direction of flow of the latter being indicated by the arrows in FIGURE 1.

Warm liquid propane, say at 45° F., then passes out of the heat exchanger 10 by way of the pipe 24 and is passed to the inlet of the turbine 26. Here, pressure energy is converted to mechanical energy by expanding the propane so that some of the liquid flashes into gas. The resulting power is added to that produced by the motor 29 and is transmitted to the drive shaft of the pump 20. Preferably the propane will be expanded to about the pressure which corresponds to the evaporation temperature of the liquid propane being delivered to the heat exchanger. For the purpose of this particular system the desired liquid propane temperature in the heat exchanger inlet is about 30° F. At this temperature liquid propane boils at 66.3 p.s.i.a. and, accordingly, the turbine 26 will preferably expand the propane to this pressure. The expanded propane is then passed to the conventional evaporation cooler 32, the operation of which has been described above. Cool liquid propane at about 30° F. and at a pressure of about 66.3 then passes from the cooler 32 to the pump 20 by way of pipe 34.

The control of propane pressure at the outlet of the pump 20 to match natural gas pressure by means of the controllers 56 and 68 and the valve 58 has been described above as have the alternative schemes of FIGURES 2 and 4. While the system has been described as seeking to have propane pressure substantially the same as the natural gas pressure it may be desirable, in some cases, to have a small fixed pressure differential between the two streams. However, any differential should be so small that it does not require rugged construction of the heat exchanger core 11.

Referring further to the recovery of energy by the turbine 26 and its reuse to drive the pump 20, it can be shown thermodynamically that compression of liquid from 30° F. and 66.3 p.s.i.a. to 600 p.s.i.a. requires less work per pound of fluid along an isentropic process path than would be produced along an isentropic path by expansion from 600 p.s.i.a. and 35° F. to a pressure of 66.3 p.s.i.a. This is a well recognized heat engine cycle of the type involving isentropic compression of a fluid, followed by constant pressure heating to a higher temperature, followed by isentropic expansion to the low pressure, followed by constant pressure cooling to the original low temperature. This is a heat-power cycle, whether the fluid remains in the liquid state throughout, the gaseous state throughout, or is partly liquid and partly gas.

In the propane circuit described herein the energy losses

in the pump 20 and the turbine 26 and the pressure drop in the flow through the heat exchanger 10 are somewhat greater than the theoretical excess energy made available by the above-described isentropic expansion compression. Therefore, the motor 29 is required for supplying the additional work for pressurizing the propane. Conceivably, if the propane temperature were raised high enough in the heat exchanger 10, the turbine 26 could drive the pump 20 without an external power source. Of course, work in the form of cooling the propane in the cooler 32 or its equivalent, is continually being done on the propane circuit.

In FIGURE 7 there is shown a system for indirectly cooling relatively high pressure natural gas with a primary refrigerant through an intermediate heat transfer fluid. The intermediate fluid is passed from the natural gas cooling stage through a power producing turbine to a primary refrigerant evaporating stage and thence to a pressurizing pump which recycles it to the natural gas cooling stage at about the pressure of the natural gas. The evaporated refrigerant is compressed, condensed and recycled to the evaporating stage.

Referring more specifically to FIGURE 7 there is shown a natural gas cooler 136 which is a heat exchanger constructed in the same manner as the heat exchanger 10 of FIGURES 1–6 and adapted to receive a stream of natural gas from an inlet line 138 and to discharge the cooled stream through an outlet line 140. The circuit which provides the natural gas cooler with a cold stream of intermediate heat transfer fluid includes an inlet line 142 leading from a pump 144 and an outlet line 146 leading to the inlet of a turbine 148. A turbine exhaust line 150 passes from the outlet of the turbine 148 to a heat exchanger 152 where the intermediate fluid is cooled with an evaporating refrigerant, and a pump inlet line 154 extends from the outlet of the heat exchanger 152 to the pump 144. As in the previously described system the shaft of the turbine 148 is connected to aid a motor 156 in driving the pump 144.

The heat exchanger 152, or refrigerant evaporator, which is also of the previously described economical construction receives the liquid refrigerant from a refrigerant condenser 158 through a line 160. A refrigerant vapor return line 162 containing a compressor 164, extends from the refrigerant evaporator 152 back to the condenser 158. The condenser 158 is cooled by a stream of cooling medium entering through a line 166 and leaving through a line 168.

The control system for the arrangement of FIGURE 7 includes a differential pressure responsive controller 170 connected across the natural gas and intermediate fluid sides of the cooler 136 and provided with an output line 172 leading to the pump 144 for controlling its outlet pressure. Another differential pressure responsive controller 174 is connected between the inlet and outlet lines 142, 146 of the intermediate fluid circuit and is provided with an output line 176 leading to a diaphragm valve 178 in the line 146 for controlling the flow rate of intermediate fluid.

For maintaining the pressure of the intermediate fluid in the refrigerant evaporator 152 at about the same level as the pressure of the refrigerant therein there is provided a differential pressure responsive controller 180 connected across the intermediate fluid and refrigerant sides of the evaporator 152. An output line 182 from this controller 180 leads to a valve actuator 184 which controls the position of two valves 186, 188 in a surge system for the intermediate fluid. This system includes a surge tank 190, an inlet line 192 from the line 150 to the tank 190 and an outlet line 194 from the tank 190 back to the line 150 at a point downstream of the connection of the inlet line 192. The valve 186 is contained in the inlet line 192 and the valve 188 is contained in the outlet line 194. The outlet line 194 contains in addition a pump 196 located between the tank 190 and the valve

188 for pumping intermediate fluid back into the line 150. The actuator 184 is of a conventional type which employs a diaphragm 198 one side of which is exposed to the control pressure from the controller 180. The stems of the valves 186 and 188 are connected to the diaphragm 198 in such a manner that movement of the latter opens one valve and closes the other. Specifically, the arrangement is such that if refrigerant pressure in the evaporator 152 exceeds intermediate fluid pressure, as measured by the controller 180 then the actuator 184 tends to open the valve 188 and to close the valve 186.

In use the system of FIGURE 7 may be employed to cool a stream of natural gas entering through the line 138 and leaving through the line 140. For purposes of illustration the natural gas is assumed to enter the cooler 136 at 600 p.s.i.a. and −30° F. and to leave the cooler 136 at about the same pressure at −60° F. A suitable primary refrigerant for producing the necessary cooling effect is R–13B1. This material is relatively expensive, and the use of a less expensive intermediate heat transfer fluid may be desirable to offset the cost of the unavoidable losses which occur in the power-producing circuit. In this example, butane has been selected as the intermediate fluid.

In accordance with the already described principles of the present invention the butane intermediate fluid at a suitable low temperature of, for example, −70° F. is pressurized to the pressure of the natural gas in the cooler 136 in order that the flow channels in the latter be subjected to minimum stress. This is accomplished by the pump 144 which raises the pressure of cold liquid butane from the refrigerant evaporator 152. The controller 170 is responsive to any pressure differential which may occur across the gas-liquid flow channels and operates through the control line 172 to change the output pressure of the pump 144 in a manner to maintain the pressure differential at a very small value. Butane flow rate through the cooler 136 is maintained substantially constant by the controller 174 which adjusts the valve 178 in a manner to maintain a predetermined pressure drop across butane inlet and outlet of the cooler 136.

Warmed liquid butane from the cooler 136 is expanded in the turbine 148 and the work thereby produced is applied to augment the motor 156 in driving the pump 144. The expanded butane passes from the turbine 148 through the refrigerant evaporator 152 where its temperature is lowered to −70° F. by the evaporating R–13B1. From the evaporator 152 the cold butane flows to the pump 144 where its pressure is again raised to 600 p.s.i.a., the pressure of the stream of natural gas.

In order to produce the desired temperature of −70° F. the R–13B1 must be expanded to 15.5 p.s.i.a. which is its saturation pressure at that temperature. The stream of expanded butane from the turbine 148 should therefore be maintained at about 15.5 p.s.i.a. in order that the refrigerant evaporator 152 may be operated with a minimum pressure differential across its flow channels. To accomplish this the surge tank 190 and its associated equipment operates in the following manner. The surge tank pump 196 withdraws liquid butane from the surge tank 190 and pressurizes it to above the pressure in the line 150. If the valve 188 is open the butane flow into the line 150 and increase the pressure therein. If the valve 186 is then opened, butane from the line 150 will flow into the tank 190 through the line 192 thus reducing the output pressure of the pump 196. Opening and closing of the valves 186 and 188 to maintain the butane in the line 150 at the same pressure as the refrigerant in the vaporator 152 is effected by means of the controller 180. Any pressure differential across the channels in the evaporator 152 results in a signal in the control line 182 to the valve actuator 184. If butane pressure falls below refrigerant pressure the actuator 184 opens the valve 188 and simultaneously closes the valve 186. If butane pressure rises above refrigerant pressure, the actuator closes the valve 188 and opens the valve 186.

The surge system may also be operated, if the pressure in the tank 190 is higher than that in the line 150 by placing the pump 196 in the line 192 so that it will pump butane into the tank 190 rather than out of the tank 190.

It will be appreciated that the heat transfer system described herein will, in practice, include various components which are conventional in the fluid flow and refrigeration art. These components such as expansion valves, shut-off valves, gauges and other control equipment are omitted from this description in the interest of simplicity.

It will be understood that the principles of the present invention are not limited to the cooling of a gas and that the disclosed embodiments are given by way of illustration only. The concept of carrying out heat exchange between fluids in a thin-walled heat exchanger by first pressurizing one fluid to the pressure of the other is applicable to both cooling and heating processes and to condensing and evaporating process as well. Further, the recovery of some of the energy of pressurizing the low pressure fluid is applicable to other processes. It is therefore not intended that the details described herein be limiting except as they appear in the appended claims.

What is claimed is:

1. Apparatus for changing the heat content of a first stream at a given superatmospheric pressure by means of a second fluid stream at another temperature and at a lower pressure than the pressure of the first stream comprising: a heat exchanger having distinct first and second sets of flow channels for the first and second fluids, said flow channels being defined by thin walls; means for passing the first fluid stream through said first set of flow channels at the pressure of the first stream; power driven pump means for pressurizing the second fluid stream from the lower pressure to the pressure of the first stream, said pump means having an inlet and an outlet; conduit means connecting the pump outlet to said second set of flow channels whereby the pressure on each side of said channel-defining walls is substantially equal thereby permitting said walls to be of thin construction conducive to high heat transfer between first and second fluids; and conduit means connecting said pump inlet with the second fluid stream at said lower pressure.

2. Apparatus in claim 1 further comprising power producing turbine means having a fluid inlet, a fluid outlet and output shaft; conduit means for conducting the second fluid from said second set of flow channels to said turbine inlet whereby rotation of said turbine shaft is effected; and means for conducting work from said shaft to said pump means to drive the same in a direction to pressurize the second fluid stream.

3. Apparatus as in claim 2 further comprising control means responsive to a pressure differential of a predetermined magnitude between said first and second sets of flow channels for adjusting the output of said pump in a manner to restore said pressure differential to a value below said predetermined magnitude.

4. Apparatus as in claim 2 further comprising an equalizing valve connected between said first and second sets of flow channels, said valve having a movable closure member normally blocking communication between said sets and movable to a position permitting fluid flow between sets upon the occurrence of a pressure difference between said sets of a predetermined magnitude whereby damage to said channel-defining walls is prevented.

5. Apparatus as in claim 3 wherein said control means includes: a differential pressure responsive device connected across said sets of flow channels for generating a signal which is a function of the difference between the pressure differential across said sets and a predetermined value; valve means in said conduit connecting said second flow channels with said turbine inlet, said

valve means being responsive to said signal to open and close between limits to thereby control turbine speed; and means responsive to the rate of flow through said second set of flow channels to change said predetermined value of said differential pressure controller.

6. Apparatus as in claim 2 further including: check valve means in said conduit means connecting said pump outlet with said second set of flow channels for preventing flow from said heat exchanger to said pump outlet; and automatic valve means in said conduit means for connecting said turbine inlet with said second set of flow channels, said automatic valve means being responsive to loss of pressure at said pump outlet for completely closing.

7. A method of changing the heat content of a mass of fluid which is available at generally constant pressure which comprises: passing a stream of said fluid at said pressure through a set of flow channels in a heat exchanger of the type having distinct flow channels for separate fluid streams; mechanically pressurizing a second fluid at an initial pressure lower than said constant pressure and at an initial temperature different from that of said first-mentioned fluid, to about said constant pressure; passing a stream of said second fluid to a different set of flow channels in said heat exchanger at said constant pressure whereby both said fluid streams undergo a change in heat content.

8. A method as in claim 7 further comprising: restoring said stream of second fluid to its initial temperature after passage through the heat exchanger and subsequently recirculating it to said pressurizing operation.

9. A method as in claim 7 wherein said generally constant pressure is substantially above atmospheric pressure, wherein said second fluid is passed through the heat exchanger in liquid form and wherein said second fluid partially vaporizes during said pressure reduction step.

10. A method as in claim 9 wherein said second fluid is maintained at a temperature below that of said first-mentioned fluid and wherein said pressure reduction step reduces the pressure of said second fluid to substantially the saturation pressure of the liquid phase of said second fluid at said temperature thereof.

11. A method of changing the heat content of a mass of fluid which is available at generally constant pressure which comprises: passing a stream of said fluid at said pressure through a set of flow channels in a heat exchanger of the type having distinct flow channels for separate fluid streams; mechanically pressurizing a second fluid at an initial pressure lower than said constant pressure and at an initial temperature different from that of said first-mentioned fluid, to about said constant pressure; passing a stream of said second fluid to a different set of flow channels in said heat exchanger at said constant pressure whereby both said fluid streams undergo a change in heat content; obtaining mechanical work from said stream of pressurized second fluid after its passage through the heat exchanger by reducing the pressure of said stream; applying said mechanical work to said second fluid at its initial pressure to thereby augment said pressurizing step.

12. A method of changing the heat content of a mass of fluid which is available at generally constant pressure which comprises: providing a source of primary heat transfer fluid at a generally constant pressure and generally constant temperature different from the temperature of said mass of fluid; passing a stream of said primary heat transfer fluid in heat exchange relationship with a stream of intermediate heat transfer fluid in a heat exchanger of the type having distinct flow channels for separate fluid streams; adjusting the pressure of said intermediate fluid in said heat exchanger to about the pressure of said primary fluid in said heat exchanger whereby mechanical stress in said flow channels due to pressure differentials is maintained at a low value; passing a stream of said intermediate fluid from said heat exchanger in heat exchange relationship with a stream of said mass of fluid to change the heat content of the latter stream to a desired value, in another heat exchanger of the type having distinct flow channels for separate fluid streams; adjusting the pressure of said intermediate fluid in said other heat exchanger to about the pressure of the stream of said mass of fluid in said other heat exchanger whereby mechanical stress in the flow channels thereof due to pressure differentials is maintained at a low value; and returning said stream of intermediate fluid to said first-mentioned heat exchanger.

13. A process as in claim 12 wherein said first-mentioned heat exchanger is operated at a pressure below the pressure of said mass of fluid whose heat content is to be changed and wherein said stream of intermediate fluid is mechanically pressurized to about the pressure of said mass of fluid before passing to said other heat exchanger, said process further comprising the step of obtaining mechanical work from said stream of intermediate fluid after its passage through said other heat exchanger by reducing the pressure of said stream, said stream being subsequently pressurized in said mechanical pressurization step.

No references cited.

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