

## **LIQUID-COOLED HEAT EXCHANGER IN A VAPOR COMPRESSION REFRIGERATION SYSTEM**

### **Cross Reference to Related Applications**

**[0001]** This application claims priority to U.S. Provisional Patent Application Serial No. 61/246,239 entitled “Liquid-Cooled Heat Exchanger in a Vapor Compression Refrigeration System” filed on September 28, 2009. The content of this application is incorporated herein by reference in its entirety.

### **Field of the Disclosure**

**[0002]** This disclosure relates generally to vapor compression refrigeration systems and, more particularly, to a method and apparatus for liquid cooling a refrigerant in a vapor compression refrigeration system.

### **Background of the Disclosure**

**[0003]** A transport refrigeration system used to control enclosed areas, such as the insulated box used on trucks, trailers, containers, or similar intermodal units, functions by absorbing heat from the enclosed area and releasing heat outside of the box into the environment. The transport refrigeration system commonly includes a compressor to pressurize refrigerant vapor, and an air-cooled condenser to decrease the temperature of the pressurized vapor exiting the compressor, thereby changing the state of the refrigerant from a vapor to a liquid. Ambient air is blown across the refrigerant coils in the condenser to effect the heat exchange. The system further includes an evaporator for drawing heat out of the box by drawing or pushing return air across refrigerant-containing coils within the evaporator. Any remaining liquid refrigerant flowing through the coils within the evaporator is vaporized, then drawn back into the compressor to complete the circuit.

**[0004]** Some transport refrigeration systems require operation in adverse conditions, such as in tropical climates or on container ships at sea. One problem that may occur under these conditions is that the ambient air blown across the refrigerant coils in the condenser may be at a relatively high temperature, thereby reducing the effectiveness of the heat exchange within the condenser. In some cases, for example in the cargo hold of a container ship, the

ambient air may not be available at all. Insufficient heat exchange within the refrigeration system results in insufficient cooling in the cargo container, which may in turn lead to spoiling of the contents in the cargo container.

### **Summary of the Disclosure**

**[0005]** In accordance with one aspect of the disclosure, a system and method is provided to cool refrigerant vapor in a refrigerant vapor compression system. The system includes a compressor having a suction port and a discharge port, an air-cooled heat exchanger operatively coupled to the discharge port of the compressor, a liquid-cooled heat exchanger operatively coupled to the air-cooled heat exchanger, a coolant pump operatively coupled to a liquid coolant inlet conduit of the liquid-cooled heat exchanger, an evaporator heat exchanger unit operatively coupled to the liquid-cooled heat exchanger and the suction port of the compressor, and a controller operatively associated with the coolant pump for controlling the flow of liquid coolant into the liquid-cooled heat exchanger.

**[0006]** In one embodiment, the liquid-cooled heat exchanger comprises a low-profile enclosure defining an interior volume. The enclosure has a liquid coolant inlet port and a liquid coolant discharge port fluidly coupled to the interior volume, and a continuous refrigerant tube sealingly disposed within the enclosure. The refrigerant tube is fluidly isolated from and in heat exchange relationship with the interior volume in which the liquid coolant flows.

**[0007]** In another embodiment, the liquid-cooled heat exchanger further comprises a baffle disposed within the enclosure. The baffle defines a second interior volume within the first interior volume, and includes at least one aperture fluidly coupling the second interior volume to a remaining portion of the first interior volume. The continuous refrigerant tube is helically-wound within the first and second interior volumes.

**[0008]** In another aspect of the disclosure, a method for liquid-cooling a vapor refrigerant in a refrigerant vapor compression system is disclosed. The method comprises the steps of providing an air-cooled heat exchanger, a liquid-cooled heat exchanger coupled to the air-cooled heat exchanger, and a continuous refrigerant tube extending through a first volume of the liquid-cooled heat exchanger. A vapor refrigerant is flowed through the air-cooled heat exchanger and, if the refrigerant is not sufficiently cooled, the refrigerant is flowed through

the liquid-cooled heat exchanger simultaneously with a liquid coolant flowing through the first interior volume of the liquid-cooled heat exchanger.

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

[0009] For a further understanding of the disclosure, reference will be made to the following detailed description of the disclosure which is to be read in connection with the accompanying drawing, wherein:

[0010] FIG. 1 is a schematic view of a refrigeration system including a liquid-cooled heat exchanger in accordance with the present disclosure;

[0011] FIG. 2 is perspective view of one embodiment of the liquid-cooled heat exchanger of FIG. 1;

[0012] FIG. 3 is a top plan view of the liquid-cooled heat exchanger of FIG. 2, in cross section;

[0013] FIG. 4 is a side plan view of the liquid-cooled heat exchanger of FIG. 2, in cross section;

[0014] FIG. 5 is a side plan view of another embodiment of the liquid-cooled heat exchanger of FIG. 1, in cross section; and

[0015] FIGS. 6A and 6B are cross sectional views of another embodiment of the liquid-cooled heat exchanger of FIG. 1.

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

[0016] One approach that has been employed to address the problem of insufficient heat exchange within the refrigeration system is to augment the typical air-cooled condenser with an inline liquid-cooled heat exchanger downstream of the air-cooled condenser. In one example transportation refrigeration system, refrigerant vapor flows through the heat exchanger and impinges on coolant tubes disposed therein. The coolant tubes flow a liquid coolant, which is often water. The warm refrigerant vapor condenses as it contacts the cold coolant tubes. In order to assure as much vapor as possible impinges on the tubes, the tubes are designed to maximize surface area, typically by utilizing small diameter tubes or coiling the tubes within the heat exchanger, or both. The condensed refrigerant is carried by gravity to a collection point in the bottom of the shell. A drain port located proximate to the collection

point allows the condensed liquid refrigerant to accumulate and be piped to the next component in the system.

**[0017]** One problem noted with this approach is that the liquid-cooled heat exchanger has to be constructed of high strength materials because the heat exchanger enclosure is a pressure vessel. The coolant tubes disposed in the enclosure occupy a large volume so as to maximize contact with the refrigerant vapor. The vapor pressure of the refrigerant flowing within the heat exchanger enclosure could be 74 bars or more (1,000 pounds per square inch). Thus, the large surface area at a high pressure requires a thick-walled pressure vessel, high strength materials, or both. High strength materials result in a condenser that is expensive to manufacture.

**[0018]** Another problem noted with such an arrangement is that the temperature of the refrigerant vapor entering the enclosure is high, approximately 140 °C. Accordingly, the enclosure must be designed to withstand the elevated temperatures without sacrificing strength.

**[0019]** Another, separate problem encountered with current refrigeration systems is that the environmental impact of common refrigerants has come under great scrutiny. Until recently, common refrigerants used in refrigeration systems included chlorofluorocarbons (CFCs) such as trichlorofluoromethane (R-11) and dichlorodifluoromethane (R-12). However, these refrigerants are being phased out due to their harmful effects to the ozone layer. Hydrochlorofluorocarbon (HCFC) refrigerants such as chlorodifluoromethane (R-22) have taken their place because they are energy-efficient, low-in-toxicity, cost effective and can be used safely. However, HCFCs still contribute to ozone depletion, although at a much smaller rate. A recent trend has been to replace HCFCs with hydrofluorocarbons (HFCs) such as tetrafluoroethane (R-134a), which contain no chlorine and have no known effects on the ozone layer. However, there is increasing concern that HFCs contribute to greenhouse gases which, although not ozone-depleting, may cause global warming. Most recently, carbon dioxide (R-744) has re-gained favor as a commercial refrigerant because of its high volumetric cooling capacity and absence of harmful environmental effects.

**[0020]** Design requirements for carbon dioxide refrigerant systems are very different from other refrigerant systems such as R-134a. One major difference attributable to carbon dioxide as a refrigerant is that it undergoes a room temperature phase transition from gas to

liquid at a pressure of approximately 6000 kilopascal (870 pounds per square inch). By comparison, R-134a undergoes a room temperature phase transition at approximately 700 kilopascal (100 pounds per square inch). Consequently, vapor compression refrigeration components designed to condense carbon dioxide to a liquid must be designed to withstand approximately eight times more pressure than systems utilizing R-134a, thereby exacerbating the problem of designing components as pressure vessels.

**[0021]** To compound the problem further, designers typically design the refrigerant system to operate in a transcritical cycle, often at pressures of 7,500-12,500 kilopascal (1100-1800 pounds per square inch). Designing a refrigeration system to safely operate at this higher pressure is problematic and expensive.

**[0022]** Another problem encountered in a carbon dioxide refrigerant system is that prior art liquid-cooled heat exchangers perform inadequately because the supercritical carbon dioxide vapor does not condense to a liquid. That is to say, the refrigerant enters the heat exchanger as a warm vapor, and exits the heat exchanger as a cool vapor, instead of a cool liquid. Therefore, the gravity-feed mechanism and the condensate collection means utilized in the prior art liquid-cooled heat exchangers are of little use.

**[0023]** The inventors have addressed these problems and limitations by designing a liquid-cooled heat exchanger that flows the refrigerant vapor through a single continuous tube, and simultaneously flows a cooling liquid around the tube. The condenser is particularly well adapted for use in a transport refrigeration system.

**[0024]** Referring to FIG. 1, there is shown schematically an exemplary embodiment of a refrigerant vapor compression system 10 according to the present disclosure. The system 10 includes a compressor 12, such as a reciprocating piston compressor, to compress refrigerant to a higher temperature and pressure. In the illustrated example, the refrigerant is carbon dioxide (CO<sub>2</sub>). The system further includes a first heat exchanger 14 which passes the supercritical refrigerant in heat exchange relationship with a cooling medium, such as ambient air. In the illustrated example ambient air is driven by a fan 16. In many refrigeration systems, first heat exchanger 14 is a condenser adapted to condense at least some of the superheated refrigerant to a liquid. However, since the working fluid of the present embodiment is CO<sub>2</sub> operating in a transcritical cycle, it does not condense to a liquid. Rather, sensible heat is

removed from the supercritical refrigerant and the now-cooler refrigerant gas exits the first heat exchanger 14 through conduit 15.

**[0025]** In some refrigeration systems, such as a transport refrigeration system on a ship at sea, ambient air may be limited or unavailable for the first heat exchanger 14. Or, prevailing ambient conditions may dictate that although ambient air is available, the temperature is too high to effect an adequate heat exchange with the refrigerant. Therefore, the system 10 further includes a second heat exchanger 18 in serial relationship with the first heat exchanger 14. The second heat exchanger 18 is adapted to cool the refrigerant to an adequate temperature in the event the first heat exchanger 14 is unable to do so. In the illustrated example, the second heat exchanger 18 is a liquid-cooled tube-in-shell heat exchanger, as will be discussed in detail below. A pump 20 is adapted to pump a liquid coolant 21 (not shown) through the second heat exchanger 18. The liquid coolant 21 discharged from the second heat exchanger 18 through conduit 60 may be re-circulated through an auxiliary heat exchanger (not shown) or dumped. In the illustrated example, the refrigerant vapor compression system 10 is adapted to utilize water as the liquid coolant 21. In one embodiment, a container ship transporting cargo, such as cargo stored in the refrigerant vapor compression system 10 disclosed herein, pumps water through a network of pipes for a variety of on-board functions. The water, sometimes referred to as “grey water,” flows or circulates continuously through the network of pipes and may be tapped into to satisfy coolant requirements. The availability of the grey water makes it an ideal choice for the liquid coolant 21. In the illustrated example, the grey water discharged from the second heat exchanger 18 is fed back to the network of pipes.

**[0026]** In one example, the cooled refrigerant vapor flows from the second heat exchanger 18 through a first expansion device 22 and a flash tank receiver 24. As the CO<sub>2</sub> refrigerant leaves the second heat exchanger 18, it passes through the first expansion device 22 where it expands to a lower pressure and enters the flash tank receiver 24 as a mixture of liquid refrigerant and vapor. The flash tank receiver 24 operates as a charge control tank. The liquid refrigerant settles in the lower portion of the flash tank receiver 24 and the refrigerant vapor collects in the upper portion of the flash tank receiver 24.

**[0027]** The liquid refrigerant passes from the flash tank receiver 24 to a second expansion device 26 where it expands to a lower pressure and temperature before entering an

evaporator 28. The evaporator 28 typically includes tubes or coils (not shown) through which the refrigerant flows in heat exchange relationship with a heat medium so as to vaporize the remaining liquid refrigerant. The heat medium is typically the return air from a refrigerated cargo box 30. The return air is preferably drawn or pushed across the tubes or coils by at least one evaporator fan 32. The low pressure refrigerant vapor leaves the evaporator 28 and then returns to the suction port 33 of the compressor 12.

**[0028]** The system 10 further includes a controller 950 to monitor and control many of the points in the refrigeration system 10. The controller 950 includes a microprocessor board 952 that contains a microprocessor 954 and its associated memory 956. The memory 956 of the controller 950 can contain operator or owner preselected, desired values for various operating parameters within the system 10 including, but not limited to, temperature set points for various locations within the system 10 or the cargo box 30, pressure limits, current limits, engine speed limits, and any variety of other desired operating parameters or limits with the system 10. In the illustrated example, the controller 950 includes an input/output (I/O) board 958, which contains an analog to digital converter 960 which receives temperature inputs and pressure inputs from various points in the system, AC current inputs, DC current inputs, voltage inputs and humidity level inputs. In addition, I/O board 958 includes drive circuits or field effect transistors ("FETs") and relays which receive signals or current from the controller 950 and in turn control various external or peripheral devices in the system 10, such as the pump 20, for example.

**[0029]** Among the specific components controlled by controller 950 are the first expansion device 22 and a motor 17 for the fan 16. Among the specific sensors and transducers monitored by controller 950 are the return air temperature (RAT) sensor which inputs into the microprocessor 954 a variable resistor value according to the evaporator return air temperature; the ambient air temperature (AAT) sensor which inputs into microprocessor 954 a variable resistor value according to the ambient air temperature read in front of the condenser 16; the compressor suction temperature (CST) sensor, which inputs to the microprocessor a variable resistor value according to the compressor suction temperature; the compressor discharge temperature (CDT) sensor, which inputs to microprocessor 954 a resistor value according to the compressor discharge temperature inside the dome of compressor 12; the evaporator outlet temperature (EVOT) sensor, which inputs to

microprocessor 954 a variable resistor value according to the outlet temperature of evaporator 28; the compressor suction pressure (CSP) transducer, which inputs to microprocessor 954 a variable voltage according to the compressor suction value of compressor 12; the compressor discharge pressure (CDP) transducer, which inputs to microprocessor 954 a variable voltage according to the compressor discharge value of compressor 12; and the evaporator outlet pressure (EVOP) transducer which inputs to microprocessor 954 a variable voltage according to the evaporator outlet pressure of evaporator 28.

**[0030]** The system 10 optionally includes an economizer circuit. In the illustrated example, the flash tank receiver 24 operates not only as a charge control tank, but also as a flash tank economizer. Vapor refrigerant collecting in the upper portion of the flash tank receiver 24 above the liquid level passes from the receiver 24 along conduit 34 through solenoid valve 36 to a vapor injection port 38. The solenoid valve 36 is controlled by the controller 950 in order to turn on and off the economizer operation. Other economizer circuits are possible without departing from the scope of the disclosure. In another embodiment, a brazed plate heat exchanger (not shown) is included in the system instead of the flash tank.

**[0031]** In the event that the compressor 12 tends to operate at elevated temperatures, it is desirable to provide liquid injection into the vapor injection port 38 or an alternative port for liquid. Accordingly, conduit 40 and associated solenoid valve 42 are provided for that purpose.

**[0032]** Referring to FIG. 2, one possible construction of the second heat exchanger 18 is shown in greater detail. In the disclosed embodiment, the second heat exchanger 18 is a liquid-cooled tube-in-shell construction including an enclosure 44. The enclosure 44 includes a first refrigerant tube opening 46, a second refrigerant tube opening 48 (hidden from view), a first coolant port 50, and a second coolant port 52. In the illustrated embodiment, the first coolant port 50 is the liquid coolant inlet port, and the second coolant port 52 is the coolant discharge port. The fluid exiting the second coolant port 52 may comprise liquid, vapor, or both. The first coolant port 50 may be adapted with a fitting 54 to sealingly secure a coolant inlet conduit 56 to the enclosure 44. The second coolant port 52 may be similarly adapted with a fitting 58 to sealingly secure the coolant discharge conduit 60.

**[0033]** The enclosure 44 has a low profile for space efficiency. By “low profile”, what is meant is that at a cross section taken through a thickness T, the length of the cross section is

at least five times greater than the thickness. Expressed as a ratio, a low-profile enclosure has an aspect ratio of at least 5:1. In this manner, a refrigerant tube encased within the enclosure 44 will have ample exposure to a coolant fluid for heat transfer without the enclosure 44 having to take up a large volume. In the illustrated embodiment, the planar portion of the enclosure 44 is rectangular, having dimensions of approximately 5.5 inches wide and 4.0 inches long. The enclosure 44 is 0.5 inches thick. Accordingly, the enclosure 44 has an aspect ratio of approximately 11:1 through the cross section that is 5.5 inches wide, and an aspect ratio of 8:1 through the cross section that is 4.0 inches long. Both aspect ratios are greater than 5:1, so the enclosure 44 is a low-profile enclosure.

**[0034]** Referring to FIG. 3 of the drawings, the enclosure 44 further includes a first interior surface 62 defining a first interior volume 64. Disposed within the first interior volume 64 is a continuous refrigerant tube 66. By continuous, what is meant is that there are no breaks or gaps in the line; refrigerant flows continuously from an inlet portion to an outlet portion. In the illustrated example, the refrigerant tube 66 is a single-piece construction to minimize the possibility for leakage. A first end 68 of the refrigerant tube 66 passes through the first refrigerant tube opening 46, and is sealed by conventional means such as a rubber grommet 70. A second end 72 of the refrigerant tube 66 passes through the second refrigerant tube opening 48, and is similarly sealed by another rubber grommet 70. The refrigerant tube 66 may be serpentine to maximize exposure to the liquid coolant flowing through the first interior volume 64. As used herein, “serpentine” means winding or turning one way and another.

**[0035]** Although not shown in the drawings, the continuous refrigerant tube 66 may be forked, divided, branched, or otherwise split into one or more passages within the enclosure 44. The branched passages converge at a point to provide continuous refrigerant flow.

**[0036]** The refrigerant tube 66 may be supported within the first interior volume 64 by fasteners or the like (not shown), or may be captured and held in place by the first interior surface 62. As illustrated in FIGS. 3 and 4, the enclosure 44 may have ribs 74 periodically spaced within the first interior surface 62. The ribs 74 may include semi-circular cutouts or grooves formed therein to capture and secure the refrigerant tube 66. The ribs 74 are spaced liberally so as to minimize the obstruction to the liquid coolant 21 flowing through the first

interior volume 64, but may also be advantageously located to channel the flow of the coolant within the interior surface.

**[0037]** Turning now to FIG. 5 of the drawings, wherein like numerals indicate like elements from FIGS. 2-4, a second embodiment of a second heat exchanger 118 includes a refrigerant tube 166 having multiple coils to increase the contact area with the liquid coolant. The second heat exchanger 118 includes a circular enclosure 144 having a top 144a, side 144b, and a bottom 144c. A first interior surface 162 of the enclosure 144 defines a first interior volume 164. The first interior surface 162 is circular to conform to the shape of the enclosure 144. In that regard, the enclosure 144 may be square (or rectangular), and the first interior surface 162 could include four sides. Other shapes for the enclosure 144 are possible. The top of the enclosure 144a may be designed as a removable cover with hinges, clasps or the like (not shown) so that the internal components may be removed for cleaning.

**[0038]** The second heat exchanger 118 further includes a baffle 176 disposed within the enclosure 144. The baffle 176 separates a second interior volume 178 from the remainder of the first interior volume 164. The baffle 176 includes at least one aperture 180 fluidly coupling the second interior volume 178 to the remaining portion of the first interior volume 164. The baffle 176 is constructed so as to secure against leakage or fluid passage from one volume to the other, thereby limiting any fluid communication between the two volumes solely to the aperture 180. In the illustrated example, the baffle 176 is cylindrical and concentrically disposed within the enclosure 144. The cylindrical baffle 176 sealingly abuts the top of the enclosure 144a, and is open at the opposing end, towards the bottom of the enclosure 144c. In this embodiment, the opening creates the aperture 180.

**[0039]** The second heat exchanger 118 further includes a first coolant port 150, which in the illustrated embodiment is the liquid coolant inlet port. The first coolant port 150 may be adapted with a fitting 154 to sealingly secure a coolant inlet conduit 156 to the top of the enclosure 144a. A second coolant port 152 provides a discharge means for the coolant exiting the second heat exchanger 118. A fitting, such as the fitting 154, secures to the top of the enclosure 144a and provides a mating connection for a coolant discharge conduit 160. In the disclosed example, the fitting 154 is illustrated as a common fitting for both the first coolant port 150 and the second coolant port 152. However, a separate fitting may be used. The coolant at the discharge may be liquid, vapor, or a mixture of both.

**[0040]** The aforementioned continuous coolant tube 166 is disposed within the enclosure 144 so as to increase the heat transfer between the surface of the refrigerant tube and the coolant flowing within the enclosure. In the embodiment shown, a first end 168 of the tube 166 passes through a first refrigerant tube opening 146 into the first interior volume 164. The first interior volume 164 is that volumetric portion that has not been supplanted by the volumetric portion of second interior volume 178. The tube 166 is coiled around the outer wall of the baffle 176 in a downward-spiraling manner. At the bottom of the enclosure 144<sub>c</sub>, the tube 166 is formed so as to produce a tighter or smaller diameter coil. The tube 166 then coils in an upwardly-spiraling manner within the second interior volume 178. At the top of the enclosure 144<sub>a</sub>, the tube 166 exits the enclosure 144 through a second refrigerant tube opening 148.

**[0041]** Other means of increasing the heat transfer between the refrigerant and the coolant are possible. For example, the velocity of the liquid coolant flow (e.g., grey water) through the enclosure 144 may be increased by the pump 20 (FIG. 1). Also, at least one flow turbulator 184 may be positioned within the coolant flow path. The flow turbulators increase turbulence of the liquid coolant to increase the heat transfer between the coolant and the refrigerant tube.

**[0042]** Turning now to FIGS. 6A and 6B of the drawings, wherein like numerals indicate like elements from FIGS. 2-4, a third embodiment of a second heat exchanger 218 includes an enclosure 244 that is circular in shape and forms a tube. The second heat exchanger 218 further includes a refrigerant tube 266 disposed within an interior volume 264 of the enclosure 244. The enclosure 244 may be formed in a coil, for example, with the refrigerant tube 266 disposed essentially concentric therein. One or more centering elements 288 may be spaced every few inches along the length of the enclosure 244 within the interior volume 264 to provide positioning for the refrigerant tube 266 therein. In the example shown in FIG. 6B, the centering elements 288, 288 are formed by crimping the enclosure 244 into contact with the refrigerant tube 266. Other embodiments are contemplated. For example, a plurality spacers (not shown) may be positioned within the interior volume 264 to support the refrigerant tube 266. Although not shown, the enclosure 244 and refrigerant tube 266 include openings on either end of the coil for connecting coolant and refrigerant lines. The refrigerant tube 266 may also include one or more heat transfer elements 286 to increase the heat

exchange relationship between the refrigerant in the tube and the coolant flowing within the interior volume 264. In the embodiment shown, the heat transfer elements 286 comprise a plurality of spokes fixed within the refrigerant tube 266. The spokes aid in transferring heat from the refrigerant to the wall of the tube 266.

**[0043]** Referring now to FIGS. 1, 2, and 5, in operation the second heat exchanger 18 may be activated when the first heat exchanger 14 does not remove sufficient heat from the refrigerant vapor. In one embodiment, the coolant inlet conduit 56 and the coolant discharge conduit 60 are coupled via quick connect couplings to the grey water network. The pump 20 continuously pumps cooling water through the grey water network, and a manual valve (not shown) may be opened or closed upon need to flow the coolant through the enclosure 44. In another embodiment, the pump 20 may be coupled to the second heat exchanger 18 and is isolated from the grey water network, activating upon a command from the controller 950. In one example, if the signal from the ambient air temperature (AAT) sensor rises above a threshold value, the controller 950 commands the pump 20 to operate. Likewise, when the signal from the AAT sensor falls below a threshold value, the controller 950 commands the pump 20 to cease operation. The refrigerant may flow continuously through the refrigerant tube 66, or may be bypassed (not shown) around the second heat exchanger 18 when the pump 20 is inoperative.

**[0044]** In one example, the fan 16 is inoperable due to constraints within the cargo ship transporting the refrigerant vapor compression system 10. The refrigerant vapor enters the first heat exchanger 14 and exits through conduit 15. The value of the ambient air temperature (AAT) is approximately 50 °C. The temperature of the refrigerant vapor exiting the first heat exchanger 14 is approximately 55 °C, which is not sufficiently cooled. Consequently, either a crew member opens the valve to the coolant inlet conduit 56 or the controller 950 activates the pump 20 responsive to a temperature sensor value. The pump 20 pumps the liquid coolant 21 through the coolant inlet conduit 56, while the coolant flows freely within the first interior volume 64, thereby effecting a heat exchange with the vaporous refrigerant flowing in the refrigerant tube 66. The liquid coolant 21 enters the first coolant port 50 at a temperature of 30 °C and a flow rate of 0.25 kg/sec. The CO<sub>2</sub> in the refrigerant tube 66 enters the first refrigerant tube opening 46 at a temperature of 120 °C, a pressure of 12,400 kPa, and a flow rate of 0.13 kg/sec. The CO<sub>2</sub> exits the second refrigerant tube opening

48 at a temperature of 35 °C and a pressure of 12,000 kPa, and the liquid coolant 21 temperature rises to approximately 60 °C (or rises approximately 30 °C).

**[0045]** The particular arrangement of the refrigerant tube 66 within the enclosure 44 allows for design possibilities not available when the liquid-cooled heat exchanger is a pressure vessel. The inventor of the present disclosure has recognized that certain advantages in manufacturing cost and operational efficiencies may be realized by an alternate construction. For example, the sea water flowing through the enclosure 44 does not need to be pressurized any more than the pump head required for proper flow rate, approximately 10 pounds per square inch. Also, the usual concerns with exposure to high temperatures are alleviated since the enclosure 44 is flooded in liquid coolant 21. As such, the enclosure 44 is not treated as a pressure vessel and may be constructed of lightweight materials. In one example, the enclosure 44 is constructed of rigid plastic or polycarbonate. Plastics and the like are non-corrosive, very inexpensive, and may be easily molded to fit the particular needs of the design. For example, the enclosure 44 illustrated in FIG. 2 may be formed in a two-piece clamshell arrangement. After separately forming the two halves, the refrigerant tube 66 may be set into one half, the second half laid over top, and molded into place using heat and pressure. An integral assembly is formed that requires no separate seal rings and the like.

**[0046]** Another advantage of the present disclosure is that the parasitic power required to operate pump 20 is less than other pumps in the prior art. This is because the pump is not required to operate at high pressure: there is very little pressure drop on the liquid coolant, as compared to flowing the coolant through long lengths of small tubing.

**[0047]** Further advantages exist. For example, the low profile design illustrated in FIG. 2 allows the heat exchanger to be positioned in spaces that were previously too small to accommodate the bulkier prior art heat exchangers.

**[0048]** While the present disclosure has been particularly shown and described with reference to the preferred mode as illustrated in the drawing, it will be understood by one skilled in the art that various changes in detail may be effected therein without departing from the spirit and scope of the disclosure as defined by the claims.

**We Claim:**

1. A refrigerant vapor compression system comprising:
  - a compressor for compressing a refrigerant, the compressor having a suction port and a discharge port;
  - an air-cooled heat exchanger operatively coupled to the discharge port;
  - a fan disposed proximate to the air-cooled condenser heat exchanger unit;
  - a liquid-cooled heat exchanger operatively coupled to the air-cooled heat exchanger, the liquid-cooled heat exchanger comprising an enclosure defining an interior volume, the enclosure having a liquid coolant inlet port and a liquid coolant discharge port fluidly coupled to the interior volume, and a continuous refrigerant tube sealingly disposed within the enclosure, the refrigerant tube fluidly isolated from and in heat exchange relationship with the interior volume;
  - a liquid coolant inlet conduit operatively coupled to the liquid coolant inlet port of the liquid-cooled heat exchanger;
  - a coolant pump operatively coupled to the liquid coolant inlet conduit for pumping a liquid coolant;
  - a liquid coolant discharge conduit operatively coupled to the liquid coolant discharge port of the liquid-cooled heat exchanger;
  - an evaporator heat exchanger unit operatively coupled to the liquid-cooled heat exchanger and the suction port;
  - an evaporator fan disposed proximate to the evaporator heat exchanger unit; and
  - a controller operatively associated with the coolant pump for controlling the flow of liquid coolant into the liquid-cooled heat exchanger.
2. The refrigerant vapor compression system of claim 1 wherein the refrigerant is carbon dioxide.
3. The refrigerant vapor compression system of claim 1 wherein the liquid coolant is water.

4. The refrigerant vapor compression system of claim 3 wherein the refrigerant vapor compression system is located on a cargo ship, and the liquid coolant is grey water.
5. The refrigerant vapor compression system of claim 4 wherein the controller is manual, and the flow of liquid coolant into the liquid-cooled heat exchanger is controlled by a hand-operated valve.
6. The refrigerant vapor compression system of claim 1 wherein the controller controls the coolant pump.
7. The refrigerant vapor compression system of claim 6 wherein the controller controls the coolant pump responsive to a parameter within the refrigerant vapor compression system.
8. The refrigerant vapor compression system of claim 7 wherein the parameter is ambient air temperature.
9. A liquid-cooled heat exchanger for use in a refrigerant vapor compression system, the liquid-cooled heat exchanger comprising:
  - an enclosure comprising a first interior surface defining a first interior volume, the enclosure further comprising a plurality of openings being fluidly coupled to the first interior volume, the plurality of openings comprising a first refrigerant tube opening, a second refrigerant tube opening, a first coolant port adapted to flow a liquid coolant into the enclosure, and a second coolant port adapted to flow the coolant out of the enclosure;
  - a first fitting sealingly engaged to the first coolant port, the fitting adapted to couple with a coolant inlet conduit;
  - a second fitting sealingly engaged to the second coolant port, the fitting adapted to couple with a coolant discharge conduit; and
  - a continuous refrigerant tube having a first end, a second end, and a first length disposed therebetween, the first end passing through and in sealing engagement with the first refrigerant tube opening, the second end passing through and in sealing engagement with the second refrigerant tube opening, the first length disposed in serpentine fashion within the first interior volume.
10. The heat exchanger of claim 9 wherein the enclosure is a low-profile enclosure.

11. The heat exchanger of claim 10 wherein the enclosure defines an aspect ratio, the aspect ratio being greater than 5:1.
12. The heat exchanger of claim 10 further comprising a rib coupled to an inner surface of the enclosure for securing the refrigerant tube and channeling a flow of coolant within the interior surface.
13. The heat exchanger of claim 9 wherein the enclosure is a coiled tube.
14. The heat exchanger of claim 13 further comprising a heat transfer element disposed within the coiled tube.
15. The heat exchanger of claim 13 further comprising a centering element disposed within the first interior volume.
16. The heat exchanger of claim 14 wherein the centering element is a crimp.
17. The heat exchanger of claim 9 further comprising a baffle disposed within the enclosure, the baffle defining a second interior volume within the first interior volume, the baffle having at least one aperture fluidly coupling the second interior volume to a remaining portion of the first interior volume.
18. The heat exchanger of claim 17 wherein the continuous refrigerant tube further comprises a second length disposed within the second interior volume.
19. The heat exchanger of claim 18 the first length and the second length are helical.
20. The heat exchanger of claim 19 wherein the first length of the refrigerant tube is spiraled helically in a direction opposite to the second length.
21. The heat exchanger of claim 9 wherein the enclosure is cylindrical.
22. The heat exchanger of claim 9 wherein the enclosure further comprises a removable cover, the refrigerant tube being removable from the enclosure when the cover is removed.

23. The heat exchanger of claim 9 further comprising a flow turbulator disposed within the first interior volume, the flow turbulator for creating turbulent flow to increase a heat transfer characteristic between the liquid coolant and the refrigerant tube.
24. The heat exchanger of claim 9 wherein the continuous refrigerant tube branches and converges.
25. A method for liquid-cooling a vapor refrigerant in a refrigerant vapor compression system, the method comprising the steps of:
- providing an air-cooled heat exchanger;
  - providing a liquid-cooled heat exchanger coupled to the air-cooled heat exchanger, the liquid-cooled heat exchanger comprising an enclosure having a first interior volume;
  - providing in serpentine fashion a continuous refrigerant tube extending through the first volume of the liquid-cooled heat exchanger;
  - flowing a refrigerant through the air-cooled heat exchanger and, if the refrigerant is not sufficiently cooled,
  - simultaneously flowing the refrigerant through the continuous refrigerant tube in the liquid-cooled heat exchanger and flowing a liquid coolant through the first interior volume of the liquid-cooled heat exchanger.
26. The method of claim 25, further comprising the step of providing a baffle within the first interior volume, the baffle defining a second interior volume within the first interior volume, the baffle comprising an aperture, the continuous refrigerant tube extending through the first volume of the liquid-cooled heat exchanger, through the aperture, and through the second interior volume of the liquid-cooled heat exchanger.
27. The method of claim 26 wherein the refrigerant is passed helically through the first interior volume and the second interior volume.
28. The method of claim 27 wherein a direction of the helical path through the first interior volume and the second interior volume is opposite to a direction of the liquid coolant flow.

29. The method of claim 25, further comprising the step of providing a controller, the controller controlling the refrigerant flow through the liquid-cooled heat exchanger responsive to a parameter on the air-cooled heat exchanger.
30. The method of claim 29, wherein the parameter is ambient air temperature.
31. The method of claim 29, wherein the controller is manual.
32. The method of claim 25, wherein the liquid-cooled heat exchanger is in serial relationship to the air-cooled heat exchanger.