

Transport Refrigeration System and Method of Operation

Technical Field

[0001] This invention relates generally to transport refrigeration systems and, more particularly, to capacity modulation in a refrigerant vapor compression system operating in a transcritical cycle.

Background of the Invention

[0002] Refrigerant vapor compression systems are well known in the art and commonly used for conditioning air to be supplied to a climate controlled comfort zone within a residence, office building, hospital, school, restaurant or other facility. Refrigerant vapor compression systems are also commonly used in transport refrigeration systems for refrigerating air supplied to a temperature controlled cargo space of a truck, trailer, container or the like for transporting perishable items. Traditionally, most of these refrigerant vapor compression systems operate at subcritical refrigerant pressures and typically include a compressor, a condenser, and an evaporator, and expansion device, commonly an expansion valve, disposed upstream, with respect to refrigerant flow, of the evaporator and downstream of the condenser. These basic refrigerant system components are interconnected by refrigerant lines in a closed refrigerant circuit, arranged in accord with known refrigerant vapor compression cycles, and operated in the subcritical pressure range for the particular refrigerant in use. Refrigerant vapor compression systems operating in the subcritical range are commonly charged with conventional fluorocarbon refrigerants such as, but not limited to, hydrochlorofluorocarbons (HCFCs), such as R22, and more commonly hydrofluorocarbons (HFCs), such as R134a, R410A and R407C.

[0003] In today's market, greater interest is being shown in "natural" refrigerants, such as carbon dioxide, for use in air conditioning and transport refrigeration systems instead of HFC refrigerants. However, because carbon dioxide has a low critical temperature, most refrigerant vapor compression systems charged with carbon dioxide as the refrigerant are designed for operation in the transcritical pressure regime. For example, transport refrigerant vapor compression systems

having an air cooled refrigerant heat rejection heat exchanger operating in environments having ambient air temperatures in excess of the critical temperature point of carbon dioxide, 31.1°C (87.8°F), must also operate at a compressor discharge pressure in excess of the critical point pressure for carbon dioxide, 7.38 MPa (1070 psia) and will operate in a transcritical cycle. In refrigerant vapor compression systems operating in a transcritical cycle, the refrigerant heat rejection heat exchanger operates as a gas cooler rather than a condenser and operates at a refrigerant temperature and pressure in excess of the refrigerant's critical point, while the evaporator operates at a refrigerant temperature and pressure in the subcritical range.

[0004] In order to optimize the capacity or efficiency of such a system using carbon dioxide as the refrigerant and operating under transcritical conditions, it is desirable to increase the high pressure of the system (gas cooler pressure) to thereby lower the specific enthalpy entering the evaporator, and thereby increase capacity. This, in turn, necessitates the increase of the pressure ratio at the compressor.

[0005] Transport refrigeration systems, such as those used in trucks/trailers and refrigerated containers, operate in very unpredictable environments. Temperature and humidity may vary widely at different times of day and over different seasons throughout the year. Also the product load may vary dramatically and in unpredictable manner. The capacity of the system must be designed for the harshest conditions (i.e. pull down at high ambient temperatures, for example), but be able to operate efficiently at less stringent conditions such as under part load.

[0006] One approach to selectively varying the capacity of a compressor is that shown in US 5, 471,120 wherein the duration of the loading and unloading periods are modulated in a time pulsed matter in order to maximize the efficiency of the overall system. This is accomplished by either modulating the relatively axially movement between the scroll members so as to form a leakage path across the wrap tips and opposed end plates or by reducing the orbital radius of one of the scroll members to thereby form a leakage path across the flank surfaces of the wrap. While such a so called "digital scroll compressor" has been implemented primarily for use in residential air conditioning systems operating in a subcritical condition, it has not been applied to transport refrigeration systems operating in a transcritical

cycle. The primary reason for this is that such a digital scroll compressor is not capable of operating for prolonged periods of time under conditions of the high pressure ratios that are necessary for the efficient operation in the transcritical range.

[0007] Further, the transportation refrigeration system is characterized by extremely tight temperature control requirements. In perishable part load operating conditions, it results in very low mass flow rate. When a system uses CO₂ as working fluid, the compressor discharge temperature will rise quickly with compression ratio. A large amount of liquid injection has to be used in order to keep the compressor discharge temperature within the operating limit such as 275°F. Under low mass flow rate, when liquid injection exceeds a certain amount, the suction superheat will become so low that it is possible to damage a compressor with liquid flooding. So the challenge is to design a system which can operate with low mass flow rate yet, have its compressor discharge temperature still within a safe limit.

Disclosure of the Invention

[0008] Briefly, in accordance with one aspect of the invention, the refrigerant vapor of a vapor compression system operating in the transcritical range is compressed by way of a digital scroll compressor.

[0009] By yet another aspect of the invention, control methods and apparatus are provided to prevent the digital scroll compressor from becoming overloaded in such a system.

Brief Description of the Drawings

[0010] FIG. 1 is a schematic illustration of the present invention as incorporated into a vapor compression system.

[0011] FIGS. 2A and 2B are graphic illustrations of compressor capacity modulation in accordance with the present invention.

[0012] FIG. 3 is a graphic illustration of the resultant cooling capacity control in accordance with the present invention.

[0013] FIG. 4 is a graphic illustration of the present invention as incorporated into an economized vapor compression system.

[0014] FIG. 5 is a schematic illustration of the present invention as incorporated into another type of economized vapor compression system.

Detailed Description of the Invention

[0015] The invention is shown generally at 10 as incorporated into a refrigerant vapor compression system operating with CO₂ as the refrigerant and having in serial flow relationship, a compressor 11, a refrigerant heat rejecting heat exchanger 12, an expansion device 13 and a refrigerant heat absorbing heat exchanger 14. The refrigerant heat absorbing heat exchanger 14, commonly referred to as an evaporator, includes a motor driven fan 16 for circulating air thereover so as to be cooled.

[0016] Since the system uses CO₂ as the working fluid, it necessarily operates in a transcritical cycle, such that the refrigerant heat rejecting heat exchanger 12, which may be referred to as a condenser, has high pressure refrigerant passing in heat exchange relationship with a cooling medium, most commonly ambient air, in air conditioning systems or transport refrigeration system. In such a refrigerant vapor compression system operating in a transcritical cycle, the refrigerant heat rejecting heat exchanger 12 constitutes a gas cooler heat exchanger through which supercritical refrigerant passes in heat relationship with the cooling medium. The condenser/gas cooler 12 has a fan 17 driven by a multi-speed motor 18 and controlled in a manner to be described hereinafter.

[0017] To accommodate the use of CO₂ as the working fluid operating in a transcritical cycle, there is also included in the circuit, between the condenser/gas cooler 12 and the evaporator 14, a first expansion device 19 and a flash tank receiver 21. As the CO₂ refrigerant leaves the gas cooler 12, it passes through the first expansion device 19 whereby it expands to a lower pressure, with the refrigerant then entering the flash tank receiver 21 as a mixture of liquid refrigerant and vapor. The liquid refrigerant settles in the lower portion of the flash tank receiver 21 and the refrigerant vapor collects in the upper portion of the flash tank receiver 21, above the liquid therein.

[0018] The liquid refrigerant passes from the flash tank receiver 21 to the expansion device 13 where it expands to a lower pressure and temperature before

entering the evaporator 14. The evaporator 14 constitutes a refrigerant evaporating heat exchanger through which expanded refrigerant passes in heat exchanger relationship with the heating fluid with the refrigerant being vaporized and typical superheated. The heating fluid passing in heat exchange relationship with the refrigerant constitutes air to be supplied to a perishable cargo storage zone associated with a transport refrigeration unit. The low pressure refrigerant vapor leaves the evaporator 14 and then returns to the suction port of the compression device 11.

[0019] The compressor 11 comprises a digital scroll compressor such as that described in US Patent 5, 741,120 and commercially available from Copeland Corporation.

[0020] The digital scroll operates in two stages – the “loaded state”, when the compressor operates like a standard scroll and delivers full capacity and mass flow, and the “unloaded state”, when there is no capacity and no mass flow through the compressor. The digital scroll compressor operates under the concept of cycle time. Each cycle time consists of a “loaded state” time and “unloaded state” time. The digital scroll compressor will effectively reduce compression ratio and therefore compressor discharge temperature through the control of cycle time. In the “loaded state”, the compressor operates like a standard scroll and delivers full capacity and mass flow. In the “unloaded state”, there is no capacity and no mass flow through the compressor. The duration of these two-time segments determine the capacity modulation of the compressor. For example, as shown in Fig. 2A, in a 20 second cycle time, if the loaded state time is 10 seconds and the unloaded state time is 10 seconds, the compressor modulation is $(10 \text{ seconds} \times 100\% + 10 \text{ seconds} \times 0\%) / 20 = 50\%$. If for the same cycle time, the loaded state time is 15 seconds and the unloaded state time is 5 seconds, as shown in Fig. 2B the compressor modulation is 75%. The capacity is a time averaged summation of the loaded state and unloaded state capacity. By varying the loaded state time and unloaded state time, any capacity from 10% to 100% can be delivered by the compressor. The refrigeration system capacity can therefore adjust precisely to match load demand over a wide range of applications.

[0021] As discussed hereinabove, such a digital compressor has not been used in systems using CO₂ as the refrigerant and operating in transcritical cycles since such a compressor would normally not be able to withstand the high compression ratio that is prevalent in such a system. However, because of the control features that are provided herein, the system may be operated in such a manner as to maintain the compression ratios at a level which will allow sustained use of such a digital scroll compressor in the system operating under transcritical conditions.

[0022] As described in US Patent 5,741,120, the digital scroll compressor is designed to have its capacity modulated by a control 22 which delivers a variable duty cycle signal S along line 23 to the digital scroll compressor 11 for that purpose. However, in accordance with the present invention, the control 22 is used to implement further control features so as to limit the compression ratio experienced by the digital scroll compressor 11 in a manner to be described hereinafter.

[0023] The drive motor 18 for the fan 17 is a multiple speed motor which can be selectively operated at a relatively high speed or a relatively low speed. Thus, it can be a two speed motor or it may be a variable speed motor which can vary its speed over a continuous range. Control of the motor speed is maintained by way of the control 22 along line 24 in a manner to be described hereinafter.

[0024] A pressure sensor 26 and a temperature sensor 27 are installed at the discharge of the compressor 11, with their respective outputs being sent to the control 22 along lines 28 and 29, respectively. Although the system is shown with both a pressure sensor 26 and a temperature sensor 27, it may operate with either of those and without the use of the other. That is, for control purposes, it is desirable to sense a condition that is indicative of the pressure ratio of the compressor 11, and either the discharge pressure or the discharge temperature can be used for that purpose as will be described hereinafter.

[0025] Another parameter that is used in the control of the system is the temperature within the space 31 being cooled. This temperature is determined by a temperature sensor 32 which sends a sensed temperature signal to a comparator 33 to be compared with a set point, with the difference being sent along line 34 to the control. Considering now the manner in which the capacity of the digital scroll

compressor 11 is controlled, provision is made to vary the control algorithm depending on the particular mode in which the system is operating. Three possible modes of operation are shown in Fig. 3 to include the perishable mode, the frozen mode, and the pull down mode.

[0026] When in the perishable (chill) mode, the controller 22 maintains the supply air temperature at set point, and the compressor 11 is operating at a part load condition. Thus, in order to remove extra capacity from the system, the control 22 first selects a low speed for the gas cooler fan motor 18 and opens valve 19 so as to thereby reduce the compressor discharge pressure. Then the “loaded state” time of the compressor is reduced as much as possible while maintaining the supply air temperature at set point. By reducing compressor discharge pressure, the compressor dome temperature is reduced due to lower compression ratios and, in turn, the compressor reliability is improved. The line A in Fig. 3 shows the manner in which the cooling capacity is varied during operation in this mode.

[0027] While operating in the frozen mode, it is recognized that frozen range cargos are not sensitive to minor temperature changes. Thus, the method of temperature control employed in this range takes advantage of this to greatly improve the energy efficiency of the unit. Temperature control in the frozen range is accomplished by cycling the compressor between the loaded state and unloaded state as the load demand requires. The compressor discharge pressure is at an optimized point through control valve 19, and the fan motor 18 can be operated either at a high or low speed. The line B indicates a typical cooling capacity variation for the frozen mode of operation. In the pull down mode, the compressor 11 is operating in a full load condition with the maximum capacity being required. Thus, the fan motor 18 is run at high speed and a compressor discharge pressure, as indicated by either the pressure sensor 26 or the temperature sensor 27 is maintained at a maximum design point. The control 22 then varies the cycle times of the “loaded state” and “unloaded state” as necessary in order to prevent overloading of the compressor. For example, a maximum compressor discharge temperature may be established at 300°F, and as that temperature is approached, the compressor modulation (i.e. the loaded state time as compared with the unloaded time) is decreased in order to prevent that temperature from being exceeded. The line C in

Fig. 3 shows a typical cooling capacity variation during the pull down mode of operation.

[0028] As shown in Fig. 1 and as described above, the present invention is applicable to a non-economized digital scroll compressor system. However, it is equally applicable to economized systems of various types as shown in Fig. 4 and Fig. 5, for example. In Fig. 4, the control system is shown as used with a flash tank economized system driven by a digital scroll compressor 11 having a vapor injection port 36. Here, the flash tank receiver 21 serves not only as a charge control tank, but also as a flash tank economizer. Vapor refrigerant collecting in the portion of the flash tank receiver 21 above the liquid level therein passes from the receiver 21 along line 37 and solenoid valve 38 to the vapor injection port 36. The solenoid valve 38 is controlled by the control 22 in order to turn on and off the economizer operation.

[0029] In the event that, despite the use of the control system as described hereinabove, the compressor 11 tends to operate at elevated temperatures, it is desirable to provide liquid injection into the vapor injection port 36 or an alternative port for liquid. Accordingly, line 39 and associated solenoid valve 41 is provided for that purposes.

[0030] It should be recognized that, in the Fig. 4 embodiment, economized operation will occur only when the flash tank vapor pressure is greater than the pressure at the vapor injection port 36. Otherwise, the economizer will not be in operation.

[0031] Another type of economized system which the present control method is applicable is shown in Fig. 5. Here, a flash tank is not included but rather an interstage economizer 42 that is simply a brazed plate heat exchanger. Leading into the interstage economizer 42 is solenoid valve 43 and an expansion valve 44. Thus, when the solenoid valve 43 is open, the refrigerant vapor flows from the gas cooler 12, through the solenoid valve 43 and the expansion valve 44 into the interstage economizer 42, with the vapor then being injected into the vapor injection port 36. When the solenoid valve is closed, the refrigerant vapor flows along line 46 to the interstage economizer 42, with no economizer operation occurring. In either

case, a charge storage vessel 45 is provided to serve only the purpose of storing excess charge.

[0032] While the present invention has been particularly shown and described with reference to the preferred mode as illustrated in the drawings, 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 invention as defined by the claims.

We Claim:

1. A vapor compression system of the type having CO₂ as the working fluid so as to require operation in a transcritical cycle, comprising in serial working fluid flow relationship:

a gas cooler for receiving CO₂ vapor at a higher temperature and discharging it a lower temperature;

a charge storage vessel for receiving CO₂ refrigerant from said gas cooler and discharging CO₂ in liquid form;

an expansion valve between said gas cooler and said charge storage vessel;

an expansion valve for receiving liquid CO₂ from said charge storage vessel and discharging a mixture of CO₂ vapor and liquid;

an evaporator for receiving a mixture of CO₂ vapor and liquid and discharging a CO₂ vapor; and

a digital scroll compressor responsive to a variable duty cycle signal to vary its capacity by modulating the respective loaded stated and unloaded state times in a pulse width modulation manner, said compressor receiving low pressure CO₂ vapor from said evaporator and discharging high pressure CO₂ vapor to said gas cooler.

2. A vapor compression system as set forth in claim 1 wherein said charge storage vessel comprises a flash tank adapted to have both CO₂ vapor and a CO₂ liquid therein.

3. A vapor compression system as set forth in claim 1 and including a gas cooler fan and a multi-speed motor capable of driving said fan at either a high or a low speed.

4. A vapor compression system as set forth in claim 3 and including a control for determining whether the vapor compression system is operating under full load or part load operating conditions and for operating said fan motor at either a high speed or at a low speed as a function thereof.

5. A vapor compression system as set forth in claim 4 wherein, if the system is operating under part load conditions, said control operates said fan at a low speed.

6. A vapor compression system as set forth in claim 5 wherein said control is then operable to reduce the loaded state time of operation.

7. A vapor compression system as set forth in claim 4 wherein if the system is operating under full load operating conditions said control is adapted to operate said fan at a high speed.

8. A vapor compression system as set forth in claim 7 wherein said control is then operable to reduce the loaded state time of operation as necessary to prevent overloading of the compressor.

9. A vapor compression system as set forth in claim 8 and including a sensor for obtaining an indication of the pressure ratio under which said compressor is operating and sending a representative signal to said control.

10. A method of controlling a refrigerant vapor compression system of the type having a gas cooler with a multi-speed fan and being designed to operate in a transcritical cycle, comprising the steps of:

providing a digital scroll compressor having the capability to vary its capacity by modulating the respective loaded state and unloaded state times in a pulse width modulation manner; and

during periods of part load operating conditions, operating the fan at a low speed and reducing the loaded state time of operation for the digital scroll compressor.

11. A method as set forth in claim 10 and including the steps of:

providing in serial flow relationship downstream of said gas cooler an expansion valve and a charge storage vessel; and

sensing a condition indicative of compressor pressure ratio and varying said expansion valve to control said pressure ratio.

12. A method of controlling a refrigerant vapor compression system of the type having a gas cooler with a multi-speed fan and being designed to operate in a transcritical cycle, comprising the steps of:

providing a digital scroll compressor having the capability to vary its capacity by modulating the respective loaded state and unloaded state times in a pulse width modulation manner;

during periods of full load operation conditions, operating the fan at a high speed; and

reducing the loaded state time of operation as necessary in order to prevent overloading of the compressor.

13. A method as set forth in claim 12 and including the steps of:

providing in serial flow relationship downstream of said gas cooler an expansion valve and a charge storage vessel; and

sensing a condition indicative of pressure ratio of the compressor and varying the expansion valve to control said pressure ratio.