(19) DANMARK

(10) **DK/EP 2491318 T3**

Patent- og Varemærkestyrelsen

(12) # Oversættelse af europæisk patentskrift

(51) Int.Cl.: *F 25 B 1/10 (2006.01)* *F 25 B 9/00 (2006.01)* *F 25 B 41/04 (2006.01)* *F 25 B 49/00 (2006.01)* *F 25 B 49/02 (2006.01)*

(45) Oversættelsen bekendtgjort den: **2018-06-25**

(80) Dato for Den Europæiske Patentmyndigheds bekendtgørelse om meddelelse af patentet: **2018-05-16**

(86) Europæisk ansøgning nr.: **10771264.8**

(86) Europæisk indleveringsdag: **2010-10-12**

(87) Den europæiske ansøgnings publiceringsdag: **2012-08-29**

(86) International ansøgning nr.: **US2010052267**

(87) Internationalt publikationsnr.: **WO2011049778**

(30) Prioritet: **2009-10-23 US 254280 P**

(84) Designerede stater: **AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR**

(73) Patenthaver: **Carrier Corporation, One Carrier Place, Farmington CT 06034, USA**

(72) Opfinder: **LIU, Lucy, Yi, 5003 Saddlebrook Drive, Fayetteville, New York 13066, USA**
DURAISAMY, Suresh, 4046 Plum Yew Circle, Liverpool, New York 13090, USA
HOFSDAL, Gilbert, B., 2381 Old Route 5, Chittenango, New York 13037, USA
HUFF, Hans-Joachim, Taunusstrasse 5, 55118 Mainz, Tyskland

(74) Fuldmægtig i Danmark: **Plougmann Vingtoft A/S, Rued Langgaards Vej 8, 2300 København S, Danmark**

(54) Benævnelse: **PARAMETERSTYRING I TRANSPORTKØLESYSTEM OG FREMGANGSMÅDER HERTIL**

(56) Fremdragne publikationer:
EP-A2- 0 217 558
EP-A2- 1 965 158
WO-A1-2009/038624
US-A1- 2006 075 771
US-A1- 2009 094 998

DESCRIPTION

TECHNICAL FIELD

[0001] This invention relates generally to transport refrigeration systems and methods for same and, more particularly, to methods and apparatus for controlling vapor compression systems.

BACKGROUND OF THE INVENTION

[0002] EP 1965158 A1 discloses a refrigerant vapor compression system in accordance with the preamble of claim 1.

[0003] A particular difficulty of transporting perishable items is that such items must be maintained within a temperature range to reduce or prevent, depending on the items, spoilage or conversely damage from freezing. A transport refrigeration unit is used to maintain proper temperatures within a transport cargo space. The transport refrigeration unit can be under the direction of a controller. The controller ensures that the transport refrigeration unit maintains a certain environment (e.g. thermal environment) within the transport cargo space. The controller can operate a transport refrigeration system and/or components thereof responsive to sensors disposed in the system.

[0004] A vapor compression system can include a compressor, a heat rejection heat exchanger (e.g., condenser or gas cooler), an expansion device, and an evaporator. Economizer cycles are sometimes employed to increase the efficiency and/or capacity of the system. Economizer cycles operate by expanding the refrigerant leaving the heat rejecting heat exchanger to an intermediate pressure and separating the refrigerant flow into two streams. One stream is sent to the heat absorbing heat exchanger, and the other is sent to cool the flow between two compression stages. In one form of an economizer cycle, a flash tank is used to perform the separation. In an economizer cycle with flash tank, a refrigerant discharged from the gas cooler passes through a first expansion device, and its pressure is reduced. Refrigerant collects in the flash tank as part liquid and part vapor. The vapor refrigerant is used to cool refrigerant exhaust as it exits a first compression device, and the liquid refrigerant is further expanded by a second expansion device before entering the evaporator. Such a flash tank economizer is particularly useful when operating in transcritical conditions, such as are required when carbon dioxide is used as the working fluid.

[0005] Due to the thermophysical properties of CO_2, the refrigeration system can operate in both the subcritical and transcritical modes. The subcritical mode is similar to the operation of systems with conventional refrigerants. In the transcritical mode the refrigerant pressure in the heat rejection heat exchanger, and possibly in the flash tank, is above the critical pressure, while the evaporator operates as in the subcritical mode.

DISCLOSURE OF THE INVENTION

[0006] In view of the background, at least preferred embodiments of the present invention provide a transport refrigeration system, transport refrigeration unit, and methods of operating the same that can maintain cargo quality by selectively controlling transport refrigeration system components.

[0007] One embodiment can include a control module for a transport refrigeration system. The control module includes a controller for controlling the transport refrigeration system to selectively verify operations of components thereof.

[0008] In at least preferred embodiments of the invention, operations of components of a transport refrigeration system can be directly measured (e.g., sensors) and/or indirectly verified (e.g., without sensors).

[0009] In accordance with one embodiment of the invention, an economizer includes a control for controlling operations of the economizer responsive to pressure in a compressor.

[0010] In accordance with a first aspect of the present invention, there is provided a refrigerant vapor compression system comprising: a refrigerant compression device; a refrigerant heat rejection heat exchanger downstream of said compression device; a refrigerant heat absorption heat exchanger downstream of said refrigerant heat rejection heat exchanger; a first expansion device disposed downstream of said refrigerant heat rejection heat exchanger and upstream of said refrigerant heat absorption heat exchanger, a sensor coupled to an output of the heat rejection heat exchanger, the sensor to measure a refrigerant temperature; and a controller to control operations of the refrigeration vapor compression system, said controller operative to indirectly verify the measured refrigerant temperature, characterized in that the compression device includes a first compression stage and a second compression stage, and in that the refrigerant temperature at the output of the heat rejection heat exchanger is first determined by measurement using the sensor, and is second determined by calculation using ambient temperature and vapor compression system capacity.

[0011] In accordance with a second aspect of the present invention, there is provided a method for determining a characteristic of a refrigerant vapor compression system having a refrigerant circuit including a refrigerant compression device, a refrigerant heat rejection heat exchanger downstream of said compression device, a refrigerant heat absorption heat exchanger downstream of said refrigerant heat rejection heat exchanger, a sensor to sense the characteristic to determine a system capacity of the refrigerant vapor compression system during operation, said characteristic being a refrigerant temperature at an output of the heat rejection heat exchanger, and interconnecting refrigerant lines as active components, the method comprising: operating the refrigerant vapor compression system in a mode where the refrigerant is circulating within the active components of the refrigerant circuit; indirectly determining the refrigerant temperature at the output of the heat rejection heat exchanger using ambient temperature and vapor compression system capacity; comparing the sensed value of the refrigerant temperature at the output of the heat rejection heat exchanger against said indirectly determined value of refrigerant temperature at the output of the heat rejection heat exchanger; and determining an error condition of a corresponding sensor when a result of the comparison does not match.

[0012] In accordance with a third aspect of the present invention, there is provided a computer program product comprising a computer usable storage medium to store a computer readable program that, when executed on a computer, causes the computer to perform operations to operate a transport refrigeration unit, the operations comprising: operating the transport refrigeration unit in a mode where a refrigerant is circulating within a refrigerant circuit; sensing a refrigerant temperature at an output of a heat rejection heat exchanger to determine a system capacity of the transport refrigeration unit during operation; indirectly determining the refrigerant temperature at the output of the heat rejection heat exchanger using ambient temperature and vapor compression system capacity; comparing the sensed value of the refrigerant temperature at the output of the heat rejection heat exchanger against said indirectly determined value; and determining an error condition of a corresponding sensor when a result of the comparison does not match.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013]

FIG. 1 is a diagram that shows an embodiment of a transport refrigeration system according to the application;

FIG. 2 is a diagram that shows another embodiment of a transport refrigeration system according to the application;

FIG. 3 is a schematic illustration of an embodiment of a vapor compression system according to the application;

FIG. 4 is a diagram graphically showing exemplary refrigerant temperature exiting a heat rejection heat exchanger as a function of system capacity;

FIG. 5 is a diagram graphically showing exemplary compressor mid-stage pressure as a function of compressor discharge pressure for various compressor suction pressures according to embodiments of the application; and

FIG. 6 is a flow diagram showing an embodiment of a method for operating a transport refrigeration system according to the application.

DESCRIPTION OF EXEMPLARY EMBODIMENTS

[0014] Reference will now be made in detail to exemplary embodiments of the application, examples of which are illustrated in the accompanying drawings. Whenever possible, the same reference numerals will be used throughout the drawings to refer to the same or like parts.

[0015] FIG. 1 is a diagram that shows an embodiment of a transport refrigeration system. As shown in FIG. 1, transport refrigeration system 100 can include a transport refrigeration unit 10 coupled to an enclosed space within a container 12. The transport refrigeration system 100 may be of the type commonly employed on refrigerated trailers. As shown in FIG. 1, the transport refrigeration unit 10 is configured to maintain a prescribed thermal environment within the container 12 *(e.g.,* cargo in an enclosed volume).

[0016] In FIG. 1, the transport refrigeration unit 10 is connected at one end of the container 12. Alternatively, the transport refrigeration unit 10 can be coupled to a prescribed position on a side or more than one side of the container 12. In one embodiment, a plurality of transport refrigeration units can be coupled to a single container 12. Alternatively, a single transport refrigeration unit 10 can be coupled to a plurality of containers 12 or multiple enclosed spaces within a single container. The transport refrigeration unit 10 can operate to induct air at a first temperature and to exhaust air at a second temperature. In one embodiment, the exhaust air from the transport refrigeration unit 10 will be warmer than the inducted air such that the transport refrigeration unit 10 is employed to warm the air in the container 12. In one embodiment, the exhaust air from the transport refrigeration unit 10 will be cooler than the inducted air such that the transport refrigeration unit 10 is employed to cool the air in the container 12. The transport refrigeration unit 10 can induct air from the container 12 having a return temperature Tr (*e.g.*, first temperature) and exhaust air to the container 12 having a supply temperature Ts *(e.g.,* second temperature).

[0017] In one embodiment, the transport refrigeration unit 10 can include one or more temperature sensors to continuously or repeatedly monitor the return temperature Tr and/or the supply temperature Ts. As shown in FIG. 1, a first temperature sensor 24 of the transport refrigeration unit 10 can provide the supply temperature Ts and a second temperature sensor 22 of the transport refrigeration unit 10 can provide the return temperature Tr to the transport refrigeration unit 10, respectively. Alternatively, the supply temperature Ts and the return temperature Tr can be determined using remote sensors.

[0018] A transport refrigeration system 100 can provide air with controlled temperature, humidity or/and species concentration into an enclosed chamber where cargo is stored such as in container 12. As known to one skilled in the art, the transport refrigeration system 100 (e.g., controller 250) is capable of controlling a plurality of the environmental parameters or all the environmental parameters within corresponding ranges with a great deal of variety of cargos and under all types of ambient conditions.

[0019] FIG. 2 is a diagram that shows an embodiment of a transport refrigeration system. As shown in FIG. 2, a transport refrigeration system 200 can include a transport refrigeration unit 210 coupled to a container 212, which can be used with a trailer, an intermodal container, a train railcar, a ship or the like, used for the transportation or storage of goods requiring a temperature controlled environment, such as, for example foodstuffs and medicines (e.g., perishable or frozen). The container 212 can include an enclosed volume 214 for the transport/storage of such goods. The enclosed volume 214 may be an enclosed space having an interior atmosphere isolated from the outside (*e.g.*, ambient atmosphere or conditions) of the container 212.

[0020] The transport refrigeration unit 210 is located so as to maintain the temperature of the enclosed volume 214 of the container 212 within a predefined temperature range. In one embodiment, the transport refrigeration unit 210 can include a compressor 218, a condenser heat exchanger unit 222, a condenser fan 224, an evaporation heat exchanger unit 226, an evaporation fan 228, and a controller 250. Alternatively, the condenser 222 can be implemented as a gas cooler.

[0021] The compressor 218 can be powered by single phase electric power, three phase electrical power, and/or a diesel engine and can, for example, operate at a constant speed. The compressor 218 may be a scroll compressor, a rotary compressor, a reciprocal compressor, or the like. The transport refrigeration system 200 requires electrical power from, and can be connected to a power supply unit (not shown) such as a standard commercial power service, an external power generation system (e.g., shipboard), a generator *(e.g.,* diesel generator), or the like.

[0022] The condenser heat exchanger unit 222 can be operatively coupled to a discharge port of the compressor 218. The evaporator heat exchanger unit 226 can be operatively coupled to an input port of the compressor 218. An expansion valve 230 can be connected between an output of the condenser heat exchanger unit 222 and an input of the evaporator heat exchanger unit 226.

[0023] The condenser fan 224 can be positioned to direct an air stream onto the condenser heat exchanger unit 222. The air stream from the condenser fan 224 can allow heat to be removed from the coolant circulating within the condenser heat exchanger unit 222.

[0024] The evaporator fan 228 can be positioned to direct an air stream onto the evaporation heat exchanger unit 226. The evaporator fan 228 can be located and ducted so as to circulate the air contained within the enclosed volume 214 of the container 212. In one embodiment, the evaporator fan 230 can direct the stream of air across the surface of the evaporator heat exchanger unit 226. Heat can thereby be removed from the air, and the reduced temperature air can be circulated within the enclosed volume 214 of the container 212 to lower the temperature of the enclosed volume 214.

[0025] The controller 250 such as, for example, a MicroLink.TM 2i or Advanced controller, can be electrically connected to the compressor 218, the condenser fan 224, and/or the evaporator fan 228. The controller 250 can be configured to operate the transport refrigeration unit 210 to maintain a predetermined environment (e.g., thermal environment) within the enclosed volume 214 of the container 212. The controller 250 can maintain the predetermined environment by selectively controlling operations of the condenser fan 224, and/or the evaporator fan 228 to operate at a low speed or a high speed. For example, if increased cooling of the enclosed volume 214 is required, the controller 250 can increase electrical power to the compressor 218, the condenser fan 224, and the evaporator fan 228. In one embodiment, an economy mode of operation of the transport refrigeration unit 210 can be controlled by the controller 250. In another embodiment, variable speeds of components of the transport refrigeration unit 210 can be adjusted by the controller 250. In another embodiment, a full cooling mode for components of the transport refrigeration unit 210 can be controlled by the controller 250. In one embodiment, the electronic controller 250 can adjust a flow of coolant supplied to the compressor 218.

[0026] FIG. 3 is a diagram that shows an embodiment of a vapor compression system according to the application. As shown in FIG. 3, an exemplary embodiment of a refrigerant vapor compression system 300 designed for operation in a transcritical cycle with a low critical point refrigerant, such as for example, but not limited to, carbon dioxide and refrigerant mixtures containing carbon dioxide. However, it is to be understood that the refrigerant vapor compression system 300 may also be operated in a subcritical cycle with a higher critical point refrigerant such as conventional hydrochlorofluorocarbon and hydrofluorocarbon refrigerants.

[0027] The refrigerant vapor compression system 300 is particularly suitable for use in a transport refrigeration system for refrigerating the air or other gaseous atmosphere within the temperature controlled enclosed volume 214 such as a cargo space of a truck, trailer, container, or the like for transporting perishable/frozen goods. The refrigerant vapor compression system 300 is also suitable for use in conditioning air to be supplied to a climate controlled comfort zone within a residence, office building, hospital, school, restaurant or other facility. The refrigerant vapor compression system could also be employed in refrigerating air supplied to display cases, merchandisers, freezer cabinets, cold rooms or other perishable/frozen product storage areas in commercial establishments.

[0028] The refrigerant vapor compression system 300 includes a multi-stage compression device 320, a refrigerant heat rejection heat exchanger 330, a refrigerant heat absorption heat exchanger 350, also referred to herein as an evaporator, and a primary expansion valve 355, such as for example an electronic expansion valve as depicted in FIG. 3, operatively associated with the evaporators 350, with refrigerant lines 302, 304, and 306 connecting the aforementioned components in a primary refrigerant circuit. As depicted in FIG. 3, the refrigerant vapor compression system 300 may also include an unload bypass line 316 that establishes refrigerant flow communication between an intermediate pressure stage of the multi-stage compression device 320 and the suction pressure portion of the refrigerant circuit, which constitutes refrigerant line 306 extending from the outlet of the evaporator 350 to the inlet of the compression device 320.

[0029] Additionally, the refrigerant vapor compression system 300 can include an economizer circuit having an economizer device 340, a secondary expansion valve 345 and a refrigerant vapor injection line 314. As shown in FIG. 3, the economizer circuit includes a flash tank economizer 340 interdisposed in refrigerant line 304 of the primary refrigerant circuit downstream with respect to refrigerant flow of the refrigerant heat rejection heat exchanger 330 and upstream with respect to refrigerant flow of the refrigerant heat absorption heat exchanger 350. The secondary expansion device 345 is interdisposed in refrigerant line 304 in operative association with and upstream of the economizer. The secondary expansion device 345 may be an expansion valve, such as a high pressure electronic expansion valve as depicted in FIG. 3. Refrigerant traversing the secondary expansion device 345 is expanded to a lower pressure sufficient to establish a mixture of refrigerant in a vapor state and refrigerant in a liquid state. The flash tank economizer 340 includes a separation chamber 342 wherein refrigerant in the liquid state collects in a lower portion of the separation chamber 342 and refrigerant in the vapor state collects in the portion of the separation chamber 342 above the liquid refrigerant.

[0030] The refrigerant vapor injection line 314 establishes refrigerant flow communication between an upper portion of the separation chamber 342 of the flash tank economizer 340 and an intermediate stage of the compression process. A vapor injection flow control device 343 is interdisposed in vapor injection line 314. The vapor injection flow control device 343 may comprise a flow control valve selectively positionable between an open position where refrigerant vapor flow may pass through the refrigerant vapor injection line 314 and a closed position where refrigerant vapor flow through the refrigerant vapor injection line 314 is reduced or blocked. In one embodiment, the vapor injection flow control valve 343 comprises a two-position solenoid valve of the type selectively positionable between a first open position and a second closed position.

[0031] The refrigeration vapor compression system 300 can also include an optional variable flow device (VFD) or a suction modulation valve (SMV) 323 interdisposed in refrigerant line 306 at a location between the outlet of the refrigeration heat absorption heat exchanger 350 and an inlet to the compression device 320. In the exemplary embodiment depicted in FIG. 3, the suction modulation valve 323 is positioned in refrigerant line 306 between the outlet of the evaporator 350 and the point at which the compressor unload bypass line 316 intersects refrigerant line 306. In one embodiment, the suction modulation valve 323 may comprise a pulse width modulated solenoid valve.

[0032] In a refrigerant vapor compression system operating in a transcritical cycle, the refrigerant heat rejection heat exchanger 330 constitutes a gas (refrigerant vapor) cooler through which supercritical refrigerant passes in heat exchange relationship with a cooling medium, such as for example, but not limited to ambient gas or liquid (e.g., air or water), and may be also referred to herein as a gas cooler. In a refrigerant vapor compression system operating in a subcritical cycle, the refrigerant heat rejection heat exchanger 330 can constitute a refrigerant condensing heat exchanger through which hot, high pressure refrigerant vapor passes in heat exchange relationship with the cooling medium and is condensed to a liquid. As shown in FIG. 3, the refrigerant heat rejection heat exchanger 330 includes a finned tube heat exchanger 332, such as for example a fin and round tube heat exchange coil or a fin and mini-channel flat tube heat exchanger, through which the refrigerant passes in heat exchange relationship with ambient air being drawn through the finned tube heat exchanger 332 by the fan(s) 334 associated with an exemplary gas cooler 330.

[0033] Whether the refrigerant vapor compression system 300 is operating in a transcritical cycle or a subcritical cycle, the refrigerant heat absorption heat exchanger 350 serves an evaporator wherein refrigerant liquid or a mixture of refrigerant liquid and vapor is passed in heat exchange relationship with a fluid to be cooled, most commonly air, drawn from and to be returned to a temperature controlled environment, such as a cargo box of a refrigerated transport truck, trailer or container, or a display case, merchandiser, freezer cabinet, cold room or other perishable/frozen product storage area in a commercial establishment, or to a climate controlled comfort zone within a residence, office building, hospital, school, restaurant or other facility. As shown in FIG. 3 the refrigerant heat absorption heat exchanger 350 comprises a finned tube heat exchanger 352 through which refrigerant passes in heat exchange relationship with air drawn from and returned to the refrigerated container 212 by the evaporator fan(s) 354 associated with the evaporator 350. The finned tube heat exchanger 352 may comprise, for example, a fin and round tube heat exchange coil or a fin and mini-channel flat tube heat exchanger.

[0034] The compression device 320 functions to compress the refrigerant and to circulate refrigerant through the primary refrigerant circuit as described in detail herein. In the embodiment depicted in FIG. 3, the compression device 320 may comprise a single multiple stage refrigerant compressor, such as for example a screw compressor or a reciprocating compressor disposed in the primary refrigerant circuit and having a first compression stage 320a and a second compression stage 320b. The first and second compression stages are disposed in series refrigerant flow relationship with the refrigerant leaving the first compression stage 320a passing directly to the second compression stage 320b for further compression. Alternatively, the compression device 320 may comprise a pair of independent compressors 320a and 320b, connected in series refrigerant flow relationship in the primary refrigerant circuit via a refrigerant line connecting the discharge outlet port of the first compressor 320a in refrigerant flow communication with an inlet port (e.g. the suction inlet port) of the second compressor 320b. In the independent compressor embodiment, the compressors 320a and 320b may be scroll compressors, screw compressors, reciprocating compressors, rotary compressors or any other type of compressor or a combination of any such compressors. In the embodiment depicted in FIG. 3, the refrigerant vapor compression system 300 includes a refrigerant bypass line 316 providing a refrigerant flow passage from an intermediate pressure stage of the compression device 320 back to the suction side of the compression device 320. An unload valve 327 is interdisposed in the bypass line 316. The unload valve 327 may be selectively positioned in an open position in which refrigerant flow passes through the bypass line 316 and a closed position in which refrigerant flow through the bypass line 316 is reduced or blocked.

[0035] In the embodiment depicted in FIG. 3, the refrigerant vapor compression system 300 further includes a refrigerant liquid injection line 318. The refrigerant liquid injection line 318 can tap into refrigerant line 304 at location downstream of the flash tank economizer 340 and upstream of the primary expansion valve 355 and open into an intermediate stage of the compression process. Thus, the refrigerant liquid injection line 318 can establish refrigerant flow communication between a lower portion of the separation chamber 342 of the flash tank economizer 340 and an intermediate pressure stage of the compression device 320. In one embodiment, the refrigerant liquid injection line 318 can establish refrigerant flow communication between a lower portion of the separation chamber 342 of the flash tank economizer 340 and a compressor suction line *(e.g.,* an inlet to the compression device). A liquid injection flow control device 353 can be interdisposed in refrigerant liquid injection line 318. The liquid injection flow control device 353 may comprise a flow control valve selectively positionable between an open position wherein refrigerant liquid flow may pass through the liquid injection line 318 and a closed position wherein refrigerant liquid flow through the refrigerant liquid injection line 318 is reduced or blocked. In an embodiment, the liquid injection flow control device 353 comprises a two-position solenoid valve of the type selectively positionable between a first open position and a second closed position.

[0036] In the exemplary embodiment of the refrigerant vapor compression system 300 depicted in FIG. 3, injection of refrigerant vapor or refrigeration liquid into the intermediate pressure stage of the compression process would be accomplished by injection of the refrigerant vapor or refrigerant liquid into the refrigerant passing from the first compression stage 320a into the second compression stage 320b of the compression device 320.

[0037] Liquid refrigerant collecting in the lower portion of the flash tank economizer 340 can pass therefrom through refrigerant line 304 and traverse the primary refrigerant circuit expansion valve 355 interdisposed in refrigerant line 304 upstream with respect to refrigerant flow of the evaporator 350. As this liquid refrigerant traverses the first expansion device 355, it expands to a lower pressure and temperature before entering the evaporator 350. The evaporator 350 constitutes a refrigerant evaporating heat exchanger through which expanded refrigerant passes in heat exchange relationship with the air to be cooled, whereby the refrigerant is vaporized and typically superheated. As in conventional practice, the primary expansion valve 355 meters the refrigerant flow through the refrigerant line 304 to maintain a desired level of superheat in the refrigerant vapor leaving the evaporator 350 to ensure that no liquid is present in the refrigerant leaving the evaporator. The low pressure refrigerant vapor leaving the evaporator 350 returns through refrigerant line 306 to the input port of the first compression stage or first compressor 320a of the compression device 320 in the embodiment depicted in FIG. 3.

[0038] The refrigerant vapor compression system 300 also includes a control system operatively associated therewith for controlling operation of the refrigerant vapor compression system 300. The control system can include a controller 390 that can determine the desired mode of operation in which to operate the refrigerant vapor compression system 300 based upon consideration of refrigeration load requirements, ambient conditions and various sensed system operating parameters. As shown in FIG. 3, the controller 390 also includes various sensors operatively associated with the controller 390 and disposed at selected locations throughout the system for monitoring various operating parameters by use of various sensors operatively associated with the controller. The control system may include, by way of example but not limitation, a pressure sensor 392 disposed in operative association with the flash tank economizer 340 to sense the pressure within the separation chamber 342, a temperature sensor 393 and a pressure sensor 394 for sensing the refrigerant inlet or suction temperature and pressure, respectively, and a temperature sensor 395 and a pressure sensor 396 for sensing refrigerant discharge temperature and pressure, respectively. In transport refrigeration applications, the refrigeration vapor compression system may also include a temperature sensor 397a for sensing the temperature of the air returning to the evaporator from the container 212 and a temperature sensor 397b for sensing a temperature of the air being supplied to the container 212. Sensors (not shown) may also be provided for monitoring ambient outdoor conditions, such as or example ambient outdoor air temperature and humidity. By way of example but not limitation; the pressure sensors 392, 394, 396 may be conventional pressure sensors, such as for example, pressure transducers, and the temperature sensors 393, 395 may be conventional temperature sensors, such as for example, thermocouples or thermistors.

[0039] The controller 390 processes the data received from the various sensors and controls operation of the compression device 320, operation of the fan(s) 334 associated with the refrigerant heat rejection heat exchanger 330, operation of the fan(s) 354 associated with the evaporator 350, operation of the primary expansion device 355, operation of the secondary expansion device 345, and operation of the suction modulation valve 323. The controller 390 also controls the positioning of the vapor injection valve 343 and liquid injection valve 353. The controller 390 positions the vapor injection valve 343 in an open position for selectively permitting refrigerant vapor to pass from the flash tank economizer 340 through refrigerant vapor injection line 314 for injection into an intermediate stage of the compression process. Similarly, the controller 390 positions the liquid injection valve 353 in an open position for selectively permitting refrigerant liquid to pass from the flash tank economizer 340 through refrigerant liquid injection line 318 for injection into an intermediate pressure stage of the compression process. In the FIG. 3 embodiment, the controller 390 can also control the positioning of the unload valve 327 to selectively open the unload valve 327 to bypass refrigerant from an intermediate pressure stage of the compression device 320 through bypass line 316 back to the suction side of the compression device 320 when it is desired to unload the first stage of the compression device 320.

[0040] According to embodiments of the application, there are selected operation characteristics in a transport refrigeration system that can affect performance or overall system performance. During transport refrigeration system operations, it is desirable to check such characteristics to determine proper component or system functions and/or operations. In one embodiment, a measured value and a calculated value for a component/system performance characteristic can be determined and compared, and then a judgment can be made responsive to or based on the comparison.

[0041] For example, a compressor mid-stage pressure and gas cooler exit temperature can be used to control or optimize CO_2 economized refrigeration system operations for capacity and/or efficiency. In one embodiment, gas cooler exit temperature is used to determine a prescribed compressor discharge pressure. In an embodiment, compressor mid-stage pressure is used to determine whether economized mode can/is entered by a vapor compression system.

[0042] In a refrigeration system, the refrigerant temperature exiting the heat rejection heat exchanger reflects the heat exchanger coil and fan performance. When the transport refrigeration system operates in a transcritical application, then the refrigerant temperature exiting the heat rejection heat exchanger is in the function that can determine or optimize compressor discharge pressure in the refrigeration system for either higher cooling capacity or higher energy efficiency. For at least this reason, embodiments of the application can determine or verify that this performance characteristic (e.g., refrigerant temperature exiting the gas cooler) is within a prescribed range or a system design range. In one embodiment, the heat rejection heat exchanger is sized for the highest capacity conditions of the system 300 (e.g., under which the system can be intended to operate). Therefore, for a majority or almost all of designed operating conditions, the heat rejection heat exchanger is oversized. As determined by the inventors, the refrigerant temperature exiting heat rejection heat exchanger (*e.g.*, shown as GCXT in the graph in FIG. 4) was determined (*e.g.*, tested) to be only slightly higher than ambient temperature. Thus, in one embodiment, the exiting temperature of refrigerant for the heat rejection heat exchanger can be calculated or verified using ambient temperature plus a variable offset. The variable offset can be determined to have a prescribed relationship to the cooling capacity of the system 300. In one embodiment, the highest offset can occur at highest cooling capacity conditions. As shown in FIG. 4, an offset is shown on the Y axis and can be defined as (Tamb-GCXT). The temperature difference between evaporator return air temperature (RTS) and supply air temperature (STS) is shown on X axis. The temperature difference (RTS-STS) is one exemplary measurement of system 300 cooling capacity. In one embodiment, the temperature difference (RTS-STS) is directly related *(e.g.,* a prescribed relationship) to the transport refrigeration system cooling capacity.

[0043] In one embodiment, the transport refrigeration system capacity can be determined responsive to an operating mode of the transport refrigeration system.

[0044] A sensor 382 can be provided in the system 300 shown in FIG. 3 to measure the refrigerant temperature exiting heat rejection heat exchanger 330. The sensor 382 can be a temperature sensor. Alternatively, the sensor 382 can be a pressure sensor where the temperature can be determined using the pressure. In one embodiment, a calculated temperature can be compared to the temperature provided using the sensor 382. When corresponding values do not match, an error condition in the sensor 382 can be identified by the controller 390 provided to an operator or the like.

[0045] In an economized refrigeration system, compressor mid stage pressure is an operation characteristic that can be monitored because the compressor mid stage pressure affects whether the system can transition into economized mode for higher capacity and higher energy efficiency. For at least this reason, the controller 390 can operate to verify proper compressor functions determined through a compressor mid stage pressure performance check during system 300 operations which can be executed according to embodiments of the application by a comparison of a measured value and a calculated (e.g., indirect) value for the compressor mid-stage pressure.

[0046] An exemplary indirect determination for the compressor mid-stage pressure will now be described. FIG. 5 shows the compressor mid-stage pressure as a function of the compressor discharge pressure for various compressor suction pressures. As shown in FIG. 5, the compressor mid-stage pressure can be determined when the suction and discharge pressure of the compressor 320 are known. The same information can be written in the form of an exemplary two-dimensional lookup table below.

	P Suction 1	**P Suction 2**	**P Suction 3**	**P Suction 4**
P Discharge 1	P Mid-Stage 1,1	P Mid-Stage 1,2	P Mid-Stage 1,3	P Mid-Stage 1,4
P Discharge 2	P Mid-Stage 2,1	P Mid-Stage 2,2	P Mid-Stage 2,3	P Mid-Stage 2,4
P Discharge 3	P Mid-Stage 3,1	P Mid-Stage 3,2	P Mid-Stage 3,3	P Mid-Stage 3,4
P Discharge 4	P Mid-Stage 4,1	P Mid-Stage 4,2	P Mid-Stage 4,3	P Mid-Stage 4,4

[0047] It should be understood that the values of the suction, discharge, and mid-stage pressures are specific to the compressor design and operating conditions (e.g., compressor 320). When the operating conditions for a given compressor machine change, for instance if the suction superheat changes, the values of the mid-stage pressure for a particular combination of suction and discharge pressure may change. This can be more pronounced if the compressor design allows to independently control the speed of the two compressor stages, for instance if the two stages are driven by different motors, for which the speed can be adjusted independently from each other. In this case, an additional dimension can be added to the graph or lookup table. For example, an additional dimension can be accomplished by providing additional graphs or tables, each for a constant value of the additional variable.

[0048] A sensor 384 can be provided in the system 300 shown in FIG. 3 to measure the compressor mid-stage pressure. The sensor 384 can be a pressure sensor. In one embodiment, a calculated compressor mid-stage pressure can be compared to the compressor mid-stage pressure provided using the sensor 384. When corresponding values do not match, an error condition in the sensor 384 can be determined by the controller 390 provided to an operator or the like.

[0049] An embodiment of a method of operating a transport refrigeration unit according to the application will now be described. The method embodiment shown in FIG. 6, can be implemented in and will be described using a refrigerant vapor compression system embodiment shown in FIG. 3, however, the method embodiment is not intended to be limited thereby.

[0050] Referring now to FIG. 6, a process as performed by the controller 390 can be shown in block diagram form. After a process starts during system operations, an operating characteristic of the system can be measured (*e.g.*, Cm) (operation block 610). Then, the operating characteristic of the system can be indirectly determined or calculated (*e.g.*, Cc) from other system components and/or characteristics according to a prescribed relationship (operation block 620). It can be determined whether Cm and Cc match (operation block 630). When the determination in operation block 630 is negative, an error condition can be processed (operation block 640). When the determination in operation block 630 is affirmative or from operation block 640, a delay period (operations block 650) can be processed before control returns to operation block 610.

[0051] In one embodiment, a calculated measurement for a system characteristic can be more accurate than a measured value. Thus, the error condition can be processed in operation block 640 by having the controller 390 stop using the measure value Cm and begin using the calculated value Cc.

[0052] In one embodiment, a calculated or indirect measurement of selected characteristics (e.g., compressor unit stage pressure and/or gas cooler refrigerant exit temperature) of transport refrigeration systems including refrigerant vapor compression systems can be determined with sufficient accuracy that sensors can be reduced or eliminated from the system, which may increase reliability and decrease size and cost. In one embodiment, the controller 390 can be responsive to a pressure difference between the flash tank and a mid-stage of the compressor to protect or prevent operation of the economizer during periods in which the pressure at the mid-stage is greater than the pressure in the flash tank or control operations of a flow control device (e.g., flow control device 343, 353) coupled thereto.

[0053] Embodiments according to the application can use remote sensors to respectively measure an environment within the container 12 such as the return air temperature RTS and the supply air temperature STS. Remote sensors, as known to one skilled in the art, can communicate with a controller (e.g., transport refrigeration unit 10) through wire or wireless communications. For example, wireless communications can include one or more radio transceivers such as one or more of 802.11 radio transceiver, Bluetooth radio transceiver, GSM/GPS radio transceiver or WIMAX (802.16) radio transceiver. Information collected by remote sensor(s) can be used as input parameters for a controller to control various components in transport refrigeration systems. In one embodiment, remote sensors may monitor additional criteria such as humidity, species concentration or the like.

[0054] It should be recognized that selected procedures described herein may result in some liquid refrigerant entering the compressor inlet. Although this is generally undesirable, it may occur for short periods of time without any significant damage to the compressor.

[0055] While the present invention has been described with reference to a number of specific embodiments, it will be understood that the scope of the invention should be determined only with respect to claims that can be supported by the present specification. Further, while in numerous cases herein wherein systems and apparatuses and methods are described as having a certain number of elements it will be understood that such systems, apparatuses and methods can be practiced with fewer than the mentioned certain number of elements. Also, while a number of particular embodiments have been set forth, it will be understood that features and aspects that have been described with reference to each particular embodiment can be used with each remaining particularly set forth embodiment. For example, features or aspects described with respect to FIG. 3 can be used, combined with or replace features described using FIGS. 4-6.

REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

- EP1965158A1 [0002]

Patentkrav

1. Kølemiddeldampkompressionssystem, der omfatter:

en kølemiddelkompressionsindretning (320);

en kølemiddelvarmeafvisningsvarmeveksler (330) nedstrøms for kompressionsindretningen;

en kølemiddelvarmeabsorptionsvarmeveksler (350) nedstrøms for kølemiddelvarmeafvisningsvarmeveksleren (330);

en første ekspansionsindretning (355), der er anbragt nedstrøms for kølemiddelvarmeafvisningsvarmeveksleren (330) og opstrøms for kølemiddelvarmeabsorptionsvarmeveksleren (350);

en sensor (382), som er koblet til en udgang i varmeafvisningsvarmeveksleren (330), idet sensoren skal måle en kølemiddeltemperatur; og

en styreenhed (390) til styring af driften af køledampkompressionssystemet, idet styreenheden er operativ til indirekte at verificere den målte kølemiddeltemperatur,

kendetegnet ved, at kompressionsindretningen indbefatter et første kompressionstrin (320a) og et andet kompressionstrin (320b), og ved, at kølemiddeltemperaturen ved varmeafvisningsvarmevekslerens (330) udgang først bestemmes ved måling under anvendelse af sensoren (382) og dernæst bestemmes ved beregning under anvendelse af omgivelsestemperatur og dampkompressionssystemkapacitet.

2. Kølemiddeldampkompressionssystem ifølge krav 1, hvor dampkompressionssystemkapaciteten har et foreskrevet forhold med en driftsmodus eller forskel mellem forsyningslufttemperatur (T_s) og returlufttemperatur (T_r); og hvor der tilføjes en forskydning til omgivelsestemperaturen, som er responderbar på dampkompressionssystemkapaciteten.

3. Kølemiddeldampkompressionssystem ifølge krav 2, idet styreenheden (390) er konfigureret til at drive dampkompressionssystemet med den beregnede værdi for kølemiddeltemperaturen, når den målte kølemiddeltemperatur er forskellig fra den beregnede temperaturværdi.

4. Dampkompressionssystem ifølge krav 1, 2 eller 3, hvor sensoren (382) er en tryksensor eller en temperatursensor.

5. Kølemiddeldampkompressionssystem ifølge et hvilket som helst foregående krav, der omfatter en anden sensor (384) til måling af et kompressormellemtrintryk, idet styreenheden (390) er konfigureret til indirekte at verificere det målte kompressormellemtrintryk.

6. Kølemiddeldampkompressionssystem ifølge krav 5, hvor kompressormellemtrintrykket beregnes under anvendelse af et afgangstryk og et indgangstryk for kompressoren (320).

7. Kølemiddeldampkompressionssystem ifølge krav 5 eller 6, hvor styreenheden (390) er konfigureret til at drive dampkompressionssystemet under anvendelse af den verificerede værdi for kompressormellemtrintrykket, hvor det målte kompressormellemtrintryk ikke passer med den indirekte verificerede værdi.

8. Kølemiddeldampkompressionssystem ifølge et hvilket som helst foregående krav, der omfatter:

en anden ventil (345), der er anbragt nedstrøms for varmeafvisningsvarmeveksleren (330);

og et fødevandsforvarmerkredsløb (340), der er anbragt nedstrøms for den anden ventil (345) og opstrøms for den første ekspansionsindretning (355), idet fødevandsforvarmerkredsløbet indbefatter en kølemiddelinjektionsledning (314, 318) til åbning til et mellemtryktrin for kompressionsindretningen og en gennemstrømningsreguleringsventil (343, 353), der er anbragt i kølemiddelinjektionsledningen.

9. Kølemiddeldampkompressionssystem ifølge krav 8, hvilken styreenhed (390) er konfigureret til at lukke gennemstrømningsreguleringsventilen (343, 353), når kompressormellemtrintrykket er operativt til at få kølemiddel til at strømme mod fødevandsforvarmerkredsløbet.

10. Kølemiddeldampkompressionssystem ifølge et hvilket som helst af kravene 1 til 7, der omfatter:

en flash-tank-fødevandsforvarmer (340), der er anbragt i seriestrømforbindelse mellem varmeafvisningsvarmeveksleren (330) og den første ekspansionsindretning (355), hvilken flash-tank-fødevandsforvarmer indbefatter:

en flash-tank (342);

en første gennemstrømningsreguleringsindretning (345), der er anbragt mellem varmeafvisningsvarmeveksleren (330) og flash-tanken (342);

en fødevandsforvarmerdampledning (314) til indbyrdes fluidforbindelse af flash-tanken til et mellemtrin for kompressoren; og

en anden gennemstrømningsreguleringsindretning (342), der er anbragt i fødevandsforvarmerdampledningen.

11. Fremgangsmåde til bestemmelse af en egenskab for et kølemiddeldampkompressionssystem med et kølemiddelkredsløb, der indbefatter en kølemiddelkompressionsindretning (320), en kølemiddelvarmeafvisningsvarmeveksler (330) nedstrøms for kompressionsindretningen, en kølemiddelvarmeabsorptionsvarmeveksler (350) nedstrøms for kølemiddelvarmeafvisningsvarmeveksleren, en sensor (382) til registrering af egenskaben med henblik på at bestemme en systemkapacitet for kølemiddeldampkompressionssystemet under drift, hvilken egenskab er en kølemiddeltemperatur ved en udgang i varmeafvisningsvarmeveksleren (330), og indbyrdes forbindelse af kølemiddelledninger som aktive bestanddele, hvilken fremgangsmåde omfatter:

drift af kølemiddeldampkompressionssystemet i en modus, hvor kølemidlet cirkulerer i de aktive bestanddele i kølemiddelkredsløbet;

indirekte bestemmelse af kølemiddeltemperaturen ved varmeafvisningsvarmevekslerens (330) udgang under anvendelse af omgivelsestemperatur og dampkompressionssystemkapacitet;

sammenligning af den registrerede værdi for kølemiddeltemperaturen ved varmeafvisningsvarmevekslerens (330) udgang mod den indirekte bestemte værdi for kølemiddeltemperatur ved varmeafvisningsvarmevekslerens (330) udgang; og

bestemmelse af en fejltilstand for en tilsvarende sensor, når et resultat af sammenligningen ikke passer sammen.

12. Fremgangsmåde ifølge krav 11, der omfatter efterfølgende anvendelse af den indirekte bestemte værdi ved drift af dampkompressionssystemet.

13. Computerprogramprodukt, der omfatter et computeranvendeligt lagermedium til lagring af et computerlæsbart program, der, når det afvikles på en computer, får computeren til at udføre operationer til drift af en transportkøleunit, hvilke operationer omfatter:

drift af transportkøleenheden i en modus, hvor et kølemiddel cirkulerer i et kølemiddelkredsløb;

registrering af en kølemiddeltemperatur ved en udgang for en varmeafvisningsvarmeveksler (330) til bestemmelse af en systemkapacitet for transportkøleenheden under drift;

indirekte bestemmelse af kølemiddeltemperaturen ved varmeafvisningsvarmevekslerens (330) udgang under anvendelse af omgivelsestemperatur og dampkompressionssystemkapacitet;

sammenligning af den registrerede værdi for kølemiddeltemperaturen ved varmeafvisningsvarmevekslerens (330) udgang mod den indirekte bestemte værdi; og

bestemmelse af en fejltilstand for en tilsvarende sensor, når et resultat af sammenligningen ikke passer sammen.

DRAWINGS

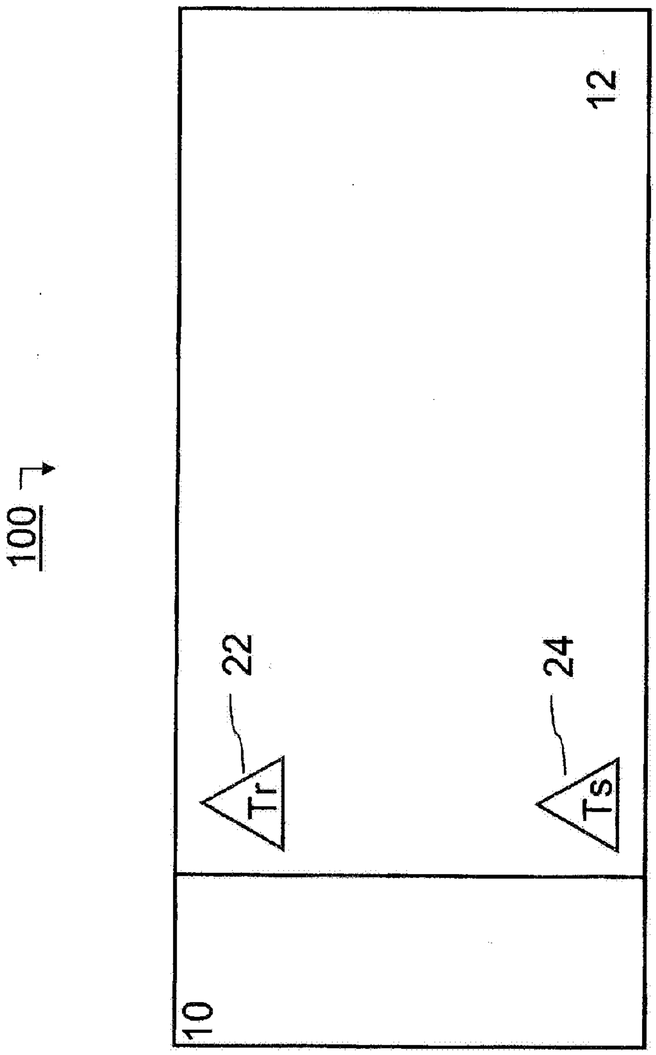

FIG. 1

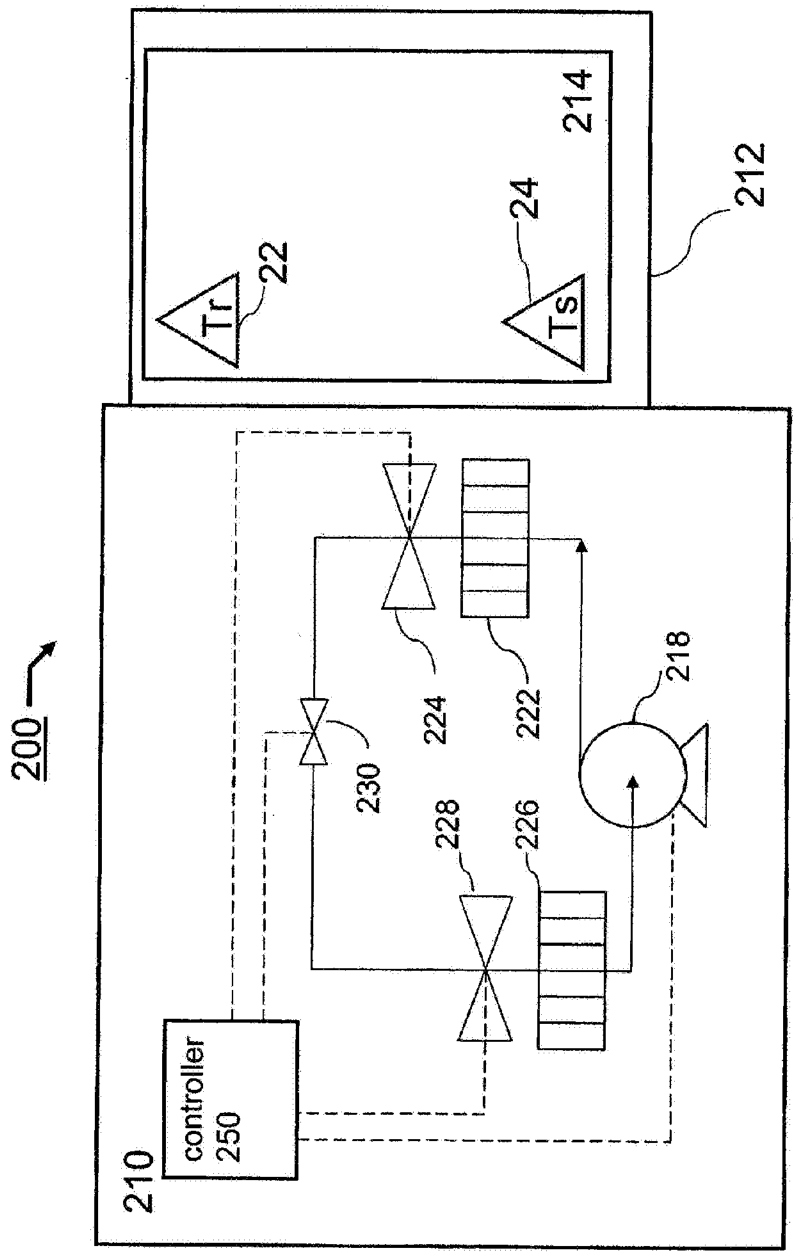
200
210
212
214
Tr
22
Ts
24
controller
250
218
222
224
226
228
230

FIG. 2

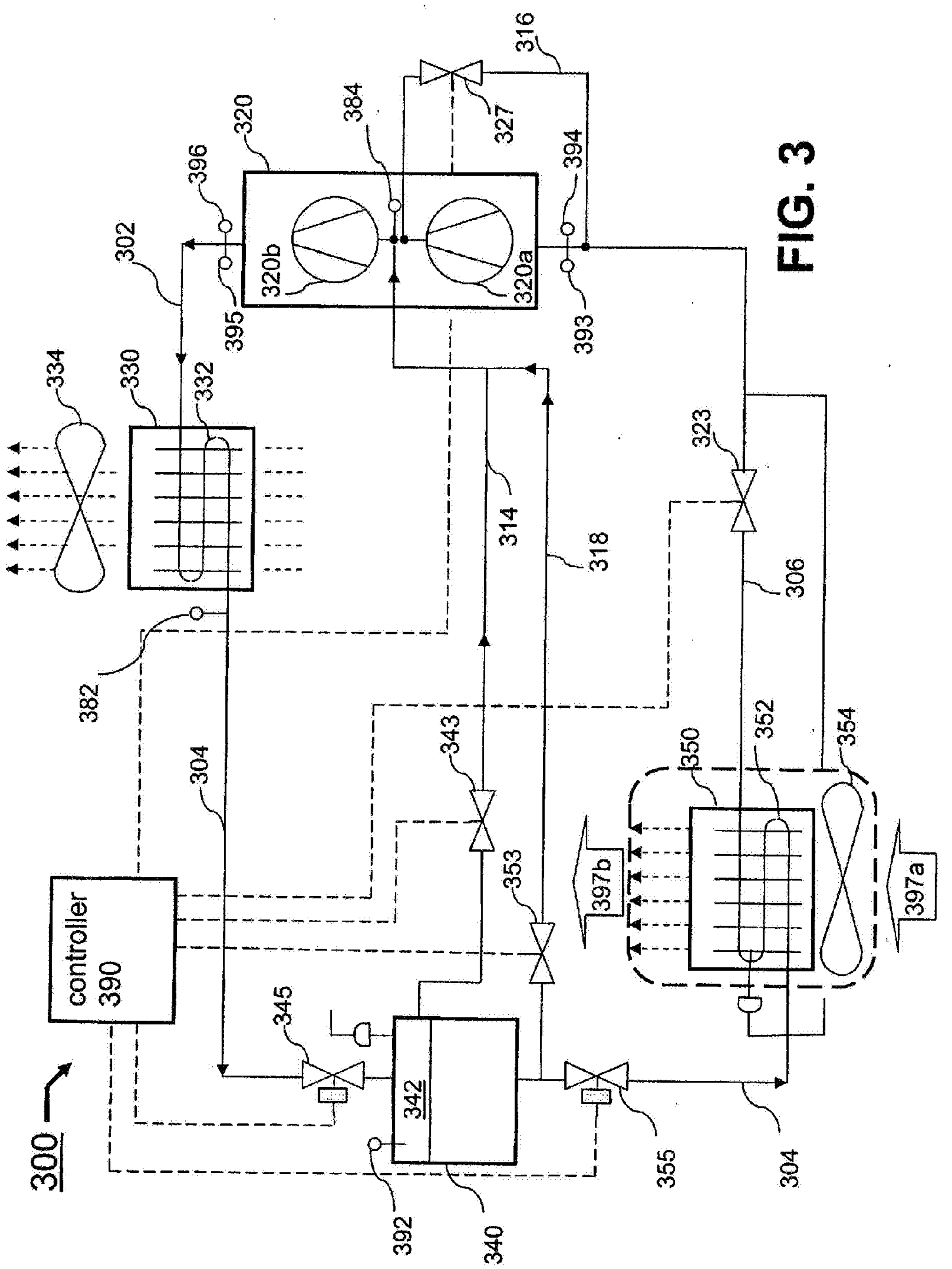
300
controller
390
302
395
396
320
320b
320a
384
327
316
394
393
334
330
332
382
304
314
318
343
353
345
342
392
340
355
304
323
306
350
352
354
397b
397a

FIG. 3

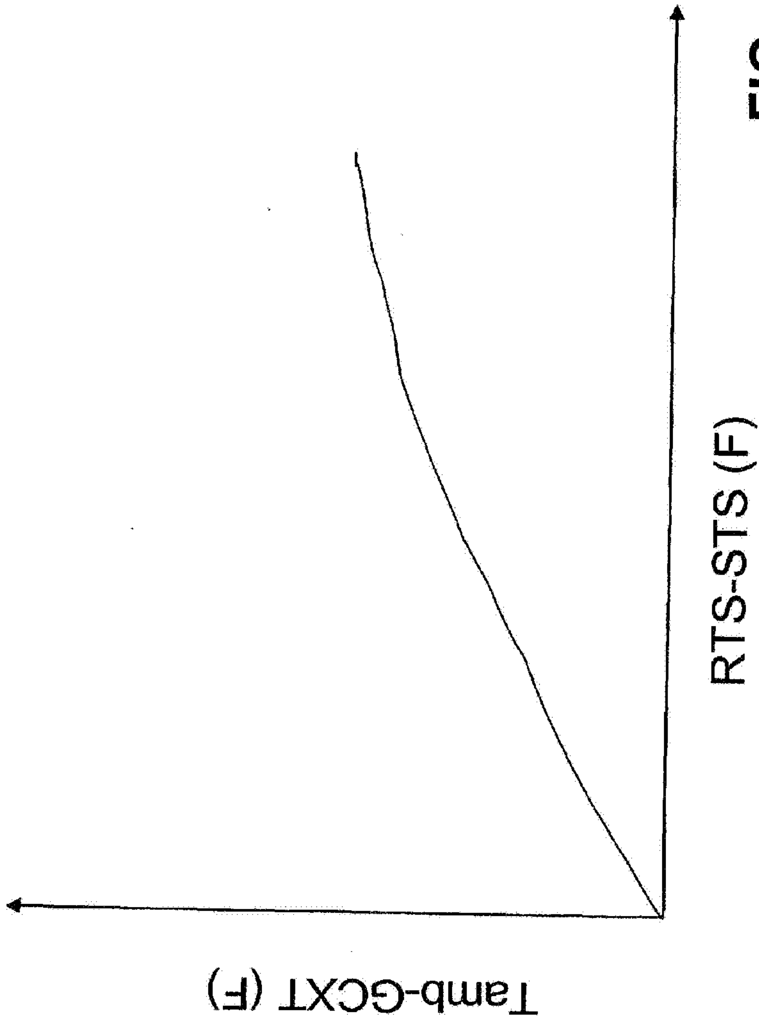

FIG. 4

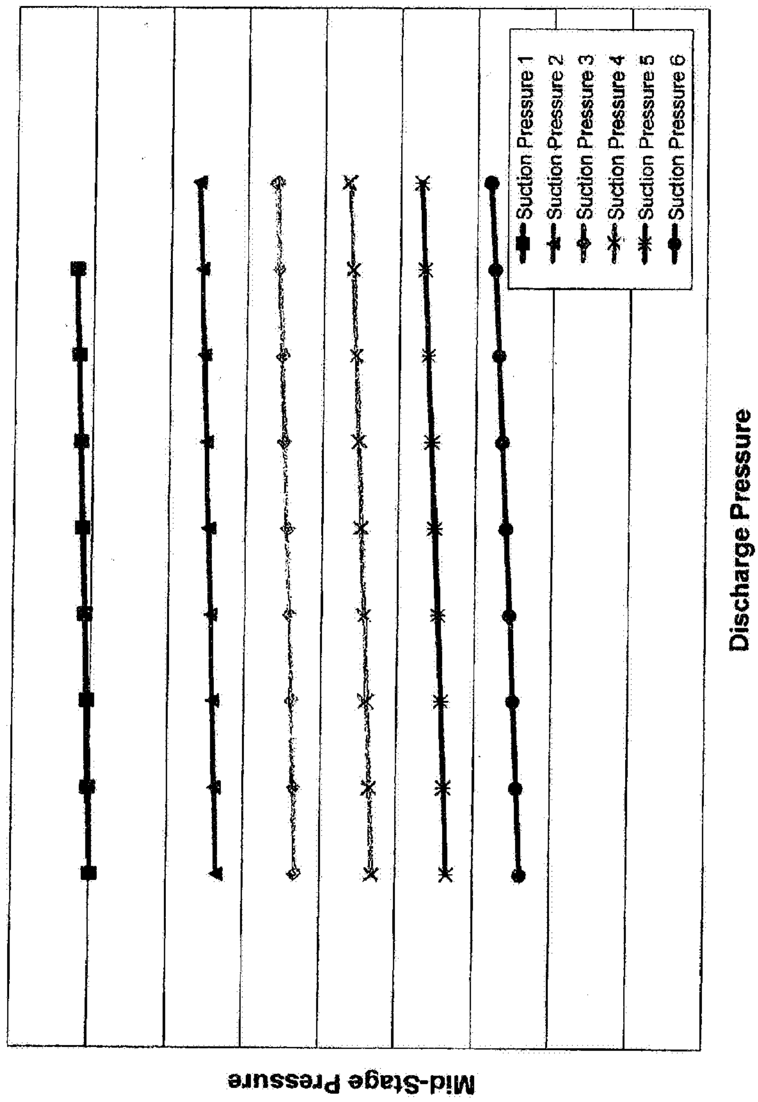
Suction Pressure 1
Suction Pressure 2
Suction Pressure 3
Suction Pressure 4
Suction Pressure 5
Suction Pressure 6
Discharge Pressure
Mid-Stage Pressure

FIG. 5

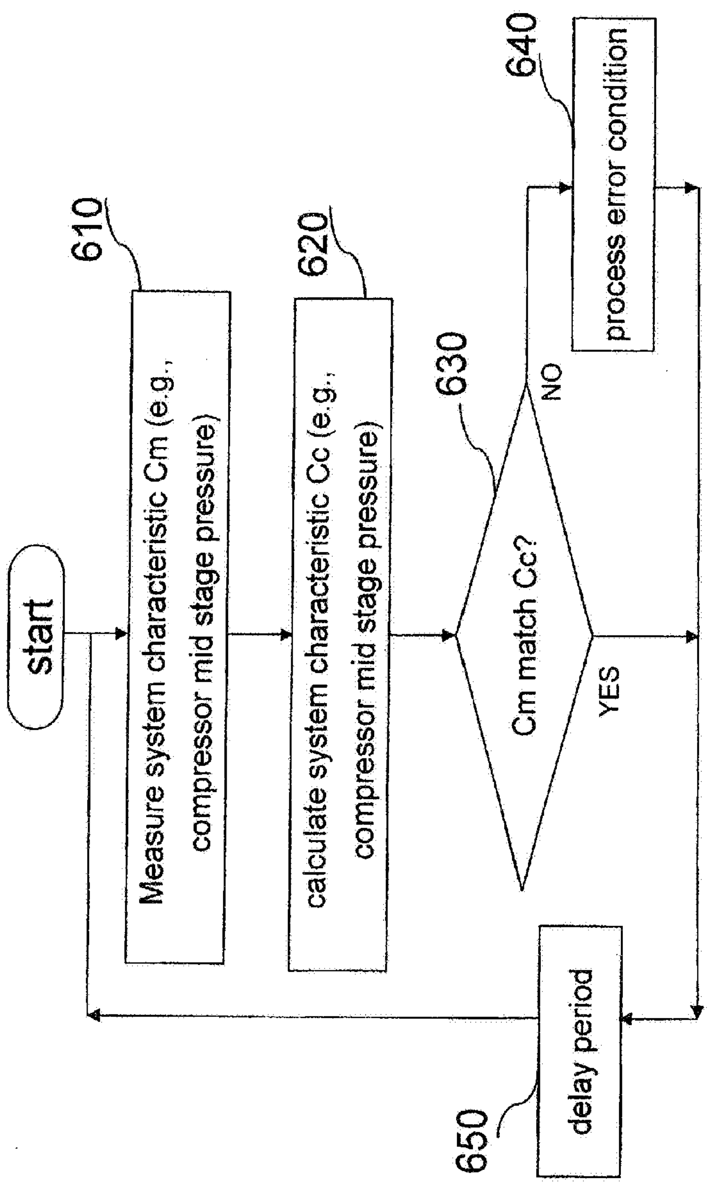
start
610
Measure system characteristic Cm (e.g., compressor mid stage pressure)
620
calculate system characteristic Cc (e.g., compressor mid stage pressure)
630
Cm match Cc?
NO
YES
640
process error condition
650
delay period

FIG. 6