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(54) REFRIGERATION/AIR-CONDITIONING APPARATUS POWERED BY AN ENGINE EXHAUST GAS DRIVEN TURBINE

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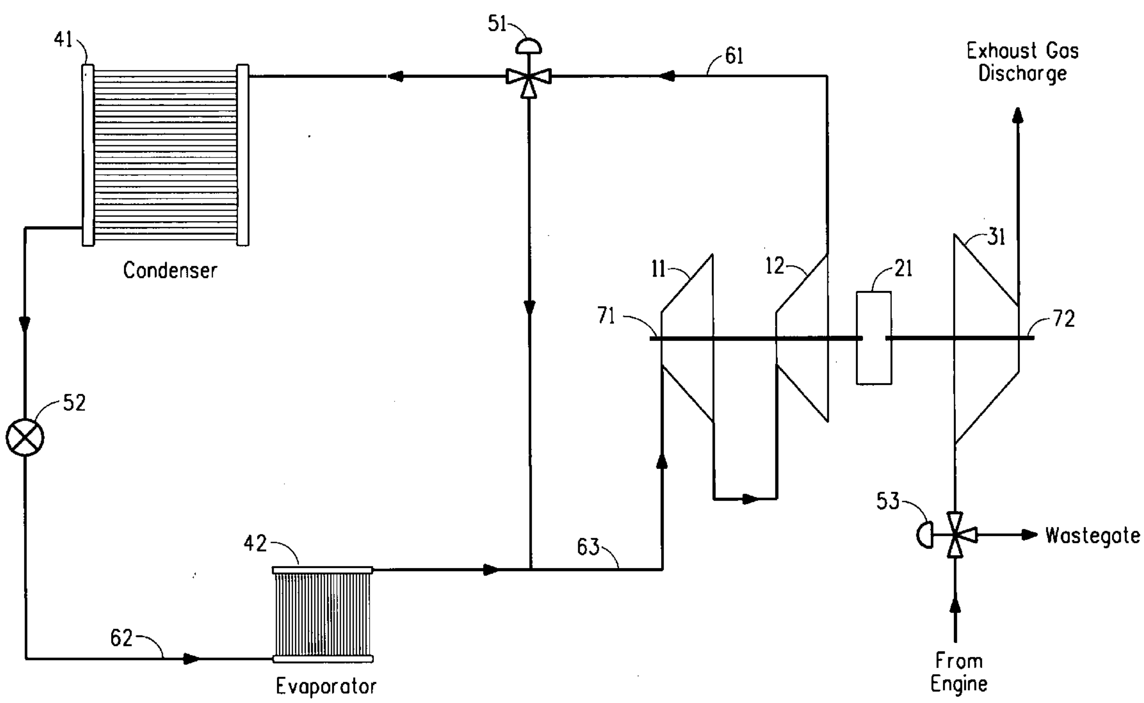
(57) **ABSTRACT**

The present invention relates to a refrigeration or air-conditioning apparatus which utilizes a vapor compression refrigeration system having a refrigerant circulating therethrough that comprises a compressor powered by an engine exhaust gas driven turbine. A mini centrifugal compressor may advantageously be used with such an apparatus, thus allowing the use of low GWP refrigerants. The present invention further relates to methods for powering a compressor, such as a mini-centrifugal compressor, in a refrigeration or air-conditioning apparatus, and methods for controlling compressor surge, impeller speed and cooling capacity.



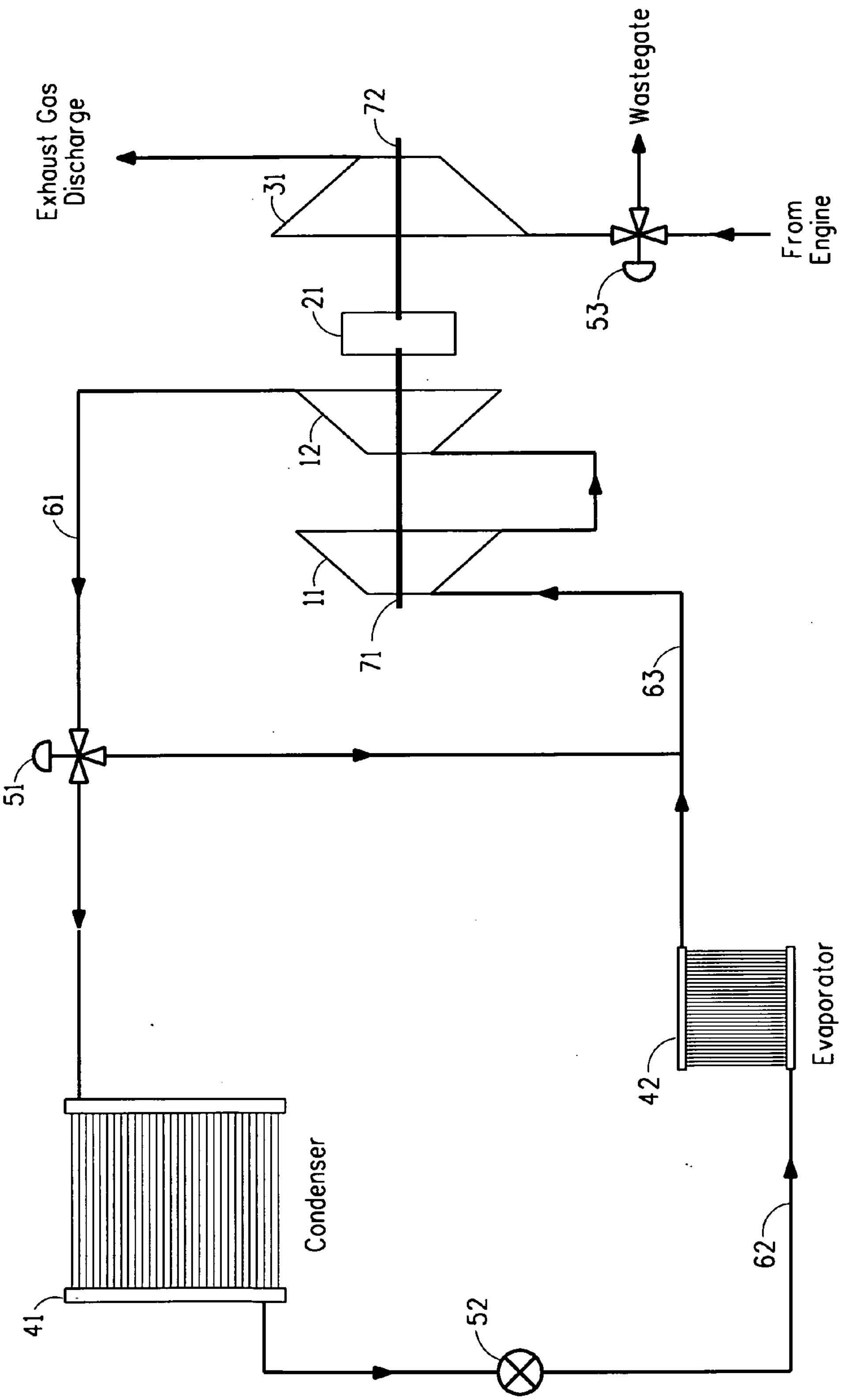

Exhaust Gas Discharge
Wastegate
72
31
From Engine
53
21
12
61
11
71
63
51
Evaporator
42
Condenser
41
52
62

FIG. 1

REFRIGERATION/AIR-CONDITIONING APPARATUS POWERED BY AN ENGINE EXHAUST GAS DRIVEN TURBINE

CROSS REFERENCE(S) TO RELATED APPLICATION(S)

[0001] This application claims the priority benefit of U.S. Provisional Application 60/658,915, filed Mar. 4, 2005.

BACKGROUND OF THE INVENTION

[0002] 1. Field of the Invention

[0003] The present invention relates to refrigeration or air-conditioning apparatus. In particular, the present invention relates to stationary or mobile refrigeration or air-conditioning apparatus utilizing a vapor compression refrigeration system which includes a compressor powered by an engine exhaust gas driven turbine.

[0004] 2. Description of Related Art

[0005] The refrigeration industry has been working for the past few decades to find replacement refrigerants for the ozone depleting chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) being phased out as a result of the Montreal Protocol. The solution for most refrigerant producers has been the commercialization of hydrofluorocarbon (HFC) refrigerants. The new HFC refrigerants, HFC-134a being the most widely used at this time, have zero ozone depletion potential and thus are not affected by the current regulatory phase out as a result of the Montreal Protocol.

[0006] Further environmental regulations may ultimately cause global phase out of certain HFC refrigerants. Currently, the automobile industry is facing regulations relating to global warming potential (GWP) for refrigerants used in mobile air-conditioning. Therefore, there is a great current need to identify new refrigerants with reduced global warming potential for the automobile air-conditioning market. Should the regulations be more broadly applied in the future, an even greater need will be felt for refrigerants that can be used in all areas of the refrigeration and air-conditioning industry.

[0007] Currently proposed replacement refrigerants for HFC-134a include HFC-152a, pure hydrocarbons such as butane or propane, or "natural" refrigerants such as CO_2 or ammonia. Many of these suggested replacements are toxic, flammable, and/or have low energy efficiency. Therefore, new alternatives are constantly being sought.

[0008] A new approach to this problem of high GWP HFC's for the refrigeration and air-conditioning market involves the use of new, low-pressure, low GWP refrigerants in an innovative type of vapor compression refrigeration or air-conditioning apparatus. Miniature scale centrifugal (mini-centrifugal) compressors would facilitate the use of these new low GWP refrigerants. However, the power requirements for such a system are not met in existing automobile designs.

[0009] Conventionally, in mobile air conditioning systems, or stationary air conditioning systems which are driven by internal combustion engines, power is transmitted from the engine to the air conditioner compressor via a system of belts and pulleys. It is known that power transmission by this method has associated inefficiencies due to heat losses from internal friction within the belt, slippage of belts, and loss of function due to belt breakage. In addition, it is difficult to achieve the step up in rotational speed from normal engine rotation speeds to the rotational speed required to run a centrifugal compressor via belts and pulleys without even greater loss of efficiency and reliability.

[0010] The use of engine exhaust gas to drive an air cycle system to produce hot or cold air that is delivered to the passenger compartment of an automobile is described in U.S. Pat. No. 5,172,753 to GM. This type of heating or cooling cycle is an open air cycle, as there is no phase change of the air. In this system, air is continuously taken from ambient and delivered to the passenger compartment to provide heating or cooling. Thus, in this system the heating or cooling air properties can be dependent on the quality and the temperature variations of the inlet ambient air.

[0011] Therefore, it would be desirable to develop a system for cooling the air in the passenger compartment of an automobile which is not dependent on the use of ambient air. It would also be desirable if such a system could meet the power requirements for mini-centrifugal compressors, so that low GWP refrigerants could be used in such a system.

BRIEF SUMMARY OF THE INVENTION

[0012] The present invention overcomes the problems associated with the prior art by providing a refrigeration or air-conditioning apparatus utilizing a vapor compression refrigeration system. In this vapor compression refrigeration system, refrigerant circulates through a closed loop and undergoes the cycle of refrigerant gas compression, condensation, expansion and evaporation to produce cooling. Through the use of a refrigerant in the vapor compression refrigeration system, desired cooling which is not dependent on the inconsistencies of ambient air can be attained.

[0013] The apparatus of the present invention includes a compressor which is driven by an engine exhaust gas driven turbine. This arrangement can provide the power requirements that are necessary to power a mini-centrifugal compressor. Thus, the present invention allows the use of the mini-centrifugal compressor, which facilitates the use of these new low GWP refrigerants.

[0014] Under conditions of low engine exhaust gas flow, such as engine idle, the power requirements for adequate cooling may not be met. When the air-conditioning system is operating, the engine idle may be set to a higher speed, only during air-conditioner operation, such that there is enough power to provide the cooling needed thus providing a further advantage of the present invention.

[0015] Therefore, in accordance with the present invention, there is provided a refrigeration or air-conditioning apparatus comprising a vapor compression refrigeration system having a refrigerant circulating therethrough, where the vapor compression refrigeration system includes a compressor driven by an engine exhaust gas driven turbine.

[0016] Further in accordance with the present invention, there is provided a method for providing power to refrigeration or air-conditioning apparatus comprising a vapor compression refrigeration system having a refrigerant circulating therethrough, the system including a compressor, said method comprising coupling the compressor to an engine exhaust gas driven turbine.

[0017] Further in accordance with the present invention, there is provided a method for controlling impeller speed in a compressor of a vapor compression system having a refrigerant circulating therethrough, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

[0018] Also in accordance with the present invention, there is also provided a method for controlling the cooling capacity of a vapor compression system having a refrigerant circulating therethrough, wherein said system comprises a compressor powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

[0019] Further in accordance with the present invention, there is provided a method for controlling impeller speed in a compressor of a vapor compression system, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the engine idle speed during air-conditioner operation.

[0020] In accordance with the present invention, there is provided a method for controlling the cooling capacity of a vapor compression system having a refrigerant circulating therethrough, wherein the system comprises a compressor powered by an engine exhaust gas turbine, said method comprising varying the engine idle speed during operation.

[0021] Also in accordance with the present invention, there is provided a process to produce cooling comprising compressing a refrigerant in a compressor powered by an engine exhaust gas driven turbine; condensing said refrigerant; and thereafter evaporating said refrigerant in the vicinity of a body to be cooled.

[0022] According to all of the above-described apparatus, methods and processes, the compressor may be centrifugal, and more preferably, centrifugal mini compressor.

BRIEF DESCRIPTION OF THE DRAWINGS

[0023] The present invention may be better understood with reference to the following figure, wherein:

[0024] **FIG. 1** is a schematic diagram of the refrigeration or air conditioning apparatus, including a vapor compression refrigeration cycle which includes a compressor coupled to an engine exhaust gas driven turbine.

DETAILED DESCRIPTION OF THE INVENTION

[0025] According to the present invention, there is provided a refrigeration or air-conditioning apparatus. Such an apparatus is shown in **FIG. 1**. The apparatus of the present invention includes a vapor compression refrigeration system. A vapor-compression system is a closed loop system which re-uses refrigerant in multiple steps producing a cooling effect in one step and a heating effect in a different step. Such a system generally includes an evaporator, a compressor, a condenser and an expansion device, as will be described below in detail with respect to **FIG. 1**. With reference to **FIG. 1**, gaseous refrigerant from an evaporator (**42**) flows through a pipeline (**63**) to a mini-centrifugal compressor where it enters at the suction of a first stage housing, having an impeller (**11**) therein, and is discharged from the first stage housing to the suction of a second stage housing having a second impeller (**12**) therein. The compressed refrigerant gas output from the second stage housing flows through a pipeline (**61**) from the compressor to a condenser (**41**). A pressure regulating valve (**51**) in pipeline (**61**) allows recycle of the refrigerant flow back to the compressor via a pipeline (**63**) providing the ability to control the pressure of the refrigerant reaching the condenser (**41**) and if necessary to prevent compressor surge. The compressed refrigerant is condensed in the condenser, thus giving off heat. The liquid refrigerant flows through an expansion device (**52**) via a pipeline (**62**) to the evaporator (**42**), which is located in the passenger compartment. In the evaporator, the liquid refrigerant is vaporized providing cooling and the cycle then repeats. The expansion device (**52**) may be an expansion valve, a capillary tube or an orifice tube.

[0026] The compressor of the present invention is driven by an engine exhaust gas driven turbine. In order to accomplish this, exhaust gas from an engine is directed through a waste gate valve (**53**) as shown in **FIG. 1** into an engine exhaust gas driven turbine (**31**). The waste gate valve (**53**) provides control on the amount of exhaust reaching the turbine and thus the speed of the turbine shaft. The engine exhaust gas flow is split at the wastegate valve (**53**) with some flow going to the turbine and the remainder of the flow going to exhaust (e.g., the automobile exhaust system).

[0027] In order to facilitate the common drive of the turbine and the compressor, the turbine and compressor may share a common rotating shaft. Alternatively, there may be separate rotating shafts, **72** and **71**, respectively, as shown in **FIG. 1** joined at the optional coupling **21**.

[0028] As noted above with respect to the description of **FIG. 1**, the vapor compression refrigeration system of the present invention includes a compressor. Compressors can be generally classified as reciprocating, rotary, jet, centrifugal, scroll, screw or axial-flow, depending on the mechanical means to compress the fluid, or as positive-displacement (e.g., reciprocating, scroll or screw) or dynamic (e.g., centrifugal or jet), depending on how the mechanical elements act on the fluid to be compressed. In one embodiment, the present inventive apparatus utilizes a centrifugal-type compressor.

[0029] A centrifugal compressor uses rotating elements to accelerate the refrigerant radially, and typically includes an impeller and diffuser housed in a casing. Centrifugal compressors usually take fluid in at an impeller eye, or central inlet of a circulating impeller, and accelerate it radially outward. Some static pressure rise occurs in the impeller, but most of the pressure rise occurs in the diffuser section of the casing, where velocity is converted to static pressure. Each impeller-diffuser set is a stage of the compressor. Centrifugal compressors are built with from 1 to 12 or more stages, depending on the final pressure desired and the volume of refrigerant to be handled.

[0030] The pressure ratio, or compression ratio, of a compressor is the ratio of absolute discharge pressure to the absolute inlet pressure. Pressure delivered by a centrifugal compressor is practically constant over a relatively wide range of capacities.

[0031] Positive displacement compressors draw vapor into a chamber, and the chamber decreases in volume to compress the vapor. After being compressed, the vapor is forced from the chamber by further decreasing the volume of the chamber to zero or nearly zero. A positive displacement compressor can build up a pressure, which is limited only by the volumetric efficiency and the strength of the parts to withstand the pressure.

[0032] Unlike a positive displacement compressor, a centrifugal compressor depends entirely on the centrifugal force of the high-speed impeller to compress the vapor passing through the impeller. There is no positive displacement, but rather what is called dynamic-compression.

[0033] A multi-stage impeller system may be used in a centrifugal compressor to improve compressor efficiency thus requiring less power in use. For a two-stage system, in operation, the discharge of the first stage impeller goes to the suction intake of a second impeller. Both impellers may operate by use of a single shaft (or axle). Each stage can build up a compression ratio of about 4 to 1; that is, the absolute discharge pressure can be four times the absolute suction pressure. Several examples of two-stage centrifugal compressor systems, particularly for automotive applications, are described in U.S. Pat. Nos. 5,065,990 and 5,363, 674,.

[0034] The pressure a centrifugal compressor can develop depends on the tip speed of the impeller. Tip speed is the speed of the impeller measured at its tip and is related to the diameter of the impeller and its revolutions per minute. Tip speed and impeller diameter can be estimated by developing fundamental relationships for refrigeration equipment that use centrifugal compressors. The torque an impeller ideally imparts to a gas is defined as

$$T=m^*(v_2 \cdot r_2 - v_1 \cdot r_1) \quad \text{Equation 1}$$

where

[0035] T=torque, Newton-meters

[0036] m=mass rate of flow, kg/sec

[0037] v_2=tangential velocity of refrigerant leaving impeller (tip speed), meters/sec

[0038] r_2=radius of exit impeller, meters

[0039] v_1=tangential velocity of refrigerant entering impeller, meters/sec

[0040] r_1=radius of inlet of impeller, meters

[0041] Assuming the refrigerant enters the impeller in an essentially axial direction, the tangential component of the velocity v**1**=0, therefore

$$T=m^*v_2^*r_2 \quad \text{Equation 2}$$

[0042] The power required at the shaft is the product of the torque and the rotational speed

$$P=T^*w \quad \text{Equation 3}$$

where

[0043] P=power, W

[0044] w=rotative speed, revolutions/second therefore,

$$P=T^*w=m^*v_2 \cdot r_2 \cdot w \quad \text{Equation 4}$$

[0045] At low refrigerant flow rates, the tip speed of the impeller and the tangential velocity of the refrigerant are nearly identical; therefore

$$r_2 \cdot w=v_2 \quad \text{Equation 5}$$

and

$$P=m^*v_2^*v_2 \quad \text{Equation 6}$$

[0046] Another expression for ideal power is the product of the mass rate of flow and the isentropic work of compression,

$$P=m^*H_i^*(1000\text{J/kJ}) \quad \text{Equation 7}$$

where

[0047] H_i=Difference in enthalpy of the refrigerant from a saturated vapor at the evaporating conditions to saturated condensing conditions, kJ/kg.

[0048] Combining the two expressions Equation 6 and 7 produces,

$$v_2^*v_2=1000^*H_i \quad \text{Equation 8}$$

[0049] Although Equation 8 is based on some fundamental assumptions, it provides a good estimate of the tip speed of the impeller.

[0050] The capacity of the centrifugal compressor is determined by the size of the passages through the impeller. This makes the size of the compressor more dependent on the pressure required than the capacity. Large centrifugal compressors typically operate at 3000 to 7000 revolutions per minute (rpm). Small scale centrifugal compressors (mini-centrifugals) are designed for high speeds, from about 20,000 RPM to about 75,000 RPM, and have small impeller diameter, typically less than about 0.15 meters (about 6 inches). The mini-centrifugal compressors which can be used with the present invention preferably operate at impeller speeds of 30,000 to 50,000 RPM and have impeller diameter of less than 0.10 meters (about 4 inches) where the vapor compression refrigeration system includes a compressor driven by an engine exhaust gas driven turbine

[0051] The power requirements for a mini-centrifugal compressor are not easily met in the present design for automobile engines. The electrical power available in current automobile design is about 14 volts. The new mini-centrifugal compressor requires electrical power of about 50 volts. The present invention allows the use of the mini-centrifugal compressor by utilizing the exhaust gas from an engine to drive a turbine that ultimately provides rotational power to the compressor impeller shaft as described above with respect to **FIG. 1**. Moreover, by making use of the present invention, the exhaust gas from the engine directly drives a turbine which powers an air-conditioning or refrigeration compressor, thereby eliminating intermediate energy transfer and conversion steps and yielding higher overall efficiency in the use of the engine's fuel.

[0052] A seal may be required around the compressor rotating shaft, in the event a common shaft is used or if there are separate shafts coupled by the optional mechanical coupling **21**, to prevent refrigerant from leaking out and to prevent air from leaking into the compressor. The present invention may include the use of a compressor shaft seal for sealing the interior of the compressor from ambient air. As the refrigerants used with the mini-centrifugal compressor are low-pressure refrigerants, the seal may be required to prevent air from leaking into the system, in particular, and causing deterioration in cooling performance. The shaft seal may be one of several designs and of several materials of construction, including, but not limited to steel, ceramic, or carbon against steel.

[0053] Alternatively, the turbine rotating shaft and compressor rotating shaft may be separate shafts coupled by means of a rotary magnetic coupling device. Magnetic couplings are used to transmit rotational motion without direct contact. Rotary couplings are principally used to eliminate the use of seals in rotating machines. Use of magnetic couplers improves the reliability and safety aspects of such machines because seals are prone to deterioration over time and cause leaks.

[0054] Rotary magnetic couplers used in the present invention will preferably be of co-axial configuration. The two halves of the coupler are mounted co-axially with each other and nested one within the other. The outer member is connected to the turbine rotating shaft and the inner member to the compressor rotating shaft. A cup shaped stationary member, mounted to the compressor body resides between the driver (turbine rotating shaft) and the follower (compressor rotating shaft) and separates the refrigerant fluid from the ambient environment and the turbine drive. Magnetic couplings of this type are available commercially.

[0055] Stationary refrigeration apparatus or stationary air-conditioning apparatus refer to the equipment used for cooling the air in a building, or cooling perishible goods such as foods, pharmaceutical materials, etc, in a conventional, non-mobile, non-vehicle mounted, system.

[0056] Such stationary refrigeration or air conditioning systems may be associated with CHP (Combined Heat and Power) systems, wherein a stationary internal combustion engine is used to drive an electrical generator. The waste heat produced by the engine may be recovered and used to perform work, by such means as a Rankine Cycle (steam engine) or Organic Rankine cycle (ORC). In a Rankine cycle, the heat is used to vaporize a liquid (an organic liquid in the case of an ORC), which in turn drives a turbine. The mechanical energy of the turbine may be used to drive an electricity generator, which runs a refrigeration or air-conditioning system.

[0057] An alternative embodiment of the present invention involves using an exhaust gas driven turbine as an alternative to using merely the exhaust heat from the internal combustion engine to provide heat to a building, drive an absorption cooling system, or drive an ORC process. An example would be use of this as part of a commercially available CHP system for homes, which are off the power grid, to generate electricity and recover exhaust heat to heat water or building air, and at the same time, use the exhaust gas driven compressor to provide cooling. The present invention is particularly useful in remote locations where access to electrical power is limited, if available at all.

[0058] Mobile refrigeration apparatus or mobile air-conditioning apparatus refers to any refrigeration or air-conditioning apparatus incorporated into a mobile transportation unit for the road, rail, sea or air. In addition, apparatus, which are meant to provide refrigeration or air-conditioning for a system independent of any moving carrier, known as "intermodal" systems, are included in the present invention. Such intermodal systems include "containers" (combined sea/land transport) as well as "swap bodies" (combined road and rail transport). The present invention is particularly useful for road transport refrigerating or air-conditioning apparatus, such as automobile air-conditioning apparatus or refrigerated road transport equipment.

[0059] The mini-centrifugal compressors of the present invention are capable of producing refrigeration capacity in the range from about 0.5 tons (1.7 kW) to about 25 tons for stationary applications. In the case of mobile systems, the refrigeration capacity produced is in the range of about 0.5 tons (1.7 kW) to about 3 tons (10.3 kW). Typically, about 1.2 tons (4.0 kW) to about 2.0 tons (6.8 kW) would be needed to cool an automobile passenger compartment. Greater capacity may be needed for many mobile refrigeration units such as road and rail refrigerated containers.

[0060] The refrigeration apparatus or air-conditioning apparatus of the present invention may additionally employ fin and tube heat exchangers, microchannel heat exchangers and vertical or horizontal single pass tube or plate type heat exchangers in the evaporator and/or the condenser.

[0061] Conventional microchannel heat exchangers may not be ideal for the new low-pressure refrigerants to be used in the refrigeration or air-conditioning apparatus of the present invention. The low operating pressure and density result in high flow velocities and high frictional losses in all components. In these cases, the evaporator and/or condenser design may be modified. Rather than several microchannel slabs connected in series (with respect to the refrigerant path) a single slab/single pass heat exchanger arrangement may be used. Therefore, a preferred heat exchanger for the refrigeration or air-conditioning apparatus evaporator and/or condenser of the present invention is a single slab/single pass heat exchanger.

[0062] Certain controls may be needed within the present refrigeration/air-conditioning apparatus to ensure optimum cooling performance. At high engine speed conditions, the exhaust gas output will increase thus driving the turbine to higher rotational speeds, which will increase the compressor impeller speeds as well. The wastegate valve (**53**) may be used to reduce the flow of exhaust gas to the turbine. This will reduce the speed at which the turbine spins and thus reduce the speed of the compressor impellers. Excessive impeller speeds above the design limits may cause damage to compressor internals, such as distortion of the impeller blades, which may result in generally reduced compressor performance and eventually, shorter compressor lifetime.

[0063] Under conditions of low engine exhaust gas flow, such as engine idle, the power requirements for adequate cooling may not be met. When the air-conditioning system is operating, the engine idle may be set to a higher speed, only during air-conditioner operation, such that there is enough power to provide the cooling needed.

[0064] A control system may be used that senses the compressor shaft rotational speed and adjusts the wastegate or engine idle speed as necessary to maintain optimum operation of the refrigeration or air-conditioning apparatus.

[0065] Under conditions when refrigeration or air-conditioning are not needed, for instance air-conditioning is seldom used in the cold winter months in an automobile passenger compartment, the compressor should be turned off. In this situation, the wastegate valve (**53**) will direct all engine exhaust gas to the conventional exhaust system.

Thus, no exhaust gas will reach the turbine and the compressor rotating shaft will be maintained in a stationary condition.

[0066] The engine exhaust gas driven turbine may become hot during operation. Transfer of this heat to the compressor is undesirable. Thus some means of thermally insulating the mini-centrifugal compressor from the engine exhaust gas driven turbine may be needed. Any usual means for insulation may be used, such as an insulating material, or use of air flow between the turbine and compressor.

[0067] The present invention further relates to a process to produce cooling comprising compressing a refrigerant in a mini-centrifugal compressor powered by an engine exhaust gas driven turbine; condensing said refrigerant; and thereafter evaporating said refrigerant in the vicinity of a body to be cooled.

[0068] The refrigerants for which this new refrigeration or air-conditioning apparatus are useful are hydrofluorocarbons (HFCs), including saturated and unsaturated compounds, fluoroethers (HFOCs), haloketones, hydrocarbons, chlorocarbons, alcohols, ketones, ethers, esters, N-(difluoromethyl)-N,N-dimethylamine, 1,1,1,2,2-pentafluoro-2-[(pentafluoroethyl)thio]ethane and combinations thereof. Depending upon the cooling capacity required, the boiling points of useful refrigerants may be anywhere from about −50° C. to about 75° C. Preferably, the refrigerants useful with the present invention possess boiling points in the range from about 0° C. to about 60° C., thus having lower vapor pressures at room temperature than the conventional CFC, HCFC and HFC refrigerants, such as R-12, R-22, or R-134a. The refrigerants also have low or zero ozone depletion potential and low global warming potential. Refrigerants useful with the new mini-centrifugal compressor in a refrigeration or air-conditioning apparatus include but are not limited to HFC-245fa (1,1,1,3,3-pentafluoropropane, $CF_3CH_2CHF_2$), HFC-365mfc ($CF_3CH_2CF_2CH_3$, 1,1,1,3,3-pentafluorobutane), HFC43-10mee (1,1,1,2,3,4,4,5,5,5-decafluoropentane, $CF_3CHFCHFCF_2CF_3$), HFC-63-14mcee (1,1,1,2,2,3,4,5,5,6,6,7,7,7-tetradecafluoroheptane, $CF_3CHFCHFCF_2CF_3$), HFC-1336mzz ($CF_3CH{=}CHCF_3$, 1,1,1,4,4,4-hexafluoro-2-butene), HFC-1429myz ($CF_3CF{=}CHCF_2CF_3$, 1,1,1,2,4,5,5,5-nonafluoro-2-pentene), HFC-1429mzy ($CF_3CH{=}CFCF_2CF_3$, 1,1,1), HFC-1438mzz ($CF_3CH{=}CHCF_2CF_3$, 1,1,1,4,4,5,5,5-octafluoro-2-pentene), HFC-153-10mzz ($CF_3CH{=}CHCF_2CF_2CF_3$, 1,1,1,4,4,5,5,6,6,6-decafluoro-2-hexene), HFC-153-10mczz ($CF_3CF_2CH{=}CHCF_2CF_3$, 1,1,1,2,2,5,5,6,6,6-decafluoro-3-hexene, HFC-153-10mmyzz ($CF_3CH{=}CHCF(CF_3)_2$, 1,1,1,4,5,5,5-heptafluoro-4-(trifluoromethyl)-2-pentene), PFBE (perfluorobutylethylene, $CF_3(CF_2)_3CH{=}CH_2$), PEIK (perfluoroethylisopropylketone, $CF_3CF_2C(O)CF(CF_3)_2$), PMIK (perfluoromethylisopropylketone, $CF_3C(O)CF(CF_3)_2$), HFOC-272fbEβγ ($CH_3OCH_2CHF_2$, 1,1-difluoro-2-methoxyethane), HFOC-347mmzEβγ ($CH_2FOCH(CF_3)_2$, 1,1,1,3,3,3-hexafluoro-2-(fluoromethoxy)propane), HFOC-365mcEγδ ($CF_3CF_2CH_2OCH_3$, 1,1,1,2,2-pentafluoro-3-methoxypropane), HFOC-356mmzEβγ ($CH_3OCH(CH_3)_2$, 1,1,1,3,3,3-hexafluoro-2-methoxypropane), HFOC467mmzEβγ ($CH_3CH_2OCF(CF_3)_2$, 2-ethoxy-1,1,1,2,3,3,3-heptafluoropropane), 2,2-dimethylbutane ($CH_3CH_2C(CH_3)_3$), cyclopentane (cyclo-$(CH_2)_5$—), trans-1,2-dichloroethylene ($CHCl{=}CHCl$), dimethoxymethane ($CH_3OCH_2OCH_3$), methyl formate ($HCOOCH_3$), $C_4F_9OCH_3$, and combinations thereof. These and other suitable refrigerants for use with the refrigeration or air-conditioning apparatus of the present invention are disclosed in U.S. Provisional Patent Applications 60/651,687, filed Feb. 9, 2005, and 60/732,581, filed Nov. 1, 2005; U.S. patent application Ser. No. 11/014,006, 11/014,000, 11/014,435, 11/014433, 11/014,438, 11/014,334, 11/013,901, 11/014,343, all filed Dec. 16, 2004; Ser. No. 11/063,178, 11/063,203, 11/063,040, and 11/062,975, all filed Feb. 22, 2005; Ser. No. 11/151,481, filed Jun. 13, 2005; Ser. Nos. 11/152,731, and 11/152,732, both filed Jun. 14, 2005; and Ser. No. 11/153,195, 11/153,168, and 11/153,804, all filed Jun. 15, 2005

[0069] A body to be cooled may be any space, location or object requiring refrigeration or air-conditioning. In stationary applications the body may be the interior of a structure, i.e. residential or commercial, or a storage location for perishables, such as food or pharmaceuticals. Numerous mobile systems are described earlier in defining mobile refrigeration apparatus and mobile air-conditioning apparatus.

[0070] The present invention further relates to a method for providing power to a compressor within a vapor compression refrigeration system of refrigeration apparatus or air-conditioning apparatus, said method comprising coupling the compressor to an engine exhaust gas driven turbine.

[0071] The present invention further relates to a method for controlling impeller speed in a compressor within a vapor compression refrigeration system of a refrigeration apparatus or air-conditioning apparatus, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

[0072] The present invention further relates to a method for controlling the cooling capacity for a refrigeration or air-conditioning apparatus, wherein said apparatus comprises a vapor compression refrigeration system including a compressor powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

[0073] The present invention further relates to a method for controlling impeller speed in a compressor within a vapor compression refrigeration system of a refrigeration or air-conditioning apparatus, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the engine idle speed during air-conditioner operation.

[0074] The present invention further relates to a method for controlling the cooling capacity for a refrigeration or air-conditioning apparatus, wherein the apparatus comprises a vapor compression refrigeration system including a compressor powered by an engine exhaust gas turbine, said method comprising varying the engine idle speed during refrigeration or air-conditioning apparatus operation.

[0075] The present invention further relates to a method for controlling surge in a compressor within a vapor compression refrigeration system of a refrigeration apparatus or air-conditioning apparatus, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising maintaining minimum flow through the compressor by controlling recycle of the refrigerant from the discharge to the suction of the compressor. Compressor surge is a condition that must be avoided due to potential damage to the compressors. If the forward flow through the compressor can no longer be maintained due to an increased pressure differential across the compressor, a momentary flow reversal may occur. The valve (**51**, in **FIG. 1**) allows some portion of the flow out of the compressor to be diverted back to the compressor suction thus balancing the pressure across the compressor under the surge point.

[0076] In all of the above-described methods, it is preferable to use a centrifugal compressor, and more preferably a mini centrifugal compressor.

EXAMPLE

[0077] In this Example, a refrigeration apparatus using PEIK (perfluoroethylisopropylketone) as refrigerant to produce cooling capacity of approximately 1.5 tons at an impeller speed of 40,000 rpm is illustrated. The conditions assumed for this example are:

Evaporator temperature	40.0° F. (4.4° C.)
Condenser temperature	110.0° F. (43.3° C.)
Liquid subcool temperature	10.0° F. (5.5° C.)
Return gas temperature	75.0° F. (23.8° C.)
Compressor efficiency is	80%

[0078] These are typical conditions under which a mini centrifugal compressor performs. Table 1 below shows the enthalpy of the refrigerant gas as it leaves the evaporator, (H Evaporator Out), the enthalpy of the refrigerant gas as in enters condenser (H Condenser In), the change in enthalpy between the evaporator and compressor, (Delta Hi), and the change in enthalpy between the evaporator and compressor multiplied by 0.8). The table below also shows theoretical tip speed and impeller diameter for a refrigeration apparatus.

TABLE 1

Refrigerant	H Evaporator Out (Btu/lb)	H Condenser In (Btu/lb)	Delta Hi (Btu/lb)	Delta Hi * 0.8 (Btu/lb)	Delta Hi * 0.8 (KJ/Kg)	Tip Speed V_2 (meter/sec)	Diameter (meters)	Diameter (inches)
PEIK	38.37	49.92	11.55	9.2	21.5	146.6	0.0700	2.76

What is claimed is:

1. A refrigeration or air-conditioning apparatus comprising a vapor compression refrigeration system having a refrigerant circulating therethrough, where the vapor compression refrigeration system includes a compressor driven by an engine exhaust gas driven turbine.

2. The apparatus of claim 1, said apparatus being a mobile refrigeration apparatus or mobile air-conditioning apparatus.

3. The apparatus of claim 1, said apparatus being a stationary refrigeration apparatus or stationary air-conditioning apparatus.

4. The apparatus of claim 1, wherein the compressor is a centrifugal compressor.

5. The apparatus of claim 1, wherein the vapor compression refrigeration system includes a condenser having an inlet connected via a pipeline to the compressor.

6. The apparatus of claim 5, wherein the vapor compression refrigeration system further includes an expansion device having an inlet connected via a pipeline to the condenser.

7. The apparatus of claim 6, wherein the vapor compression refrigeration system further includes an evaporator having an inlet connected via a pipeline to the expansion device and an outlet connected to the compressor.

8. The apparatus of claim 1, wherein the compressor and the turbine include a common rotating shaft extending therebetween for driving both the compressor and the turbine.

9. The apparatus of claim 1, wherein the compressor includes a rotating shaft, and the turbine includes a rotating shaft, and the compressor rotating shaft is coupled to the turbine rotating shaft of said engine exhaust gas driven turbine by a rotary magnetic coupling.

10. The apparatus of claim 4, wherein the compressor is a mini centrifugal compressor.

11. A method for controlling the compressor surge within a refrigeration or air conditioning apparatus comprising a vapor compression refrigeration system including a compressor coupled to an engine exhaust gas driven turbine, the method comprising maintaining minimum flow of refrigerant through the compressor by controlling recycle of the refrigerant from the discharge to the suction of the compressor.

12. A method for providing power to a compressor within a refrigeration or air-conditioning apparatus, said apparatus including a vapor compression refrigeration system, said method comprising coupling the compressor to an engine exhaust gas driven turbine to power the compressor.

13. A method for controlling impeller speed in a compressor within a refrigeration or air-conditioning system, said apparatus including a vapor compression refrigeration system, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

14. A method for controlling the cooling capacity for a refrigeration or air-conditioning apparatus, wherein the apparatus comprises a vapor compression refrigeration system including a compressor powered by an engine exhaust gas driven turbine, said method comprising varying the amount of exhaust gas allowed to enter the turbine.

15. A method for controlling impeller speed in a compressor within a refrigeration or air-conditioning apparatus, said apparatus including a vapor compression refrigeration system, wherein the compressor is powered by an engine exhaust gas driven turbine, said method comprising varying the engine idle speed during apparatus operation.

16. A method for controlling the cooling capacity for a refrigeration or air-conditioning apparatus, wherein the apparatus comprises a compressor powered by an engine exhaust gas turbine and connected to a vapor compression refrigeration system, said method comprising varying the engine idle speed during apparatus operation.

17. The method of any of claims **11-16**, wherein the compressor is a mini-centrifugal compressor.

18. A process for producing cooling comprising compressing a refrigerant in a mini-centrifugal compressor powered by an engine exhaust gas driven turbine; condensing said refrigerant; and

thereafter evaporating said refrigerant in the vicinity of a body to be cooled.

19. The process of claim 18, further including recycling the refrigerant back to the compressor after the evaporating step.

20. The process of claim 13, wherein said body to be cooled is an automobile passenger compartment or a stationary structure.

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