SPECIFICATION

Refrigeration system utilizing a gaseous refrigerant bypass

The present invention relates generally to a refrigeration system and more particularly to a refrigeration system having a high pressure gaseous refrigerant bypass which introduces high pressure gaseous refrigerant into the compressor resulting in increased heating capacity and faster pressure rise.

A large variety of refrigeration systems are known based on a heat pump type refrigeration cycle, a cycle which is reversibly assembled so that the heat discharged at a condenser can be used as a heating resource.

A drawback to the heat pump type refrigeration system results from the insufficient heat supply capacity which often occurs when the outdoor temperature gradually decreases or when it is very cold outside.

Since users need larger heating capacity as the temperature drops the insufficiency of heat supply becomes a serious technical problem.

The heat supply capacity heretofore has been supplemented by electric heaters incorporated in the interior unit near the heat exchanger, thus producing the necessary amount of heat.

Such an electric heater, however, consumes a large quantity of electricity, so that the power supply must be at least as large as the total amount of the supply needed for the refrigeration system itself and for the electric heater.

Furthermore, care must be taken to provide safety measures in the interior unit against possible dangers, such as a fire set off by an electrical leakage or by the direct contact of the electrical heater with inflammable components inside the interior unit.

On the other hand, in the field of air-conditioning which uses a refrigeration cycle, a skill has been developed for preventing overheating of the compressor by providing the cylinder thereof with a small hole, through which the liquid refrigerant is injected and evaporated.

The introduced liquid refrigerant, while instantly evaporating upon entering the compression chamber, absorbs enthalpy from the refrigerant which is being compressed and soon reaches a normal temperature.

The temperature of the refrigerant which comes out of the compressor, therefore, can be subject to control depending on the amount of injected liquid refrigerant.

Although this injection type compressor can at any rate contribute to the decrease in the temperature of the discharged refrigerant for its cooling function, it by no means helps either to quicken the pressure rise or to add to the heating capacity available.

Accordingly, the present invention seeks to improve the heating capacity of the refrigeration system.

The invention also seeks to provide for the refrigeration system a faster rise for the heating capacity which leads to the prevention of initial cold air flow from the interior unit. The invention also seeks to provide a compact compressor for the refrigeration system. The invention also seeks to provide a refrigeration system with an improved compressor which creates an increased heating capacity.

The invention also seeks to provide a refrigeration system which can operate with a decreased capacity when the load is light and with an increased capacity when the load is heavy.

According to this invention we provide a refrigerant system of the type including a hermetic type compressor having a motor connected thereto, a condenser, an expansion device, an evaporator and interconnection means connected to form a refrigerant flow path therethrough, said system further including a first gaseous refrigerant bypass through which a portion of the refrigerant having at least a condensing pressure is able to be injected into said compressor, wherein injected refrigerant increases the load of said motor thereby adding to the heat generated therefrom so as to supply the refrigerant with said additional heat.

The invention will now be further described by way of example with reference to the following description when taken in connection with the drawings, in which:

Figure 1 is a schematic view of the fundamental refrigeration cycle having a gaseous refrigerant bypass;

Figure 2 is a refrigeration cycle of this invention applied to a heat pump type refrigeration system;

Figure 3 shows a cross-sectional view of the compressor employed in the refrigeration system of the invention which is provided with an injection hole formed on the cylinder thereof;

Figure 4 shows the characteristics of the pressure P in the compressing chamber to the rotation angle θ of the roller shown in Figure 3;

Figure 5 shows the characteristics of the heating capacity Q of this invention as compared with that of the conventional type to the outdoor temperature T;

Figure 6 shows on the vertical scale the faster rise of the heating capacity Q of this invention as compared with that of the conventional prior art to the time t on the horizontal scale;

Figure 7 is the pressure (P) — enthalphy (i) diagram which shows the characteristics for the refrigeration cycle of the present invention;

Figure 8 shows another example of the present invention applied to a heat pump type refrigeration system which has a flow control means in the gaseous refrigerant bypass;

Figure 9 shows another example of the present invention applied to a heat pump type refrigeration cycle having a pair of gaseous refrigerant bypasses for both its cooling cycle line and heating cycle line;

Figure 10 shows another example of the present invention applied to a heat pump type refrigeration system having a liquid injection line;

Figure 11 shows another example of the

present invention applied to a heat pump type refrigeration system including a liquid-gas refrigerant separator placed between the condenser and the evaporator from which separated gaseous refrigerant is supplied into the compressor;

Figure 12 shows another example of the present invention applied to a heat pump type refrigeration system having a gaseous bypass which is joined by another high pressure, liquid refrigerant flow bypass;

Figure 13 shows another example of the present invention applied to a heat pump type refrigeration system in which the mixture of gaseous refrigerant and liquid refrigerant can be injected into the compressor;

Figure 14 shows another example of the present invention applied to a heat pump type refrigeration system in which the mixture of gaseous refrigerant and liquid refrigerant is injected into the compressor; and

Figure 15 shows another example of the present invention applied to a heat pump type refrigeration cycle in which the gaseous refrigerant bypass is used as a defrosting heater.

Referring now to Figure 1 the refrigerating system comprises a compressor 1 which compresses refrigerant and pumps it out firstly to a condenser 3 connected therewith. The condenser 3 is connected to the compressor 1 which introduces the compressed refrigerant for condensation. Condenser 3 then connects with a capillary tube 4 functioning as an expanding means. An evaporator 5 is connected to the capillary tube 4 at which refrigerant evaporates. The vaporized refrigerant flows out of the evaporator 5 and returns to the compressor 1. In the meantime a gaseous refrigeration bypass 7 is additionally provided between the condenser 3 and the compressor 1. In order to control the refrigerant flow an electromagnetic valve 6 is attached therein.

It should be understood that in operation part of the refrigerant discharged from the compressor diverges into the bypass line 7, flows through the control valve 6 and then returns to the compressor 1 through injection hole 12. Clearly the amount of diverged refrigerant flow can be controlled depending on the operating conditions.

The same skill can be applied to a heat pump type refrigeration system as shown in Figure 2. The like characters therein indicate like parts as described above. A four-way reversing valve 2 is utilised to direct the refrigerant from the compressor by way of conduit 21 into either of the two heat exchangers and simultaneously direct the output from the other coil and into the input of the compressor 1. An outdoor heat exchanger 3*a* functions both as a condenser when the cycle is in the cooling mode and as an evaporator when the cycle is in the heating mode. An indoor heat exchanger 5*a* serves both as an evaporator when the cycle is in the cooling mode and as a condenser when the cycle is in the heating mode. In this embodiment also a gaseous refrigerant bypass 7 is provided between the injection hole 12 of the compressor 1 and the indoor heat exchanger 5*a*. Note that the solid line arrow denotes the refrigerant flow in the cooling mode and the broken line arrow denotes the refrigerant flow in the heating mode.

A control valve 6 is provided in the bypass line 7 in order to ensure stable return supply of the gas refrigerant, so that in operation, in proportion to the extent that the control valve 6 is open, gas refrigerant diverges at the junction between the control valve 6 and the indoor heating coil 5*a* and flows back to the inlet 12 of the compressor 1.

In Figure 3 there is shown a schematic cross sectional view of the compressor 1 shown in Figure 2.

A particular preferred feature of the present invention will now be described to enable one to perform the needed effect for the compressor 1. In order to define a compression chamber 9 a cylinder 9′ is provided inside an interior casing 20. In the centre of the cylinder 9′ is placed a shaft 13, which is connected with rotating means (not shown) and the shaft 13 is surrounded by a roller 14 in a close relationship with the inner surface of the cylinder 9′. As the shaft 13 and the roller 14 are eccentric, the roller 14 moves around in such a manner in the compression chamber 9 that it touches and presses the inner surface of the cylinder at one variable point, namely, the point of contact circulates the cylinder's inner surface continuously, dividing the compression chamber 9 into two chambers, a suction side chamber and a discharge side chamber. This can be achieved with the aid of a blade 15 which is provided in contact with the roller 14. The blade 15 is inserted in a recess 8 in which a supporting spring 16 is accommodated, and moves reciprocally along a slot 15 without failing to contact and press the roller 14.

In the embodiment shown in Figure 3, the shaft 13 and the roller 14 rotate in the counterclockwise direction. But due to the slip between the shaft 13 and the roller 14, the rate of rotation of the roller 14 is considerably less than that of the shaft 13. For the compressor to suck in the refrigerant the inner wall of the cylinder 9′ is provided with a suction port 10 and a discharge port 11. The suction port 10 is in communication with the evaporator through the four-way reversing valve 2. The evaporated refrigerant enters the compression chamber 9 by way of the suction port 10 while the roller 14 is, at the same time, compressing the refrigerant. The roller 14 keeps on compressing the refrigerant until it discharges it from the discharge port 11. The discharge port 11 is aimed at correctly discharging the compressed refrigerant at a predetermined pressure level and at obviating adverse current. For this purpose the discharge port 11 is provided with a discharge plate valve 17 made of an elastic metal member. The plate valve 17 is designed to open and to allow the refrigerant out. To avoid malfunctioning of the plate valve 17, a check plate 18 is also attached in such a way that, during

operation. it controls the lift of the plate valve 17. The plate valve 17 and the check plate 18 are secured together at their ends to the cylinder 9' by a bolt in a conventional manner.

As can be seen from Figure 3, no valve is provided for the suction port 10. To attain the efficiency envisaged by the present invention, the bottom wall of the cylinder 9' is provided with a aperture or a hole 12. The aperture 12 is in communication with the high pressure gaseous refrigerant zone, namely, as shown in Figure 1, the aperture 12 communicates with the refrigerant conduit connecting the compressor 1 and the condenser 3. Of course the aperture can be connected to somewhere in the first half of the condenser so far as it is in communication with the high pressure gaseous refrigerant.

With the application of adequate pressure, part of the high pressure gaseous refrigerant returns to the compression chamber 9 during the compressing process through the aperture 12. The size, shape and the location of the aperture 12 may be individually determined.

Let us assume that the roller is in the "0°" position when the blade 15 is most deeply submerged in the recess 8, "180°" when the blade 15 protrudes to the maximum and "360°" when the blade 15 returns to the most deeply submerged position after a single rotation of the roller 14. Desirably the aperture 12 is so located that it begins to open immediately after the roller 14 passes its "20°" position at which the suction port 10 is located, and completely closes when the roller 14 comes somewhere between "180°" and "210°" position. For example, the position of the aperture 12 is set to be "290°" of the roller position and at an adequate distance from the cylinder wall, if aimed at beginning to open at "20°", opening in full at "110°" and again completely closing at "200°". So it is very clear that during a single rotation of the roller 14 the refrigerant flows in from the suction port 10, is compressed, and is discharged from the discharge port 11 into the discharge chamber 19. In the meantime high pressure refrigerant comes in through the aperture 12 of its own accord when it is opened, and at this time the pressure inside begins to rise sharply till it reaches discharge level. All the mixed refrigerant flows out of the discharge valve 17, while part of the discharged refrigerant returns to the compression chamber 9 by way of the aperture 12, thus helping to produce a larger compression work for the compressor 1.

The other characteristics of the aperture 12 should also be determined individually. The shape, for example, is not necessarily circular. It may be oval, semi-triangular or crescent shaped.

Further particular advantages obtainable by the invention are clearly demonstrated with reference to the following Figures 4 to 6.

In Figure 4 the characteristics of the pressure in the compression chamber (P) is shown plotted against the roller position (θ) for the current invention (line ii) in a clear comparison with the results for a conventional prior compressor (line i).

Thus, in the case of the conventional compressor the pressure starting at θ_1 rises relatively slowly, reaches the maximum pressure at θ_3, then levels off. In contrast, however, the pressure rise for the current invention (line ii) is steeper than that of the prior art (line i). The reason for this is that, just after the suction process is over at the pressure V_o, high pressure refrigerant is introduced into the compression chamber through the aperture 12, causing the pressure to rise more rapidly to the maximum pressure Vc at θ_2. θ_2 falls at approximately 200° of the roller's position. Obviously, it should be understood that, in Figure 4, the oblique-lined area bounded by the line (i) and the line (11) corresponds to the amount of increase in the work to be given the compressor: consequently, the work that the compressor of this invention is expected to perform will become greater than that of the prior compressors. The refrigerant pressure continues to rise so long as the aperture 12 remains open, but stops increasing when the discharge valve 17 opens to let out the compressed refrigerant. The maximum pressure Vc remains at approximately the same level once the discharge valve 17 has been opened at about "200°" of the roller position, since the opening timing of the discharge valve 17 is so adjusted that the valve opens soon after the aperture 12 is completely closed by the bottom surface of the cylinder 14.

In Figure 5 there is shown the characteristics of the heating capacity Q in relation to the outdoor ambient temperature T. The heating capacity hitherto has been such that, with the fall of the outdoor ambient temperature T, the heating capacity Q lowers, as can be seen from the broken line (i) in Figure 5. This has presented a major technical problem because normally the lower the ambient temperature becomes the more heating capacity is needed.

According to this invention, as the compressor is designed to achieve more work, when the aperture 12 is open and gas injection is added at To, it offers substantially more heating capacity as can be seen from the solid line (ii) in Figure 5, and at the same time, as the compressor is able to warm up in substantially less operating time (denoted by the reference letter *t* in Figure 6), there is no problem resulting from what is called the "cold draft", i.e. an initial cold air flow from the indoor heat exchanger caused by prolonged warm-up.

Naturally a person skilled in the art may easily smooth the characteristic of the Q—T curve in Figure 5 in the design of a refrigeration system simply by controlling with the control valve the amount of refrigerant injected into the compressor 1.

In Figure 7 a further explanation is given using a pressure-enthalpy diagram. In winter the refrigeration starts with the cycle formation drawn in the dotted-lined trapezoid A'E'F'G'. Here the lines JH and J'H', respectively represent the saturated liquid and saturated vapor lines of a

refrigerant. Point F' indicates the state of a kilogram of the refrigerant upon issuance from the condenser of a simple conventional refrigeration system; and point G' represents the state of said refrigerant after constant enthalpy expansion along line F'G'. The liquid refrigerant passes through the evaporator, changing its condition along line G'A'. The evaporated gas refrigerant is then recompressed along line A'E', necessitating an expenditure of work per kilogram which, in terms of heat units, is equal to the projection K—K' on the enthalpy axis. The compressed refrigerant is then returned through the condenser to the state F' and the cycle is ready to repeat.

During operation, as the compressor gradually becomes warmed up, the refrigerant cycle turns to a new higher formation, namely the one shown by the trapezoid A E F G. Since in the early stage of the operation the aperture 12 is opened, forming the gas refrigerant bypass cycle, the time needed to warm up the compressor is comparatively shorter. Because the high pressure refrigerant at point E, which has just been discharged from the compressor, again enters and joins the refrigerant in the compression process in the compression chamber, the resultant refrigerant reaches a higher pressure than at the previous stage. So after every cycle the pressure of the discharged refrigerant becomes progressively higher, but within a certain operation period the pressure will converge into a balanced pressure level, point D. In addition, as the refrigerant being compressed is joined by the high-enthalpy refrigerant, the total enthalpy of the discharged refrigerant per kilogram will increase as from the point B to the point C. The compressor will continue to compress the mixed refrigerant as from the point C to point D.

Finally the refrigeration cycle will settle down to the formation ABCDEFGA. Note that there are numerous transitional formations between the initial one and the finally stabilized one. Also note that the amount of work added to the refrigeration cycle is represented by the segment ED. In the heating-mode operation this additional enthalpy will result in the increase in the form of the heat which can be retrieved at the condenser. By controlling the amount of refrigerant which directly returns to the compressor a greater heat capacity will be available per kilogram of refrigerant.

In Figure 8 another embodiment of this invention is explained. In this embodiment the gaseous refrigerant bypass line 7 is further provided with a capillary tube 22. This gives the bypass line a resistance that can ensure a stable expansion of compressed refrigerant.

Figure 9 shows another example of the present invention. In this example, another gas refrigerant bypass 23 is also provided, which links the outdoor coil 3*a*, when the heat exchanger is functioning as a condenser in the cooling mode, to the injection aperture 12 of the compressor 1. The gas refrigerant bypass 23 is provided with a control valve 24 so as to restrict the return of refrigerant back into the injection hole 12 of the compressor 1. Since the gas bypass line 23 connects the outdoor coil 3*a* to the compressor 1, gas refrigerant can also be injected into the compressor when the refrigeration cycle is in the cooling mode.

Further referring to Figure 9 a solid-line arrow denotes the refrigerant flow for the cooling mode and a dotted-line arrow for the heated mode.

In the heating mode, when both of the control valves 6, 24 are closed, the compressor 1 discharges compressed refrigerant to the four-way reversible valve 2 and to the indoor coil 5*a* as in a normal manner. At the indoor coil 5*a* the refrigerant releases heat and coldness. The condensed refrigerant then passes the capillary tube 4 and expands, part of the refrigerant becoming flash gas. The liquid gas then proceeds to the outdoor coil 3*a* where it absorbs heat and evaporates. The evaporated gas as a final step returns to the compressor 1 by way of the four-way valve 2.

In the heating mode, when the control valve 24 is closed and the control valve 6 is open, though the compressor discharges compressed refrigerant to the indoor coil 5*a* by way of the four-way reversible valve 2 in the same manner as aforementioned, part of the compressed refrigerant digresses from the main flow and returns to the compressor 1 through the control valve 6. This, as we have so far seen, will add to the heating capacity that the refrigeration cycle affords, in addition to particularly improving the pressure rise of the system. In the heating mode, when the load for the compressor is very heavy, which often occurs when the ambient temperature is relatively high for a heating operation, the control valve 24 should be open and the control valve 6 closed. In this case, since the aperture 12 is in communication with the low pressure area, a large part of the compressed refrigerant escapes therefrom during the compressing process, thus alleviating the excessive load.

On the other hand in the cooling mode, when both the valves are closed, a normal type operation can be achieved. The compressor 1 first sends compressed refrigerant to the outdoor coil 3*a* via the four-way valve 2, then refrigerant is directed to the capillary tube 4 where it expands and then proceeds to the indoor coil 5*a*. The refrigerant evaporates as it absorbs heat from the surrounding air while passing through the indoor coil 5*a*, then returns to the compressor through the reversible valve 2. In the cooling mode, when the control valve 24 is open and the valve 6 closed, part of the compressed refrigerant discharged from the compressor directly returns to the compressor 1 through the control valve 24. This will prompt the compressor to rise up to normal capacity faster than would otherwise be the case.

Also in the cooling mode, when the ambient temperature is not so high as to need much cooling supply, the valve 24 should be closed and the valve 6 opened. In such as case, the high-pressure compressed refrigerant escapes from the aperture 12 and joins the refrigerant which has

pased through the indoor coil 5*a* and has become low-pressure so that the load for the compressor will be markedly reduced while maintaining the cooling capacity at a certain level. The cooling capacity available can be controlled by the control valve 6 from the maximum level to nearly half or lower depending on the amount of refrigerant escaping from the compression chamber and on how far the valve 6 is open.

Figure 10 shows an additional modification to the embodiment shown in Figure 9, in which a capillary tube is divided into two separate expanders 25 and 26, and a refrigerant bypass 28 is provided between the junction of the bypass 6, 23 and the junction between the two expanders 25 and 26. The two expanders 25, 26 in cooperation with each other can function as a single capillary tube. In either mode of operation the refrigerant expands to an intermediate pressure after passing through one of the expanders 25 or 26. A mixture of flash gas and liquid refrigerant flows into the capillary 27 and expands to a low pressure thus forming mostly liquid refrigerant. Because the refrigerant bypass line 28 assures a constant supply of liquid refrigerant to the compressor, if the hot, high pressure refrigerant directly returns to the compressor or if the compressor is about to be subject to excessive heat, the injected liquid refrigerant immediately evaporates in the compression chamber and absorbs heat and prevents overheating of the compressor.

Figure 11 shows a further embodiment of the invention. Two capillary expanders 25, 26 are provided, between which an accumulator 29 is connected. A refrigerant bypass 7 links the compressor 1 to the indoor coil 5*a*, so that during the heating mode operation it can return compressed refrigerant directly to the compressor 1 and during cooling mode operation it can release the refrigerant in the compression process to the low pressure zone. The accumulator 29 and the bypass line 7 are associated with the bypass line 28 which has a control valve 30 therebetween. Gas refrigerant which has been separated from liquid refrigerant in the accumulator 29 flows through the refrigerant bypass 28. Since the accumulator 29 is in the middle of the two expanders 25 and 26, both the gas refrigerant and the liquid refrigerant have intermediate pressure and an intermediate temperature. The compressor 1 allows the gas refrigerant into the compression chamber. The amount of gas is controlled by the valve 30 and the injected refrigerant absorbs the enthalpy from the ambient refrigerant and helps to decrease the temperature.

A person skilled in the art could without difficulty introduce liquid refrigerant from the accumulator 29 by submerging the end opening of the refrigerant bypass 28 in the refrigerant reservoir, or he could mix both gas refrigerant and liquid refrigerant into an appropriate moist gas and inject it into the compression chamber, resulting in the decrease in the temperature of the compressor 1. The temperature inside the accumulator can be set to a desired level by giving the appropriate balance to the expanders 25, 26.

Figure 12 diagrammatically illustrates a refrigeration cycle which comprises a set of gas bypasses 33, 34, 36 and 38 employing control valves 37, 35 and a pair of check valves 31 and 32. A gas refrigerant bypass 38 is provided immediately after the compressor 1 and before the four-way reversible valve 2. The bypass is provided with a control valve 37. A pair of bypass lines 33, 34 are provided so as to form a bypass bridge over the capillary tube 4 and each comprises a check valve therein. Both of the check valves 31, 32 are directed to the compressor 1 via the bypass 36 which connects both ends of the check valves 31, 32 and the compressor 1.

The whole cycle is designed to operate in a manner described below.

In the heating mode operation the control valve 37 should be opened and the control valve 35 should be desirably shut at an earlier stage of the operation, so as to increase the compression work and to produce more heat and prompt high heating capacity. When a normal operation is desired, both of the control valves 37, 35 are closed. The refrigerant follows the normal cycle, namely, in the heating opejtion in the following order, the compressor 1, the four-way reversible valve 2, the indoor coil 5*a*, the capillary tube 4, the outdoor coil 3*a*, the four-way valve 2 and finally the compressor 1, and in the cooling mode, vice versa.

If the compressor 1 has become rather overheated the control valve 35 should be opened. Then the refrigerant which has passed through the first heat exchanger working as a condenser will flow by way of either of the check valves, the control valve 35 and will flow into the compressor 1. The control valve 37 on this occasion should be closed in order to prevent the discharged, high pressure refrigerant from obstructing the condensed refrigerant flowing into the compression chamber, as the discharged refrigerant has a higher pressure than it does after it has passed through the condenser.

Figure 13 shows another modified embodiment of the invention. A gaseous refrigerant bypass 7 connects the compressor to one end of the indoor coil 5*a* which works as the condenser when the entire cycle operates as a heating apparatus. The refrigerant bypass 7 is provided with a control valve 6 for controlling the amount of refrigerant bypass flow.

Another gas bypass line 40 is provided for connecting the compressor 1 and the exit of the indoor coil 5*a* when it functions as a condenser in the heating mode operation. This arrangement of elements enables the mixture of gaseous refrigerant from the line 7 and condensed refrigerant from the line 40 to be injected into the compression chamber of the compressor 1 in order to enlarge the compression work of the compressor but without creating excessive heat.

According to this embodiment the compressor

1 can be unloaded in the cooling mode operation by discharging halfway-compressed refrigerant through the control valve 6 to the exit of the indoor coil to be operated as the evaporator in the low pressure zone, so that the cooling capacity can be reduced to a desired level.

Figure 14 represents a further modification of the refrigeration cycle shown in Figure 13. The capillary tube which in Figure 13 appears as 39 is separated into two capillary tubes 41 and 42. The capillary tube 42 is provided with a bypass conduit with a control valve 43. In this example the pressure and flow of the liquid refrigerant can be controlled depending on the extend to which control valve 43 is open.

Figure 15 illustrates a heat-pump type refrigeration cycle employing a defrost means. Near the outdoor coil 3*a* a high-temperature defrost conduit 45 is provided which forms part of the gaseous refrigerant bypass 7. Because discharged refrigerant from the compressor 1 has a large enthalpy and a high temperature, it can give the outdoor coil 3*a* a considerable amount of heat, so that, if in a heating mode operation part of the outdoor coil 3*a* is frozen, it can be defrosted while continuing to function as an evaporator. Two capillary tubes 39, 44 are provided in parallel across the indoor coil 5*a*, and a control valve 6 is connected between the capillary tubes linking the defrost conduit 45 to one end of the indoor coil 5*a*. These constitute preferred features of the invention. Since the capillary tube 39 is connected between the compressor 1 and the conduit between the indoor coil 5*a* and the expander 4, condensed refrigejnt flows into the defrost conduit 45 and then into the compressor 1. This is advantageous to the compressor 1 because a certain amount of condensed liquid refrigerant flows into the compression chamber of the compressor 1 and evaporates, thus avoiding overheating.

The defrost conduit 45 should preferably be placed adjacent the outdoor coil 3*a*. Since ice is usually formed at the bottom part of the outdoor coil 3*a* the defrost conduit 45 should also be provided for the bottom half thereof. This arrangement can ensure efficient thawing for the indoor coil 5*a*.

CLAIMS

1. A refrigeration system of the type including a hermetic type compressor having a motor connected thereto, a condenser, an expansion device, an evaporator and interconnection means connected to form a refrigerant flow path therethrough, said system further including a first gaseous refrigerant bypass through which a portion of the high pressure refrigerant having at least a condensing pressure is able to be injected into said compressor, wherein injected refrigerant increases the load of said motor thereby adding to the heat generated therefrom so as to supply the refrigerant with said additional heat.

2. A refrigeration system according to claim 1 wherein said first gaseous refrigerant bypass has one end connected to the compression chamber of said compressor and the other end in communication, with the high pressure gaseous refrigerant having at least a condensing pressure and a portion of high pressure gaseous refrigerant from the second end is able to be injected into said compression chamber of said compressor through said one end.

3. A refrigeration system according to claim 1 or 2, wherein said first gaseous refrigerant bypass is provided with a control device therein for controlling the amount of refrigerant flow therethrough.

4. A refrigeration system according to claim 3 in which said control device comprises a capillary tube.

5. A refrigeration system according to claim 1, 2, 3 or 4 comprising an indoor coil which is connected to said compressor and functions as a condenser in a heating mode operation, an expansion device connected to said indoor coil for expanding the refrigerant, an outdoor coil which is connected to said expansion device and functions as an evaporator in a heating mode operation, a reversing device which is connected to said compressor for changing the direction of the refrigerant flow and a first high pressure gaseous refrigerant bypass having one end connected to the compression chamber of said compressor and the other end in communication with the high pressure gaseous refrigerant zone including said indoor coil.

6. A refrigeration system according to claim 5 which further comprises a second gaseous refrigerant bypass having one end in communication with said compression chamber of said compressor and the other end in communication with the high pressure gaseous refrigerant zone including said outdoor coil.

7. A refrigeration system according to claim 6 in which said second gaseous refrigerant bypass has a control device for controlling the refrigerant flow therethrough.

8. A refrigeration system according to claim 7 in which said control device comprises an electric valve for closing and opening said second gaseous refrigerant bypass.

9. A refrigeration system according to any one of claims 5 to 8 further comprising a third refrigerant bypass which has one end connected between said indoor coil and said expansion device and the other end connected to said compressor whereby a mixture of refrigerant from said high pressure gaseous refrigerant bypass and from said third refrigerant bypass is to be injected into said compressor.

10. A refrigeration system according to claim 9 in which said third refrigerant bypass comprises a control device for controlling the refrigerant flow therethrough.

11. A refrigeration system according to claim 10 in which said control device for said third refrigerant bypass comprises a capillary tube.

12. A refrigeration system according to claim 9, 10 or 11 which further comprises a plurality of

1 can be unloaded in the cooling mode operation by discharging halfway-compressed refrigerant through the control valve 6 to the exit of the indoor coil to be operated as the evaporator in the low pressure zone, so that the cooling capacity can be reduced to a desired level.

Figure 14 represents a further modification of the refrigeration cycle shown in Figure 13. The capillary tube which in Figure 13 appears as 39 is separated into two capillary tubes 41 and 42. The capillary tube 42 is provided with a bypass conduit with a control valve 43. In this example the pressure and flow of the liquid refrigerant can be controlled depending on the extend to which control valve 43 is open.

Figure 15 illustrates a heat-pump type refrigeration cycle employing a defrost means. Near the outdoor coil 3*a* a high-temperature defrost conduit 45 is provided which forms part of the gaseous refrigerant bypass 7. Because discharged refrigerant from the compressor 1 has a large enthalpy and a high temperature, it can give the outdoor coil 3*a* a considerable amount of heat, so that, if in a heating mode operation part of the outdoor coil 3*a* is frozen, it can be defrosted while continuing to function as an evaporator. Two capillary tubes 39, 44 are provided in parallel across the indoor coil 5*a*, and a control valve 6 is connected between the capillary tubes linking the defrost conduit 45 to one end of the indoor coil 5*a*. These constitute preferred features of the invention. Since the capillary tube 39 is connected between the compressor 1 and the conduit between the indoor coil 5*a* and the expander 4, condensed refrigejnt flows into the defrost conduit 45 and then into the compressor 1. This is advantageous to the compressor 1 because a certain amount of condensed liquid refrigerant flows into the compression chamber of the compressor 1 and evaporates, thus avoiding overheating.

The defrost conduit 45 should preferably be placed adjacent the outdoor coil 3*a*. Since ice is usually formed at the bottom part of the outdoor coil 3*a* the defrost conduit 45 should also be provided for the bottom half thereof. This arrangement can ensure efficient thawing for the indoor coil 5*a*.

CLAIMS

1. A refrigeration system of the type including a hermetic type compressor having a motor connected thereto, a condenser, an expansion device, an evaporator and interconnection means connected to form a refrigerant flow path therethrough, said system further including a first gaseous refrigerant bypass through which a portion of the high pressure refrigerant having at least a condensing pressure is able to be injected into said compressor, wherein injected refrigerant increases the load of said motor thereby adding to the heat generated therefrom so as to supply the refrigerant with said additional heat.

2. A refrigeration system according to claim 1 wherein said first gaseous refrigerant bypass has one end connected to the compression chamber of said compressor and the other end in communication, with the high pressure gaseous refrigerant having at least a condensing pressure and a portion of high pressure gaseous refrigerant from the second end is able to be injected into said compression chamber of said compressor through said one end.

3. A refrigeration system according to claim 1 or 2, wherein said first gaseous refrigerant bypass is provided with a control device therein for controlling the amount of refrigerant flow therethrough.

4. A refrigeration system according to claim 3 in which said control device comprises a capillary tube.

5. A refrigeration system according to claim 1, 2, 3 or 4 comprising an indoor coil which is connected to said compressor and functions as a condenser in a heating mode operation, an expansion device connected to said indoor coil for expanding the refrigerant, an outdoor coil which is connected to said expansion device and functions as an evaporator in a heating mode operation, a reversing device which is connected to said compressor for changing the direction of the refrigerant flow and a first high pressure gaseous refrigerant bypass having one end connected to the compression chamber of said compressor and the other end in communication with the high pressure gaseous refrigerant zone including said indoor coil.

6. A refrigeration system according to claim 5 which further comprises a second gaseous refrigerant bypass having one end in communication with said compression chamber of said compressor and the other end in communication with the high pressure gaseous refrigerant zone including said outdoor coil.

7. A refrigeration system according to claim 6 in which said second gaseous refrigerant bypass has a control device for controlling the refrigerant flow therethrough.

8. A refrigeration system according to claim 7 in which said control device comprises an electric valve for closing and opening said second gaseous refrigerant bypass.

9. A refrigeration system according to any one of claims 5 to 8 further comprising a third refrigerant bypass which has one end connected between said indoor coil and said expansion device and the other end connected to said compressor whereby a mixture of refrigerant from said high pressure gaseous refrigerant bypass and from said third refrigerant bypass is to be injected into said compressor.

10. A refrigeration system according to claim 9 in which said third refrigerant bypass comprises a control device for controlling the refrigerant flow therethrough.

11. A refrigeration system according to claim 10 in which said control device for said third refrigerant bypass comprises a capillary tube.

12. A refrigeration system according to claim 9, 10 or 11 which further comprises a plurality of

capillary tubes as the control device for said third refrigerant bypass and a fourth refrigerant bypass which is provided for the refrigerant to bypass at least one of said plurality of capillary tubes of said third refrigerant bypass.

13. A refrigeration system according to claim 12 in which said fourth refrigerant bypass has a control device for controlling the refrigerant flow therethrough.

14. A refrigeration system according to any one of claims 5 to 13 which further comprises a fifth refrigerant bypass which has one end connected between said indoor coil and said expansion device, the other end to said compressor and a check valve therein which is directed to said compressor, and a sixth refrigerant bypass which has one end connected between said outdoor coil and said expansion device, the other end to said compressor and a check valve therein which is directed to said compressor.

15. A refrigeration system according to any one of the preceding claims which further comprises a seventh refrigerant bypass which has one end connected to said compressor and the other end to an intermediate portion of said expansion device whereby said seventh refrigerant bypass introduces a portion of saturated refrigerant into said compressor.

16. A refrigeration system according to claim 15 in which said seventh refrigeration bypass comprises a control device for controlling the flow of refrigerant therethrough.

17. A refrigeration system according to any one of the preceding claims in which said expansion device is divided into two parts, further comprising a gas liquid separator connected between said two parts of said expansion device and an eighth refrigerant bypass having one end connected to said gas liquid separator and the other end to said compressor whereby a portion of refrigerant in said gas liquid separator is able to be introduced into said compressor through said eighth refrigerant bypass.

18. A refrigeration system according to claim 17 in which said eighth refrigerant bypass comprises a control device for controlling the refrigerant flow therethrough.

19. A refrigeration system according to any one of the preceding claims in which said high pressure gaseous refrigerant bypass is provided inside a hermetic shell of said compressor whereby a portion of the refrigerant is discharged and returned within said hermetic shell of said compressor.

20. A refrigeration system substantially as shown in Figures 1 to 3 and 8 to 15 of the accompanying drawings and described herein with reference thereto.

21. A refrigeration system according to claim 1 substantially as described herein.

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