ABSTRACT

An expansion valve for a refrigerating cycle, in which the body dimensions and the weight of the whole valve can be reduced and a reduction in cost can be achieved.

An expansion valve of the invention comprising a temperature-sensing portion arranged in a refrigerant passage leading to an evaporator from a gas cooler or an internal heat exchanger in a vapor compression type refrigerating cycle and varied in internal pressure according to a refrigerant temperature at an outlet side of the gas cooler or on an outlet side of the internal heat exchanger, a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of a valve port, and a body that accommodates therein the valve member, and wherein the body is provided with a flow passage, through which a refrigerant reduced in pressure by the valve member is led to the evaporator while a refrigerant temperature at the outlet side of the gas cooler or on the outlet side of the internal heat exchanger is transmitted to the temperature-sensing portion. Also, that density, at which a refrigerant is charged in a temperature-sensing body, is set in the range of about 200 kg/m³ to about 600 kg/m³. Further, a ratio of a temperature-sensing cylinder corresponding portion to the temperature-sensing body is made at least 60%.
Fig. 5

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INTERNAL HEAT EXCHANGER 31a 33a 33

MIXED GAS OF REFRIGERANT AND NITROGEN

EVAPORATOR

Fig. 6

34a

34

B

GAS COOLER OR INTERNAL HEAT EXCHANGER

MIXED GAS OF REFRIGERANT AND NITROGEN

EVAPORATOR
Fig. 7

35b MIXED GAS OF REFRIGERANT AND NITROGEN

INTERNAL HEAT EXCHANGER

35 GAS COOLER

INTERNAL HEAT EXCHANGER

EVAPORATOR
Fig. 8

Fig. 9

MIXED GAS OF REFRIGERANT AND NITROGEN

GAS COOLER OR INTERNAL HEAT EXCHANGER

EVAPORATOR
Fig. 11

INTERNAL HEAT EXCHANGER

GAS COOLER

INTERNAL HEAT EXCHANGER

EVAPORATOR
Fig. 12
PRIOR ART

GAS COOLER (RADIATOR)

EVAPORATOR
**Fig. 13**

Improvement in COP when internal heat exchanger is used

![Graph showing COP vs. Superheat (°C)]

- TS = 0
- TS = 10
- TS = 20

**Fig. 14**

When refrigerant temperature in evaporator is 0°C

![Graph showing Control Pressure vs. Refrigerant Temperature at Gas Cooler Outlet (°C)]

- SH = 0
- SH = 10
- SH = 20
- SH = 30
WHEN REFRIGERANT TEMPERATURE IN EVAPORATOR IS 20°C

CONTROL PRESSURE WHEN COP BECOMES MAXIMUM (MPa)

REFRIGERANT TEMPERATURE AT GAS COOLER OUTLET (°C)

SH = 0
SH = 10
SH = 20
SH = 30
**Fig. 17**

- Charged Refrigerant Density of about 450 kg/m³
- Charged Refrigerant Density of about 600 kg/m³
- Flow Rate
- Temperature (°C)
- Valve Opening
- Temperature of Temperature-Sensing Part
- Blow-Off Temperature

*Flow Rate* vs. *Time (s)*

*Pressure (MPa)* vs. *Temperature (°C)*
Fig. 19

INFLUENCE OF AMBIENT TEMPERATURE
(WHEN CHARGED REFRIGERANT DENSITY IS 450kg/m³ AND
REFRIGERANT TEMPERATURE AT GAS COOLER OUTLET IS 60°C.)

TARGET CONTROL PRESSURE

DIRECT TEMPERATURE-SENSING PORTION VOLUME RATIO (%)

CONTROL PRESSURE (MPa)

0.5 MPa

13.8 13.6 13 13.2 12.8 12.6 12.4 12.2

0 50 60 70 80 90 100
Fig. 20

INFLUENCE OF AMBIENT TEMPERATURE (REFRIGERANT TEMPERATURE AT GAS COOLER OUTLET IS 40°C)

- 50°C, 60°C, 70°C, 80°C, 90°C

TARGET CONTROL PRESSURE

DIRECT TEMPERATURE-SENSING PORTION VOLUME RATIO (%)

CONTROL PRESSURE (MPa)

0.5 MPa
EXPANSION VALVE FOR REFRIGERATING CYCLE

BACKGROUND OF THE INVENTION

[0001] 1. Field of the Invention

[0002] The present invention relates to an expansion valve for a refrigerating cycle that controls a refrigerant on a radiator outlet side on the basis of a refrigerant temperature at the radiator (gas cooler) outlet side of a vapor-compression-type refrigerating cycle, and is especially suited to a supercritical refrigerating cycle that uses a refrigerant, such as carbon dioxide (CO₂) or the like, in a supercritical range.

[0003] 2. Description of Related Art

[0004] Generally, it is known to use, as a vehicular air conditioning apparatus, a vapor-compression-type refrigerating cycle that circulates CO₂ as a refrigerant in a closed circuit comprising a compressor 1, a gas cooler (radiator) 2, an expansion valve 3, an evaporator 4, an accumulator 5, etc. Conventionally, a pressure control valve as disclosed in JP-A-2000-193347 and JP-A-2003-254460 is known as a mechanical type expansion valve used in such a vapor compression type refrigerating cycle.

[0005] As shown in FIG. 12, the pressure control valves disclosed in JP-A-2000-193347 and JP-A-2003-254460 control a refrigerant pressure at an outlet side of a radiator 2 by passing a refrigerant at an outlet of a radiator 2 in a casing 30, which covers a valve member part, in which gases such as refrigerant or the like are charged in an enclosed space A formed on one side of a diaphragm 32 with the diaphragm therebetween, and a pressure of high pressure refrigerant before pressure reduction acts on the other side to displace the diaphragm 32 to make a valve member 31 move, and detecting a refrigerant in the enclosed space (temperature-sensing portion) A.

[0006] However, the pressure control valve of the conventional type involves a problem that the weight is increased to lead to an increase in cost as there is a need for the casing 33 that covers the enclosed space (temperature-sensing portion).

[0007] Also, there is also known a pressure control valve (expansion valve) of a type in which the casing 30 is eliminated to achieve reduction in cost, an enclosed space is connected to a temperature-sensing cylinder 7 through a capillary tube 6, the temperature-sensing cylinder 7 is provided in contact with a pipe at an outlet of a radiator 2, and the temperature-sensing cylinder 7 detects a refrigerant temperature at the outlet of the radiator 2, but this type of expansion valve involves a problem of an increase in cost as there is a need of a process of assembling the temperature-sensing cylinder 7.

[0008] Also, the case where CO₂ is used as a refrigerant involves a problem that the theoretical cycle efficiency is low as compared to HFC134a as conventionally used. Therefore, there is a need of enhancing an efficiency COP of a refrigerating cycle through heat exchange between a gas cooler outlet refrigerant and a refrigerant sucked by a compressor with the use of an internal heat exchanger shown in FIG. 3. When the internal heat exchanger is used, a sucked refrigerant of the compressor is heated and enthalpy is increased to bring about a superheat state. In order to efficiently operate a refrigerating cycle in which a refrigerant, such as CO₂, with high pressure becomes supercritical, there is proposed a construction in which density in an enclosed space is prescribed but this takes no account of a refrigerating cycle using an internal heat exchanger (see JP-A-9-264622).

[0009] Further, as a gas cooler outlet refrigerant temperature or an internal heat exchanger outlet refrigerant temperature is detected in a CO₂ cycle, a high pressure control valve is arranged in an engine room in the case where the cycle is applied to a vehicular air conditioning apparatus. As the temperature in the engine room is higher than that of an outside air and a refrigerant cooled by a gas cooler does not flow to the control valve when the cycle is stopped, the control valve is heated to the ambient temperature in the engine room, which is higher than that of an outside air, and sometimes reaches 100°C to 120°C.

[0010] As a refrigerant is charged in a temperature-sensing portion in the control valve, the pressure in the temperature-sensing portion rapidly rises when an ambient temperature rises and the charged refrigerant is heated. As a refrigerant temperature at a gas cooler outlet is cooled close to the ambient temperature, a maximum temperature in the engine room reaches 30 to 60°C above a maximum temperature of the refrigerant at the gas cooler outlet. Therefore, the pressure in the temperature-sensing portion at the time of stoppage becomes higher than a maximum pressure of the CO₂ cycle, so that a very high pressure-resistance, above that for other high pressure parts, is demanded of the temperature-sensing portion.

[0011] In this manner, when the control valve is heated to an ambient temperature in the engine room, the pressure in the temperature-sensing portion becomes higher than a normal high-pressure control pressure to bring about a valve-closed state at the startup of the CO₂ cycle. Therefore, cooling of the temperature-sensing portion is conventionally performed by circulating a small quantity of refrigerant through a bleed hole provided near the valve part and causing the refrigerant cooled by a gas cooler to flow to the control valve. Thereafter, the control valve is opened until temperature of the temperature-sensing portion is decreased and internal pressure of the temperature-sensing portion is decreased to a range of high-pressure control pressure, so that the refrigerant is increased in flow rate and a maximum cooling capacity is obtained. Accordingly, in order to reduce the time elapsed until the maximum cooling capacity is attained, that is, cool-down, it becomes important to quickly lower the internal pressure of the temperature-sensing portion to a normal control pressure.

[0012] Besides, in a supercritical cycle using CO₂, a refrigerant in a temperature-sensing portion is put in a supercritical state as the temperature of a refrigerant at a gas cooler outlet, in which high pressure is attained, or an internal heat exchanger outlet is detected. With a conventional HFC134a, a refrigerant in a temperature-sensing portion is used in a gas-liquid two-phase and a refrigerant pressure is determined at a saturation temperature, that is, a liquid refrigerant temperature, so that pressure in the temperature-sensing portion is not affected by temperatures in other regions. However, a refrigerant put in a supercritical state is affected by temperatures of those regions, which are communicated to and other than the temperature-sensing
portion, to cause a problem that the internal pressure of the temperature-sensing portion is not determined and control pressure is varied.

SUMMARY OF THE INVENTION

[0013] The invention has been made in view of the above problems and has as its object to provide an expansion valve for a refrigerating cycle that does not need any casing and any temperature-sensing cylinder, can reduce the body dimensions and the weight of the whole valve and enables a reduction in cost. A further object is to provide an expansion valve for a refrigerating cycle that can decrease the pressure-resistance of a temperature-sensing portion by optimizing control characteristics in the case where an internal heat exchanger is used in combination. A still further object is to provide an expansion valve or a refrigerating cycle comprising an expansion valve which, when used in a supercritical cycle, decreases variation in control pressure and enables miniaturization of an expansion valve member.

[0014] The invention provides, as means for solving the problem, an expansion valve for a refrigerating cycle according to the respective claims.

[0015] An expansion valve for a refrigerating cycle according to the first aspect of the present invention is arranged in a refrigerant passage leading from a gas cooler to an evaporator in a vapor compression type refrigerating cycle, and comprises a temperature-sensing portion, inner pressure of which is varied according to the refrigerant temperature at the outlet side of the gas cooler. The valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and a body that accommodates therein the valve member, and the body is provided with a flow passage, through which a refrigerant reduced in pressure by the valve member is led to the evaporator while the refrigerant temperature at the outlet side of the gas cooler is transmitted to the temperature-sensing portion, whereby it is possible to omit a casing that covers the temperature-sensing portion, or a capillary tube and a temperature-sensing cylinder, into which a refrigerant is introduced, and to achieve miniaturization of the expansion valve and reduction in cost.

[0016] An expansion valve for a refrigerating cycle according to the second aspect of the present invention is applied to a vapor compression type refrigerating cycle provided with an internal heat exchanger, and arranged in a refrigerant passage leading from an internal heat exchanger to an evaporator, the expansion valve comprising a temperature-sensing portion, inner pressure of which is varied according to the refrigerant temperature at the outlet side of the gas cooler, and a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and a body that accommodates therein the valve member, and wherein the body is provided with a first flow passage, through which a refrigerant flows to the internal heat exchanger, and a second flow passage, through which a refrigerant reduced in pressure by the valve member is led to the evaporator from the internal heat exchanger, while the refrigerant temperature at the outlet side of the gas cooler is transmitted to the temperature-sensing portion, whereby it is possible in the same manner as the first aspect to achieve miniaturization of the expansion valve and a reduction in cost.

[0017] An expansion valve for a refrigerating cycle according to the third aspect of the present invention is applied to a vapor compression type refrigerating cycle provided with an internal heat exchanger, and is arranged in a refrigerant passage leading from an internal heat exchanger to an evaporator, the expansion valve comprising a temperature-sensing portion, inner pressure of which is varied according to the refrigerant temperature at the outlet side of the internal heat exchanger, a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and a body that accommodates therein the valve member, and wherein the body is provided with a flow passage, through which a refrigerant reduced in pressure by the valve member flows to the evaporator while the refrigerant temperature at the outlet side of the internal heat exchanger is transmitted to the temperature-sensing portion.

[0018] With the expansion valve, the temperature-sensing portion can comprise a diaphragm, and a lid and a lower support member, which interpose therebetween a peripheral edge of the diaphragm from upper and lower directions to define an enclosed space above the diaphragm, transmission of a refrigerant temperature to the temperature-sensing portion is performed by a clearance, which is formed by the valve member and the lower support member to be communicated to the refrigerant passage, whereby it is possible to transmit a refrigerant temperature to the temperature-sensing portion through the clearance and to omit a casing, or a capillary tube and a temperature-sensing cylinder.

[0019] With the expansion valve, the enclosed space of the temperature-sensing portion can be charged with a refrigerant and provided with an adjustment spring, which biases the valve member in a Solv closing direction, and a valve closing force provided by internal pressure in the temperature-sensing portion and the adjustment spring and a valve opening force provided by a refrigerant pressure balance to operate the valve member.

[0020] With the expansion valve, the enclosed space of the temperature-sensing portion can be charged with a mixed gas of a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, and an adjustment spring, which biases the valve member in a valve closing direction, is omitted, whereby it is possible to simplify the construction and reduced the number of parts.

[0021] An expansion valve for a refrigerating cycle according to the fourth aspect of the present invention is one provided with an internal heat exchanger, and has a feature in that a density, at which a refrigerant is charged in the temperature-sensing portion, is 200 to 600 kg/m² in a valve closed state. Thereby, it is possible to optimize control characteristics when an internal heat exchanger is used, and to decrease pressure-resistance of the temperature-sensing body.

[0022] With the expansion valve, the density, at which a refrigerant is charged in the temperature-sensing portion, can be 200 to 450 kg/m² in a valve closed state, whereby it is possible to further optimize control characteristics and to decrease pressure-resistance of the temperature-sensing body.

[0023] With the expansion valve, the valve member can be opened when high pressure at the outlet side of the gas
cooler or at the outlet side of the internal heat exchanger becomes higher by a predetermined magnitude than inner pressure in the temperature-sensing portion.

[0024] With the expansion valve, a load corresponding to the predetermined magnitude can be given by an elastic member, or a non-condensed gas charged in the temperature-sensing portion together with a refrigerant, or the elastic member and the non-condensed gas.

[0025] With the expansion valve, the elastic member can be any one of a coil spring, a diaphragm, and a bellows, or an optional combination thereof.

[0026] With the expansion valve, when a refrigerant temperature at the outlet side of the gas cooler is 50°C or higher, the internal heat exchanger can heat a refrigerant sucked into a compressor so that superheat becomes 10°C or higher.

[0027] An expansion valve for a refrigerating cycle according to the fifth aspect of the present invention is one that uses a refrigerant in a supercritical state, and comprises a temperature-sensing portion having a first enclosed space provided above a diaphragm and charged with a refrigerant, and a second enclosed space provided below the diaphragm to be communicated to the first enclosed space. Thereby, it is possible to enlarge a volume of the temperature-sensing body and to improve the temperature-sensing body in accuracy.

[0028] With the expansion valve, the second enclosed space can be provided inside a valve member fixed to the diaphragm.

[0029] With the expansion valve, the sum of a half of a volume of the first enclosed space and a volume of the second enclosed space can amount to 60% or more of the sum of a volume of the first enclosed space and the second enclosed space. Thereby, it is possible to lessen influences of temperature at a portion of the temperature-sensing portion except the temperature-sensing cylinder corresponding portion.

[0030] The expansion valve can further comprise a lid that covers a wall surface of the first enclosed space in contact with an outside air to provide an air layer between the wall surface and the outside air, and can lessen the influence of the temperature of the outside air.

[0031] With the expansion valve, at least a part of the wall surface of the first enclosed space in contact with an outside air can be covered by a thermal insulating material, and it is possible to further lessen the influence of temperature of the outside air.

BRIEF DESCRIPTION OF THE DRAWINGS

[0032] FIG. 1 is a view illustrating a vapor compression type refrigerating cycle, in which CO₂ is circulated as a refrigerant;

[0033] FIG. 2 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a first embodiment of the invention, in the refrigerating cycle illustrated in FIG. 1;

[0034] FIG. 3 is a view illustrating a vapor compression type refrigerating cycle including an internal heat exchanger;

[0035] FIG. 4 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a second embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 3;

[0036] FIG. 5 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a third embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 3;

[0037] FIG. 6 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a fourth embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 1 or 3;

[0038] FIG. 7 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a fifth embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 3;

[0039] FIG. 8 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a sixth embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 1 or 3;

[0040] FIG. 9 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a seventh embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 1 or 3;

[0041] FIG. 10 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to an eighth embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 3;

[0042] FIG. 11 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a ninth embodiment of the invention, applied to the refrigerating cycle illustrated in FIG. 3;

[0043] FIG. 12 is a cross sectional view showing a conventional expansion valve for a refrigerating cycle (pressure control valve);

[0044] FIG. 13 is a view showing an improvement in COP in the case where an internal heat exchanger is used;

[0045] FIG. 14 is a view showing control pressure, at which COP becomes maximum, versus a gas cooler outlet temperature when a refrigerant in an evaporator is 0°C;

[0046] FIG. 15 is a view showing control pressure, at which COP becomes maximum, versus a gas cooler outlet temperature when a refrigerant in an evaporator is 20°C;

[0047] FIG. 16 is a view showing a collier chart representative of physical properties Of CO₂ refrigerant;

[0048] FIG. 17 is a view schematically showing effects at the time of cool-down;

[0049] FIG. 18 is a view schematically showing a temperature-sensing cylinder corresponding portion of a temperature-sensing body and a portion except the portion;

[0050] FIG. 19 is a view (first) showing a change in control pressure versus a ratio of a temperature-sensing cylinder corresponding portion;

[0051] FIG. 20 is a view (second) showing a change in control pressure versus a ratio of a temperature-sensing cylinder corresponding portion; and
FIG. 21 is a view showing an embodiment obtained by providing a lid on a temperature-sensing portion of the ninth embodiment.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

An expansion valve for a refrigerating cycle according to an embodiment of the invention will be described below with reference to the drawings. FIG. 1 is a view illustrating a vapor compression type refrigerating cycle (supercritical refrigerating cycle), in which CO₂ is circulated as a refrigerant, and FIG. 2 is a cross sectional view showing an expansion valve for a refrigerating cycle, according to a first embodiment of the invention, applied to the vapor compression type refrigerating cycle illustrated in FIG. 1. In FIG. 1, the reference numeral 1 denotes a compressor that sucks and compresses a refrigerant (CO₂), and 2 a gas cooler (radiator) that cools the refrigerant compressed by the compressor 1. An expansion valve 3 is arranged on an outlet side of the gas cooler 2 to control a refrigerant pressure at the outlet side of the gas cooler 2 on the basis of a refrigerant temperature at the outlet side of the gas cooler 2, the expansion valve also functioning as a decompressor that decompresses a refrigerant at high pressure. In FIG. 1, a temperature-sensing cylinder 7 is mounted on an outlet-side pipe of the gas cooler 2 and connected to the expansion valve 3 through a capillary tube 6. Accordingly, a valve opening degree of the expansion valve 3 is controlled according to the change in internal pressure, which is based on a refrigerant temperature of gases charged in the temperature-sensing cylinder 7.

The reference numeral 4 denotes an evaporator that evaporates a gas-liquid two-phase refrigerant decreased in pressure by the expansion valve 3, and 5 an accumulator that separates a gaseous phase refrigerant and a liquid phase refrigerant from each other and temporarily accumulates a surplus refrigerant in the refrigerating cycle. The compressor 1, the gas cooler 2, the expansion valve 3, the evaporator 4, and the accumulator 5 are connected together by means of piping to form a closed circuit.

Subsequently, an expansion valve for a refrigerating cycle 3A according to the first embodiment will be described with reference to FIG. 2. Formed in a body 33 of the expansion valve 3A is a part of a refrigerant low passage leading from the gas cooler 2 to the evaporator 4 via a valve port 33a. Formed in the body 33 are an inflow port 33b connected to a side of the gas cooler 2, an outflow port 33c connected to a side of the evaporator 4, a first opening 33d, to which a temperature-sensing portion described later is mounted, and a second opening 33e in which an adjustment spring 36 is set. A valve member 31 is received in the body 33 to open and close the valve port 33a whereby an upstream space C₁ is connected to an outlet side of the gas cooler 2 and a downstream space C₂ connected to an inlet side of the evaporator 4, which spaces are disposed in the body 33, are put into communication and non-communication to each other.

The temperature-sensing portion is mounted to the first opening 33d of the body 33. The temperature-sensing portion mainly comprises the diaphragm 32, a lid 35, and a lower support member 34, and is formed therein with an enclosed space A. That is, a concave portion 35a is formed centrally of the lid 35 to define the enclosed space A, and the lid 35 and the lower support member 34 interpose and secure a peripheral edge of the diaphragm 32 therewith to form the temperature-sensing portion. The diaphragm 32 is in the form of a thin film made of a stainless steel material to be deformed and displaced according to a pressure difference inside and outside the enclosed space A. The lower support member 34 comprises a cylindrical portion 34a and a flange portion 34b, and a threaded portion formed on an outer periphery of the cylindrical portion 34a is threaded into the first opening 33d of the body 33 to mount the temperature-sensing portion to the body 33. Also, a charge pipe 35b is mounted to the lid 35 and a refrigerant is charged into the enclosed space A through the charge pipe 35b. After the refrigerant is charged, the charge pipe 35b is sealed.

One end 31b of the valve member 31, extending upwardly, of a valve portion 31a through the cylindrical portion 34a of the lower support member 34 is fixed to the diaphragm 32, and a clearance B having an annular-shaped cross section is formed between an inner surface of the cylindrical portion 34a and an outer peripheral surface of the valve member 31. The clearance B is communicated to an upstream space C₁ connected to the outlet side of the gas cooler 2. Accordingly, a refrigerant on the outlet side of the gas cooler 2 flows into the clearance B, so that a refrigerant temperature is transmitted to a refrigerant in the enclosed space A and at the same time pressure of the refrigerant on the outlet side of the gas cooler 2 acts on the diaphragm 32.

Further, an adjustment nut 37 is threaded onto the other end 31c of the valve member 31 extending downwardly of the valve portion 31a through the valve port 33a. The adjustment spring 36 that biases the valve member 31 in a valve closing direction is interposed between a neighborhood of an underside of the valve port 33a and the adjustment nut 37, and an initial set load (an elastic force in a state, in which the valve port 33a is closed) of the adjustment spring 36 can be optionally adjusted by rotating the adjustment nut 37. The adjustment spring 36, the adjustment nut 37, etc. are provided in the downstream space C₂ connected to the inlet side of the evaporator 4. Also, a cap 38 is fitted into the second opening 33e of the body 33 whereby a lower part of the downstream space C₂ is closed.

With the expansion valve for a refrigerating cycle 3A, according to the first embodiment, constructed in the above manner, a valve closing force of the valve member 31 is provided by inner pressure in the enclosed space A and the adjustment spring 36, a valve opening force of the valve member 31 is provided by a refrigerant pressure at the outlet side of the gas cooler 2, and balance of the both forces affords opening and closing the expansion valve 3A. Also, the inner pressure in the enclosed space A is varied depending upon temperature of that refrigerant on the outlet side of the gas cooler 2, which flows into the clearance B, whereby the valve port 33a is varied in opening degree to control the refrigerant pressure at the outlet side of the gas cooler 2.

FIG. 3 is a view illustrating a vapor compression type refrigerating cycle, in which an internal heat exchanger is incorporated. In this manner, the vapor compression type refrigerating cycle including an internal heat exchanger is a conventionally known refrigerating cycle to improve a cooling capacity. In this case, an internal heat exchanger 8 is arranged in the cycle as shown in FIG. 3 so as to make heat
exchange between a refrigerant going to the expansion valve 3 from the gas cooler 2 and a refrigerant returning to the compressor 1 from the accumulator 5. Accordingly, the evaporator valve 3 is arranged in a refrigerant passage leading from the internal heat exchanger 8 to the evaporator 4. The remaining construction is the same as the vapor compression type refrigerating cycle illustrated in FIG. 1 and so an explanation therefor is omitted. The refrigerating cycle according to the invention is also applicable to a vapor compression type refrigerating cycle including such an internal heat exchanger.

[0061] FIG. 4 is a cross sectional view showing an expansion valve for a refrigerating cycle 3B, according to a second embodiment, applied to a vapor compression type refrigerating cycle including an internal heat exchanger. A first flow passage D making a part of a refrigerant flow passage leading from a gas cooler 2 to an internal heat exchanger 8 and a second flow passage E making a part of a refrigerant flow passage leading from the internal heat exchanger 8 to an evaporator 4 via a valve port 33a, respectively, are formed independently in a body 33 of the expansion valve 3B according to the second embodiment. According to the second embodiment, a clearance B, through which a refrigerant temperature at an outlet side of the gas cooler 2 is transmitted to a refrigerant in an enclosed space A of a temperature-sensing portion, is provided on a side of the first flow passage D, and a valve portion 31b of a valve member 31, which opens and closes the valve port 33a, is provided on a side of the second flow passage E.

[0062] That is, one end 31b of the valve member 31 extending upwardly of a valve portion 31a across the first flow passage D and through a cylindrical portion 34a of a lower support member 34 is fixed to a diaphragm 32, and a clearance B having an annular-shaped cross section is provided between an inner surface of the cylindrical portion 34a and an outer peripheral surface of the valve member 31. The clearance E is communicated to the first flow passage D connected to the outlet side of the gas cooler 2. Accordingly, a refrigerant on the outlet side of the gas cooler 2 flows into the clearance B, so that the refrigerant temperature is transmitted to a refrigerant in the enclosed space A and at the same time pressure of the refrigerant on the outlet side of the gas cooler 2 acts on the diaphragm 32.

[0063] A valve port 33a providing for communication between the internal heat exchanger 8 and the evaporator 4 is provided in the second flow passage E. Accordingly, the valve portion 31a of the valve member 31, which opens and closes the valve port 33a, an adjustment spring 36 provided on the other end 31c of the valve member 31 extending downwardly through the valve port 33a, an adjustment nut 37, etc. are provided in the second flow passage E. The remaining detailed construction is the same as that of the first embodiment and so an explanation therefor is omitted.

[0064] FIG. 5 is a cross sectional view showing an expansion valve for a refrigerating cycle 3C, according to a third embodiment, applied to a vapor compression type refrigerating cycle including an internal heat exchanger. According to the third embodiment, a part of a refrigerant flow passage leading from an internal heat exchanger 8 to an evaporator 4 via a valve port 33a is formed in a body 33 of the expansion valve 3C. That is, the remaining construction is the same as that of the expansion valve 3A of the first embodiment except that an inflow port 33b of the body 33 is connected to the internal heat exchanger 8 in place of a gas cooler 2.

[0065] Accordingly, according to the third embodiment, a refrigerant on an outlet side of the internal heat exchanger 8 flows into a clearance B, so that a refrigerant temperature at the outlet side of the internal heat exchanger 8 is transmitted to a refrigerant charged in an enclosed space A of a temperature-sensing portion. Likewise, a refrigerant pressure at the outlet side of the internal heat exchanger 8 acts on a diaphragm 32. The same function and effect as those in the first embodiment are produced also in the third embodiment.

[0066] FIG. 6 is a cross sectional view showing an expansion valve for a refrigerating cycle 3D, according to a fourth embodiment, applied to the vapor compression type refrigerating cycle illustrated in FIG. 1 or FIG. 3. According to the fourth embodiment, in place of the adjustment spring 36, for example, nitrogen gases (N2), helium gases (He), etc., which are lower in the coefficient of thermal expansion than a refrigerant, together with the refrigerant are charged in an enclosed space A in the expansion valve according to the first embodiment in FIG. 2 or in the expansion valve according to the third embodiment in FIG. 5. That is, according to the fourth embodiment, a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, are charged in the enclosed space A of the temperature-sensing portion, a second opening 33c of a body 33 is closed, and a portion extending downwardly of a valve portion 31a of a valve member 31, an adjustment spring 36, an adjustment nut 37, etc. are removed. The remaining construction is the same as that of the first embodiment or the third embodiment and so an explanation therefor is omitted.

[0067] Accordingly, according to the fourth embodiment, only inner pressure of those mixed gases charged in the enclosed space A, to which temperature of a refrigerant on the outlet side of the gas cooler 2 flowing into a clearance B is transmitted, acts as a valve closing force of the valve member 31, and a refrigerant pressure at the outlet side of the gas cooler 2 acts as a valve opening force. In this manner, according to the fourth embodiment, gases, which are lower in the coefficient of thermal expansion than the refrigerant, function as an adjustment spring 36. Also, in the case where a refrigerant is carbon dioxide (CO2) and the gas being mixed are nitrogen gas (N2), it is preferred that carbon dioxide (CO2) be charged at a density in the order of 500 to 700 kg/m3 and nitrogen gas (N2) be charged at a density in the order of 10 to 40 kg/m3.

[0068] FIG. 7 is a cross sectional view showing an expansion valve for a refrigerating cycle 3E, according to a fifth embodiment, applied to a vapor compression type refrigerating cycle including the internal heat exchanger shown in FIG. 3. According to the fifth embodiment, as in the fourth embodiment, in place of the adjustment spring 36, for example, nitrogen gas (N2) helium gas (He), etc., which are lower in the coefficient of thermal expansion than a refrigerant, together with the refrigerant are charged in an enclosed space A in the expansion valve 3B according to the second embodiment. That is, according to the fifth embodiment, a mixed gas of a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, are charged in an enclosed space A of a temperature-
sensing portion, a second opening 33e of a body 33 is closed, and a portion extending downwardly of a valve portion 31a of a valve member 31, an adjustment spring 36, an adjustment nut 37, etc. are removed from a second flow passage E. The remaining construction is the same as that of the second embodiment and so an explanation therefor is omitted.

Accordingly, according to the fifth embodiment, only inner pressure of those mixed gases charged in the enclosed space A, to which temperature of a refrigerant on an outlet side of a gas cooler 2 flowing into a clearance B is transmitted, acts as a valve closing force of the valve member 31, and a refrigerant pressure at the outlet side of the gas cooler 2 acts as a valve opening force. In this manner, according to the fifth embodiment, gases, which are lower in the coefficient of thermal expansion than the refrigerant, functions as an adjustment spring 36. Also, in the case where a refrigerant is carbon dioxide (CO₂) and the gases being mixed are nitrogen gas (N₂), it is preferred that carbon dioxide (CO₂) be charged at a density in the order of 400 to 550 kg/m³ and nitrogen gases (N₂) be charged at a density in the order of 10 to 40 kg/m³.

FIG. 8 is a cross sectional view showing an expansion valve for a refrigerating cycle 3F, according to a sixth embodiment of the invention, applied to the refrigerating cycle shown in FIG. 1 or FIG. 3. According to the sixth embodiment, a cavity 31d communicated to an enclosed space A of a temperature-sensing portion is formed in the valve member 31 of the expansion valve 3 according to the first embodiment in FIG. 2 or according to the third embodiment in FIG. 5. Accordingly, the enclosed space of the temperature-sensing portion can comprise the sum of (the enclosed space A+the cavity 31d+the charge pipe 35b), and the enclosed space charged with a refrigerant can be enlarged, so that it is possible to improve the temperature-sensing portion in accuracy. The remaining construction is the same as that of the first embodiment or the third embodiment and so an explanation therefor is omitted.

FIG. 9 is a cross sectional view showing an expansion valve for a refrigerating cycle 3G, according to a seventh embodiment of the invention, applied to the refrigerating cycle shown in FIG. 1 or FIG. 3. According to the seventh embodiment, like the sixth embodiment, a cavity 31d communicated to an enclosed space A of a temperature-sensing portion is formed in the valve member 31 of the expansion valve 3 according to the fourth embodiment in FIG. 6. Also, according to the seventh embodiment, the enclosed space of the temperature-sensing portion can be further increased by a volume of the cavity 31d, so that it is possible to improve the temperature-sensing portion in accuracy. The remaining construction is the same as that of the fourth embodiment and so an explanation therefor is omitted.

FIG. 10 is a cross sectional view showing an expansion valve for a refrigerating cycle 3H, according to an eighth embodiment of the invention, applied to the refrigerating cycle shown in FIG. 3. According to the eighth embodiment, like the sixth and seventh embodiments, a cavity 31d communicated to an enclosed space A of a temperature-sensing portion is formed in the valve member 31 of the expansion valve 3 according to the fifth embodiment in FIG. 7. Also, according to the eighth embodiment, the enclosed space of the temperature-sensing portion can be further increased by a volume of the cavity 31d, so that it is possible to improve the temperature-sensing portion, in accuracy. The remaining construction is the same as that of the fifth embodiment and so an explanation therefor is omitted.

FIG. 11 is a cross sectional view showing an expansion valve for a refrigerating cycle 3I, according to a ninth embodiment of the invention, applied to the refrigerating cycle shown in FIG. 3. According to the ninth embodiment, like the sixth, seventh, and eighth embodiments, a cavity 31d communicated to an enclosed space A of a temperature-sensing portion is formed in the valve member 31 of the expansion valve 3 according to the second embodiment in FIG. 4. Also, according to the ninth embodiment, the enclosed space of the temperature-sensing portion can be further increased by a volume of the cavity 31d, so that it is possible to improve the temperature-sensing portion, in accuracy. The remaining construction is the same as that of the second embodiment and so an explanation therefor is omitted.

In addition, while the embodiments have described an expansion valve used for a vapor compression type refrigerating cycle, in which carbon dioxide (CO₂) is used as a refrigerant, the expansion valve for a refrigerating cycle according to the invention is not limited thereto but is also applicable to a vapor compression type refrigerating cycle, in which the refrigerant is fluorocarbon or the like, not to mention a vapor compression type refrigerating cycle, in which a refrigerant, such as ethylene, ethane, nitrogen oxide, etc., used in a supercritical zone, is used.

Subsequently, an explanation is given to an embodiment of an expansion valve suited to a supercritical refrigerating cycle, in which the internal heat exchanger 8 shown in FIG. 3 is incorporated. The expansion valve according to the embodiment is intended for firstly, improving COP of a refrigerating cycle including an internal heat exchanger, secondly, enabling decreasing the pressure-resistance of a temperature-sensing portion to achieve reduction in cost, and thirdly, accelerating the cool-down. Therefore, the embodiment prescribes the density at which a refrigerant is charged in a temperature-sensing portion. An explanation is given below.

FIG. 13 shows effects of an improvement in COP in the case where an internal heat exchanger is used to provide for superheat in a sucked refrigerant. TS in the figure indicates a refrigerant evaporating temperature in an evaporator. Accordingly, the higher a refrigerant temperature in an evaporator, the higher an improvement in COP. In a vehicular air conditioner, a compressor is decreased in rotating speed at the time of idling and a cooling capacity becomes minimum. However, as a refrigerant evaporating temperature in an evaporator rises, COP of a vehicular air conditioner is enhanced when an internal heat exchanger is used. In this manner, the use of an internal heat exchanger in a vehicular air conditioner produces a great advantage.

FIGS. 14 and 15 show high pressure control pressures, at which COP becomes a maximum, relative to a gas cooler outlet refrigerant temperature in the case where a refrigerant temperature in an evaporator is 0° C. and in the case where a refrigerant temperature in an evaporator is 20° C., and show characteristics that in the case where an
internal heat exchanger is used to heat a compressor sucked refrigerant, the lower control pressure in case of possessing superheat, the higher a refrigerant evaporating temperature in an evaporator and the higher a gas cooler outlet refrigerant temperature.

[0078] This is apparent in the Mollier chart shown in FIG. 16 and representative of physical properties of CO₂ refrigerant. That is, a refrigerant sucked by a compressor ideally follows along an isentropic line to be compressed to a high temperature high pressure refrigerant. An isentropic line for the physical properties of CO₂ refrigerant is less inclined as it goes to the right side in the Mollier chart where enthalpy is increased. This is because, when a comparison is made at the same pressure, an increase in enthalphy (=compressor power) in case of compression to the same pressure becomes large in the case where a refrigerant with superheat is heated, as compared with the case where a saturated gas refrigerant is sucked and compressed.

[0079] For a cycle with the use of CO₂ refrigerant, there is known a method of exercising control to high pressure, at which COP becomes maximum, relative to a gas cooler outlet refrigerant temperature. In case of the provision of an internal heat exchanger, there is produced an advantage that a high pressure, at which COP becomes maximum, is decreased since a compressor power is increased. Also, the ability of making a control pressure low produces an advantage in improving other high-pressure parts such as a compressor, a gas cooler, etc. in durability.

[0080] For example, with vehicles, since a traveling wind is not generated at the time of idling, a gas cooler is decreased in wind velocity, and additionally a sucked air temperature rises and a gas cooler outlet refrigerant temperature rises due to blowing-in of a hot wind from an engine room. Accordingly, control pressure becomes low in the case where an internal heat exchanger is used.

[0081] Accordingly, in order to make effective use of a refrigerating cycle, in which an internal heat exchanger is used, there is a need for a high-pressure control valve having control characteristics that control pressure is further decreased for the same gas cooler outlet refrigerant temperature. Also, it is necessary to charge a refrigerant into a control valve having such characteristics at a lower density than that at which a refrigerant is charged into a conventional temperature-sensing portion (see JP-A-9-264622).

[0082] As seen from FIG. 15, assuming that a refrigerant temperature in an evaporator is 20° C. and superheat of a sucked refrigerant is 10° C. in a refrigerating cycle, in which an internal heat exchanger having a small heat exchanging capacity is used, COP assumes a maximum value when a refrigerant temperature at a gas cooler outlet is 60° C. and control pressure is 15 MPa. In order to make a control pressure attain 15 MPa, it is necessary to adopt a charged refrigerant density (hereinafter called a charging density) in the order of about 600 kg/m³.

[0083] Since COP is improved when an internal heat exchanger having a large heat exchanging capacity is used, it is conceivable to increase a quantity of superheat further. As a discharge temperature also rises when a sucked refrigerant temperature of a compressor becomes high, however, a quantity of superheat is preferred to be in the range of 15 to 25° C. In case of adopting a quantity of superheat in the range of 15 to 25° C., COP becomes maximum in the case where control pressure is made 14.2 MPa, for example, when a gas cooler outlet refrigerant temperature is 60° C. In order to make a control pressure attain 14.2 MPa, it is necessary to adopt a charging density in the order of about 570 kg/m³.

[0084] Also, as that density, at which a refrigerant is charged in a temperature-sensing portion of an expansion valve, is desirably low in terms of pressure-resistance of the expansion valve described later, inner pressure in the temperature-sensing portion is set low by about 2 MPa by further using in combination a spring for biasing the valve in a valve closing direction whereby control pressure, at which COP becomes maximum, can be ensured even in a charging density in the order of about 450 kg/m³ when a gas cooler outlet refrigerant temperature is 60° C. Subsequently, an explanation is given to pressure-resistance of a temperature-sensing portion. As pressure in the temperature-sensing portion at the time of stoppage of a vehicle becomes very high, a large pressure-resistance is required. As is apparent from the Mollier chart of CO₂ refrigerant shown in FIG. 16, the higher a density, the more rapid pressure rises relative to temperature, so that in order to decrease an increase in internal pressure of a temperature-sensing portion, it is necessary to lower a charging density. In particular, there is caused a problem that since an inclination of an isothermal line intersecting an equidensity line becomes large when the charging density exceeds 600 kg/m³, an increase in internal pressure relative to temperature rise becomes also large.

[0085] Also, since a maximum allowable pressure of high-pressure parts is set to about 18 MPa, an upper limit of pressure in a temperature-sensing portion is made in the same order as the pressure to eliminate the need of excessively heightening only the temperature-sensing portion in strength to enable making the same equal to other high-pressure parts in strength, thus enabling providing a control valve at low cost.

[0086] Therefore, while it is required that a temperature-sensing portion charging density be set to at most about 550 kg/m³ when a maximum ambient temperature is 80° C., at most about 450 kg/m³ when a maximum ambient temperature is 100° C., and at most about 360 kg/m³ when a maximum ambient temperature is 120° C., it is desired that the charging density be set to at most 450 kg/m³ since 100° C. at the highest must be taken account of even when a position of low temperature is selected as a mount position in an engine room.

[0087] Further, since a charging density for an intended control pressure can be reduced by a quantity corresponding to a spring load by giving a load in a direction of closure with the use of a spring or the like, it is effective to use the spring or the like in combination.

[0088] When a temperature-sensing portion charging density is made small, control pressure for a gas cooler outlet refrigerant temperature is decreased but the control pressure, at which COP becomes maximum, is also decreased in case of using an internal heat exchanger, so that the use of the internal heat exchanger makes it possible to decrease that density, at which a refrigerant is charged in a temperature-sensing portion of an expansion valve, without decreasing COP.
In addition, as shown in the Mollier chart in FIG. 16, a tendency is demonstrated, in which an inclination of the isothermal line becomes rapidly small and a change in enthalpy become large relative to pressure change when temperature and pressure of a refrigerant come near to a critical point. Since a quantity of discharged heat is decreased and a cooling capacity is decreased when enthalpy at a gas cooler outlet increases, it is desired that, for example, high pressure at 40°C of refrigerant temperature be equal to or higher than 9 MPa (T point in FIG. 16).

Even when a method of giving an initial load by means of a spring or the like is used in combination, a decrease in cooling capacity becomes conspicuous unless inner pressure in a temperature-sensing portion when at 40°C is set to be 7 MPa or higher (at 2 MPa corresponding to a spring load). Accordingly, the temperature-sensing portion charging density is desirably 200 kg/m³ or higher.

Finally, an explanation is given to an acceleration of cool-down. As described above, at the start of CO₂ cycle, cooling of a temperature-sensing portion is performed by circulating a small quantity of refrigerant through a bleed hole provided near a valve part and causing the refrigerant cooled by a gas cooler to flow to a control valve, and the control valve is opened when the temperature-sensing portion is decreased in temperature and internal pressure of the temperature-sensing portion is decreased to a range of high-pressure control pressure. Accordingly, in order to accelerate cool-down, it becomes important to quickly lower the internal pressure of the temperature-sensing portion to a normal range of control pressure. In order to quickly lower the internal pressure of the temperature-sensing portion to a normal range of control pressure, it is effective to use an internal heat exchanger to set a control pressure to a little low and to decrease that density, at which a refrigerant is charged in a temperature-sensing portion of a mechanical type control valve.

FIG. 17 schematically shows effects at the time of cool-down. When a refrigerating cycle is stopped, an expansion valve in an engine room is heated to high temperature, for example, about 80°C. When the refrigerating cycle is started in this state, the valve is closed because the internal pressure of a temperature-sensing portion exceeds an upper limit pressure (in this case, 13 MPa) in operation of the cycle. Therefore, a small quantity of refrigerant cooled by a gas cooler flows through a bleed hole provided near a valve part to cool the temperature-sensing portion. At this time, a compressor is varied in capacity so as not to exceed the upper limit pressure in operation, thus controlling high pressure.

When the temperature-sensing portion is decreased in temperature and the internal pressure thereof becomes equal to or lower than the upper limit pressure in operation, the valve is opened and the compressor becomes maximum in capacity, so that the refrigerant is increased in flow rate and a maximum cooling capacity is demonstrated.

When that density, at which the refrigerant is charged in the temperature-sensing portion, is high, there is a need for cooling to a further low temperature as compared with the case where the charging density is low, in order that the internal pressure of the temperature-sensing portion become equal to or lower than the upper limit pressure in operation. Thus time (time, during which the refrigerant is small in flow rate), during which the temperature-sensing portion is cooled at the start, is prolonged and a decrease in blow-off temperature is delayed.

According to the embodiment, the above is taken into consideration and an optimum value of a temperature-sensing portion charging density in a refrigerating cycle, in which an internal heat exchanger is used, is prescribed in the following manner.

Typically, in the expansion valve 31 used in a refrigerating cycle provided with the internal heat exchanger according to the ninth embodiment illustrated with reference to FIG. 11, that density, at which a refrigerant is charged into the enclosed space A of the temperature-sensing portion of the expansion valve 31, is set in the range of about 200 kg/m³ to about 600 kg/m³. In the case where a quantity of superheat is to be increased, an upper limit value of the range of charging density may be made in the order of about 570 kg/m³, and in the case where an elastic member for biasing in a valve closing direction is used in combination, the charging density can be made in the order of about 450 kg/m³. More desirably, that density, at which a refrigerant is charged into the temperature-sensing portion of the expansion valve, is set in the range of about 200 kg/m³ to about 450 kg/m³.

For the expansion valve 31L used in a refrigerating cycle, in which the internal heat exchanger according to the seventh embodiment and illustrated with reference to FIG. 10 is used, the expansion valve being provided with no adjustment spring, it is preferable to adopt a charged refrigerant density being the same as that described above. That is, that density, at which a refrigerant is charged into the enclosed space A of the temperature-sensing portion of the expansion valve 31L and the cavity 31dL, is set in the range of about 200 kg/m³ to about 600 kg/m³. In the case where a quantity of superheat is to be increased, an upper limit value in the range of charging density may be made in the order of about 570 kg/m³ and, further, in the case where an elastic member for biasing in a valve closing direction is used in combination, the charging density can be made in the order of about 450 kg/m³. More desirably, that density, at which a refrigerant is charged into the temperature-sensing portion of the expansion valve, is set in the range of about 200 kg/m³ to about 450 kg/m³.

Also, in a refrigerating cycle, in which the internal heat exchanger according to the second, third, and fifth embodiments (FIGS. 4, 5, and 7) is provided, and a refrigerating cycle, in which the internal heat exchanger according to the fourth, sixth, and seventh embodiments (FIGS. 6, 8, 9) is provided, that density, at which a refrigerant is charged into the temperature-sensing portion of the expansion valve, is set in the range of about 200 kg/m³ to about 600 kg/m³. In the case where a quantity of superheat is to be increased, an upper limit value in the range of charging density may be in the order of about 570 kg/m³ and, further, in the case where an elastic member for biasing in a valve closing direction is used in combination, the charging density can be made in the order of about 450 kg/m³. More desirably, that density, at which a refrigerant is charged into the temperature-sensing portion of the expansion valve, is set in the range of about 200 kg/m³ to about 450 kg/m³.

Subsequently, an explanation is given to those embodiments, which solve a problem that control pressure is
varied in a temperature-sensing portion, in which a refrigerant is used in a supercritical state is used.

As shown in FIGS. 8 to 11, according to the sixth to ninth embodiments, the cavity 31a being an enclosed space is formed below the diaphragm so as to be communicated to the enclosed space A of the temperature-sensing portion and is formed above the diaphragm 32. Consequently, the enclosed space of the temperature-sensing portion is enlarged to the enclosed space A, and the cavity 31c from a conventional enclosed space A. In addition, while the charge pipe 35 is separated from the enclosed space in the foregoing explanation, it is included in the enclosed space A in this case. Accordingly, it can be said that the temperature-sensing portion according to the sixth to ninth embodiments comprises the enclosed space A and the cavity 31c. As described above, the cavity 31c increases a volume of the enclosed space, in which a refrigerant is charged, and improves the temperature-sensing portion in accuracy.

The enclosed space A is a flat space formed above the diaphragm, temperature of a refrigerant is transmitted to the enclosed space through the diaphragm, and an outer wall of the enclosed space A contacts with an outside air to be susceptible to influences of an outside air temperature. Accordingly, it can be said in the construction of the temperature-sensing portion that, the portion to which temperature of a refrigerant is transmitted and which is heated, that is, the cavity 31c below the diaphragm and a lower half of the enclosed space A in contact with the diaphragm correspond to a temperature-sensing cylinder, and an upper half of the enclosed space A susceptible to influences of an outside air temperature, corresponds to another portion different from the temperature-sensing cylinder. Accordingly, by attaching an insulating material to the outer wall portion, temperature variation of the upper half of the enclosed space A is lessened to enable ensuring a minimum temperature-sensing volume.

According to the embodiment, a ratio of the portion (here, the lower half of the enclosed space A and the cavity 31c) corresponding to the temperature-sensing cylinder to the whole temperature-sensing portion is prescribed to lessen variation in control pressure.

FIG. 18 schematically shows a temperature-sensing cylinder corresponding portion P and another portion Q. FIG. 19 shows temperature effects of the portion Q versus a ratio of the portion P to a whole volume, that is, a volume ratio of a direct temperature-sensing cylinder corresponding portion P and another portion Q in the case where the charged refrigerant density assumes a standard value of 450 kg/m³ and temperature of the portion P is 60°C. Temperature of the portion Q is 65°C, 70°C, and 80°C. A target control pressure is in the case where temperature of the portion Q is 60°C. to be the same as that of the portion P.

For example, at a point S in FIG. 19 the portion P is at 60°C, the portion Q is at 60°C, a volumetric ratio of the portion P is 50% (the ratio of 0.5), the refrigerant density at the portion P is 550 kg/m³, the refrigerant density at the portion Q is 362 kg/m³, internal pressures of the both balance at 13.51 MPa, and an average density is 450 kg/m³. A point S indicates pressures balance at the respective temperatures and the volumetric ratio.

In this manner, in the case where a refrigerant is in a supercritical state, control pressure for the expansion valve is varied by an ambient temperature in the engine room being affected by temperature of other portions than the temperature-sensing cylinder corresponding portion. Accordingly, it is necessary to lessen influences of temperature of other portions than the temperature-sensing cylinder corresponding portion.

Therefore, according to the embodiment, the volumetric ratio of the temperature-sensing cylinder corresponding portion is ensured, which amounts to a predetermined magnitude or more. Further, an insulating material may be attached to the other portion than the temperature-sensing cylinder corresponding portion to prevent heating due to an ambient temperature.

While the larger a difference between a refrigerant temperature and an ambient temperature, the more conspicuous influences of the volumetric ratio, it is necessary to decrease a change in control pressure in order to avoid an abnormally high pressure because control pressure becomes also high and a margin for an upper limit pressure of the cycle is small. In the case where a gas cooler outlet temperature is high.

While it is desired that the variation in pressure be small, it is necessary to make the variation equal to or less than about 0.5 MPa in order to make the same in the order of dispersion in a pressure sensor or the like, and assuming that a maximum temperature of a gas cooler outlet refrigerant is 65°C and temperature in the engine room is 80 to 100°C, the outer wall portion above the diaphragm rises 5 to 6°C in temperature even in the case where an insulating material is attached thereto. Accordingly, in order to make the variation equal to or less than about 0.5 MPa, it suffices to ensure a volume of at least 50% at the minimum for the volumetric ratio of the temperature-sensing cylinder corresponding portion as seen from FIG. 19.

In the case where a gas cooler outlet refrigerant temperature is low, the control pressure is low so that there is a margin for an upper limit pressure in the cycle. Since a temperature difference between the refrigerant temperature and an ambient temperature becomes large, the influence of the ambient temperature becomes large.

FIG. 20 shows effects of temperature of the portion Q (other than the temperature-sensing cylinder corresponding portion) due to an ambient temperature in the case where a refrigerant temperature is 40°C, 50°C, 60°C, 80°C, and 100°C are shown for temperature of the portion Q. A target control pressure is attained when temperature of the portion except the temperature-sensing portion is 40°C. While for example, when a refrigerant temperature is 40°C, temperature of the outer wall rises about 10°C to attain 50°C in the case where a temperature difference between the refrigerant temperature and an ambient temperature is increased to 60°C, it is found desirable to make the volumetric ratio of the temperature-sensing cylinder equal to or more than 60% in order to make variation in high pressure equal to 0.5 MPa.

Also, it can be seen from FIG. 20 that when the volumetric ratio is made equal to or more than 70%, variation in control pressure can be made equal to about 0.5 MPa even when an insulating material is omitted for the portion except the temperature-sensing cylinder corresponding portion.

Accordingly, the larger the ratio to the whole temperature-sensing portion is made by increasing a volume
(the lower half of the enclosed space above the diaphragm and the enclosed space below the diaphragm) of the temperature-sensing cylinder corresponding portion, the smaller the variation in operating value, due to an ambient temperature, can be made. According to the embodiment, the volumetric ratio of the temperature-sensing cylinder corresponding portion to the enclosed space is made equal to or more than 60%. In addition, the volumetric ratio in the embodiment is represented by the following formula

\[(V_o + 0.5V_i) / (V_o - 0.7V_i) = 0.6\]

where \(V_o\) indicates a volume of the enclosed space above the diaphragm and \(V_i\) indicates a volume of the enclosed space below the diaphragm.

[0113] Typically, the expansion valve 3G, according to the seventh embodiment, used in a refrigerating cycle with no internal heat exchanger, illustrated with reference to FIG. 9, is formed such that the volumetric sum of \(1/2\) of the enclosed space A (including the charge pipe 35b) and the cavity 31/2 amounts to at least 60% of the volumetric sum of the enclosed space A (including the charge pipe 35b) and the cavity 31. In addition, while the embodiment is directed to an expansion valve used in a refrigerating cycle with no internal heat exchanger, it may be applied to a refrigerating cycle with an internal heat exchanger.

[0114] Further, the expansion valve, according to the sixth, eighth, and ninth embodiments illustrated with reference to FIGS. 8, 10, 11 can also be formed such that the volumetric sum of \(1/2\) of the enclosed space A (including the charge pipe 35b) and the cavity 31 amounts to at least 60% of the volumetric sum of the enclosed space A (including the charge pipe 35b) and the cavity 31/2.

[0115] Further, as shown in FIG. 21, variation in control pressure can be suppressed in the expansion valve 31, according to the ninth embodiment, shown in FIG. 11 by providing a lid 39, which covers the outer wall of the temperature-sensing portion and the charge pipe 35b, and forming an air layer between the outer wall of the temperature-sensing portion and an outside air to thermally insulate a portion except the temperature-sensing cylinder corresponding portion of the temperature-sensing portion.

[0116] As described above, according to the invention, as a refrigerant temperature is transmitted to an interior of the enclosed space A through the clearance B, it is possible to omit a casing or a capillary tube and a temperature-sensing cylinder, which are used in the related art, and to achieve miniaturization and lightening of an expansion valve and reduction in cost. By composing gases, which are charged in an enclosed space, of mixed gas of a refrigerant and gases which are lower in the coefficient of thermal expansion than the refrigerant, it is possible to omit an adjustment spring or the like and to further simplify an expansion valve. Also, by prescribing that the density, at which a refrigerant is charged into a temperature-sensing body, it is possible to optimize control characteristics when an internal heat exchanger is used, and to decrease pressure-resistance of the temperature-sensing body. Further, as a ratio of a temperature-sensing body to a whole temperature-sensing cylinder corresponding portion is prescribed, it is possible to lessen the partial influences of temperature of the temperature-sensing body.

What is claimed is:

1. An expansion valve for a refrigerating cycle, arranged in a refrigerating passage leading from a gas cooler to an evaporator in a vapor compression type refrigerating cycle to adjust an opening degree of a valve port on the basis of a refrigerant temperature at an outlet side of the gas cooler to thereby control a refrigerant pressure at the outlet side of the gas cooler, the expansion valve comprising

   a temperature-sensing portion, the inner pressure of which is varied according to the refrigerant temperature at the outlet side of the gas cooler,

   a valve member that mechanically interlocks with a chance in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and

   a body that accommodates therein the valve member, and wherein the body is provided with a flow passage, through which a refrigerant reduced in pressure by the valve member is led to the evaporator while the refrigerant temperature at the outlet side of the gas cooler is transmitted to the temperature-sensing portion.

2. The expansion valve for a refrigerating cycle according to claim 1, wherein the temperature-sensing portion comprises a diaphragm, and a lid and a lower support member, which interpose therebetween a peripheral edge of the diaphragm from upper and lower directions to define an enclosed space above the diaphragm, and transmission or a refrigerant temperature to the temperature-sensing portion is performed by a clearance, which is formed by the valve member and the lower support member to be communicated to the refrigerant passage.

3. The expansion valve for a refrigerating cycle according to any one of claim 1, wherein the enclosure, space of the temperature-sensing portion is charged with a refrigerant and provided with an adjustment spring, which biases the valve member in a valve closing direction.

4. The expansion valve for a refrigerating cycle according to any one of claim 1, wherein the enclosed space of the temperature-sensing portion is charged with a mixed gas of a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, and an adjustment spring, which biases the valve member in a valve closing direction, is omitted.

5. The expansion valve for a refrigerating cycle according to any one of claim 1, further comprising a lid that covers a wall surface of the first enclosed space in contact with an outside air to provide an air layer between the wall surface and the outside air.

6. The expansion valve for a refrigerating cycle according to any one of claim 1, wherein at least a part of the wall surface of the first enclosed space in contact with an outside air is covered by a thermal insulating material.

7. An expansion valve for a refrigerating cycle arranged in a refrigerant passage leading from an internal heat exchanger to an evaporator in a vapor compression type refrigerating cycle to adjust an opening degree of a valve port on the basis of a refrigerant temperature at an outlet side of the gas cooler to thereby control a refrigerant pressure at the outlet side of the gas cooler, the expansion valve comprising

   a temperature-sensing portion, the inner pressure of which is varied according to the refrigerant temperature at the outlet side of the gas cooler,
a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and

a body that accommodates therein the valve member, and

wherin the body is provided with a first flow passage, through which a refrigerant flows to the internal heat exchanger, and a second flow passage, through which a refrigerant reduced in pressure by the valve member is led to the evaporator from the internal heat exchanger, while the refrigerant temperature at the outlet side of the gas cooler is transmitted to the temperature-sensing portion.

8. The expansion valve for a refrigerating cycle according to claim 7, wherein the temperature-sensing portion comprises a diaphragm, and a lid and a lower support member, which interpose therebetween a peripheral edge of the diaphragm from upper and lower directions to define an enclosed space above the diaphragm, and transmission of a refrigerant temperature to the temperature-sensing portion is performed by a clearance, which is formed by the valve member and the lower support member to be communicated to the refrigerant passage.

9. The expansion valve for a refrigerating cycle according to any one of claim 7, wherein the enclosed space of the temperature-sensing portion is charged with a refrigerant and provided with an adjustment spring, which biases the valve member in a valve closing direction.

10. The expansion valve for a refrigerating cycle according to any one of claim 7, wherein the enclosed space of the temperature-sensing portion is charged with a mixed gas of a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, and an adjustment spring, which biases the valve member in a valve closing direction, is omitted.

11. The expansion valve for a refrigerating cycle according to any one of claim 7, further comprising a lid that covers a wall surface of the first enclosed space in contact with an outside air to provide an air layer between the wall surface and the outside air.

12. The expansion valve for a refrigerating cycle according to any one of claim 7, wherein at least a part of the wall surface of the first enclosed space in contact with an outside air is covered by a thermal insulating material.

13. An expansion valve for a refrigerating cycle arranged in a refrigerant passage leading from an internal heat exchanger to an evaporator in a vapor compression type refrigerating cycle to adjust an opening degree of a valve port on the basis of a refrigerant temperature at an outlet side of the internal heat exchanger to thereby control a refrigerant pressure at the outlet side of the internal heat exchanger, the expansion valve comprising

a temperature-sensing portion, inner pressure of which is varied according to the refrigerant temperature at the outlet side of the internal heat exchanger, a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and

a body that accommodates therein the valve member, and

wherin the body is provided with a flow passage, through which a refrigerant reduced in pressure by the valve member flows to the evaporator while the refrigerant temperature at the outlet side of the internal heat exchanger is transmitted to the temperature-sensing portion.

14. The expansion valve for a refrigerating cycle according to claim 13, wherein the temperature-sensing portion comprises a diaphragm, and a lid and a lower support member, which interpose therebetween a peripheral edge of the diaphragm from upper and lower directions to define an enclosed space above the diaphragm, and transmission of a refrigerant temperature to the temperature-sensing portion is performed by a clearance, which is formed by the valve member and the lower support member to be communicated to the refrigerant passage.

15. The expansion valve for a refrigerating cycle according to any one of claim 13, wherein the enclosed space of the temperature-sensing portion is charged with a refrigerant and provided with an adjustment spring, which biases the valve member in a valve closing direction.

16. The expansion valve for a refrigerating cycle according to any one of claim 13, wherein the enclosed space of the temperature-sensing portion is charged with a mixed gas of a refrigerant and gases, which are lower in coefficient of thermal expansion than the refrigerant, and an adjustment spring, which biases the valve member in a valve closing direction, is omitted.

17. An expansion valve for a refrigerating cycle arranged in a refrigerant passage leading to an evaporator from a gas cooler through an internal heat exchanger in a vapor compression type refrigerating cycle to adjust an opening degree of a valve port on the basis of a refrigerant temperature at an outlet side of the gas cooler or a refrigerant temperature at an outlet side of the internal heat exchanger to thereby control a refrigerant pressure at the outlet side of the internal heat exchanger, the expansion valve comprising

a temperature-sensing portion charged with a refrigerant and varied in inner pressure according to the refrigerant temperature at the outlet side of the gas cooler or the refrigerant temperature at the outlet side of the internal heat exchanger, and

a valve member that mechanically interlocks with a change in internal pressure of the temperature-sensing portion to adjust an opening degree of the valve port, and

wherin the density, at which a refrigerant is charged in the temperature-sensing portion, is 200 to 600 kg/m³ in a valve closed state.

18. The expansion valve for a refrigerating cycle according to claim 17, wherein the density, at which a refrigerant is charged in the temperature-sensing portion, is 200 to 450 kg/m³ in a valve closed state.

19. The expansion valve for a refrigerating cycle according to claim 17, wherein the valve member is opened when high pressure at the outlet side of the gas cooler or at the outlet side of the internal heat exchanger becomes higher, by a predetermined magnitude, than inner pressure in the temperature-sensing portion.

20. The expansion valve for a refrigerating cycle according to claim 19, wherein a load corresponding to the predetermined magnitude is given by an elastic member, or a non-condensed gas charged in the temperature-sensing portion together with a refrigerant, or the elastic member and the non-condensed gas.
21. The expansion valve for a refrigerating cycle according to claim 20, wherein the elastic member comprises any one of a coil spring, a diaphragm, and a bellows, or an optional combination thereof.

22. The expansion valve for a refrigerating cycle according to claim 17, wherein when a refrigerant temperature at the outlet side of the gas cooler is 50° C. or higher, the internal heat exchanger heats a refrigerant sucked into a compressor so that superheat becomes 10° C. or higher.

23. An expansion valve for a refrigerating cycle that uses a refrigerant in a supercritical state, the expansion valve comprising a temperature-sensing portion having

a first enclosed space provided above a diaphragm and charged with a refrigerant, and

a second enclosed space provided to be communicated to the first enclosed space, and

wherein a refrigerant on an outlet side of a gas cooler, or a refrigerant on an outlet side of an internal heat exchanger is introduced below the diaphragm to apply high pressure below the diaphragm, a refrigerant temperature at the outlet side of the gas cooler, or a refrigerant temperature at the outlet side of the internal heat exchanger is transmitted to a refrigerant charged in the temperature-sensing portion, and the valve is opened and closed by that displacement of the diaphragm, which is caused by a pressure difference between above and below the diaphragm.

24. The expansion valve for a refrigerating cycle according to claim 23, wherein the second enclosed space is provided inside a valve member fixed to the diaphragm.

25. The expansion valve for a refrigerating cycle according to claim 23, wherein the sum of a half of a volume of the first enclosed space and a volume of the second enclosed space amounts to 60% or more of the sum of a volume of the first enclosed space and the second enclosed space.

26. The expansion valve for a refrigerating cycle according to any one of claim 23, further comprising a lid that covers a wall surface of the first enclosed space in contact with an outside air to provide an air layer between the wall surface and the outside air.

27. The expansion valve for a refrigerating cycle according to any one of claim 23, wherein at least a part of the wall surface of the first enclosed space in contact with an outside air is covered by a thermal insulating material.

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