

EJECTOR

FIELD OF THE INVENTION

The present invention relates to an ejector configured to draw a fluid by a jet flow of a high-speed fluid jetted from a nozzle. For example, the ejector can be suitably used for a refrigeration cycle device.

BACKGROUND OF THE INVENTION

Conventionally, an ejector is known, which includes a nozzle for decompressing and expanding a high-pressure fluid, and is configured to draw a fluid from a fluid suction port by a suction action of a jet flow of a high-speed fluid jetted from the nozzle. In the ejector, the jet fluid from the nozzle and the suction fluid from the fluid suction port are mixed in a mixing portion, and the pressure of the mixed fluid is increased in a diffuser portion by converting the kinetic energy of the mixed fluid to the pressure energy of the mixed fluid. Therefore, the pressure of the fluid flowing out of the outlet of the ejector is increased more than the pressure of the suction fluid.

In an ejector described in JP 2004-340136A (corresponding to US 2004/0206111 A1), a passage sectional area at an inlet side of a suction passage through which a suction fluid introduced from a fluid suction port flows into a mixing portion of the ejector is set equal to or larger than a passage sectional area of the fluid suction port. Therefore, the pressure loss, caused when the suction fluid is drawn from the fluid suction port, can be reduced, and the flow amount of the suction fluid flowing from the fluid suction port can be increased, thereby improving ejector efficiency η_e that is an energy converting efficiency in the ejector.

In an ejector for a refrigeration cycle device described in JP 2003-14318A (corresponding to US 2002/0000095A1), an expanding angle of a passage wall surface of a diffuser portion is suitably set in an axial section including the center axial of a nozzle so that a pressurizing amount in the diffuser portion is increased, thereby improving the ejector efficiency η_e .

In another ejector described in JP 2004-116807A, a passage wall surface of a diffuser portion is formed into a smoothly covered line in an axial section including the

center axial of a nozzle so that an energy loss such as a scroll flow loss in the diffuser portion can be restricted, thereby improving the ejector efficiency η_e .

The ejector efficiency η_e is defined as in the following formula (F1).

$$\eta_e = (1 + G_e/G_{noz}) \times (\Delta P/\rho) / \Delta i \quad \dots\dots\dots (F1)$$

Here, G_e is the flow amount of the suction fluid, G_{noz} is the flow amount of the jet fluid, ΔP is the pressurizing amount in the diffuser portion, ρ is the density of the suction fluid, and Δi is the enthalpy difference between the inlet and the outlet of the nozzle.

However, JP 2004-340136A does not describe regarding the pressure loss on a downstream side in the suction passage downstream of the fluid suction port. If the pressure loss in the suction passage changes, the flow amount of the suction fluid or the flow velocity of the fluid flowing into the mixing portion through the suction passage is changed. In addition, when the fluid flowing through the mixing portion and the diffuser portion is in a gas-liquid two-phase state, the inertial force becomes different in the gas fluid and the liquid fluid due to the density difference between the gas fluid and the liquid fluid, and thereby it is difficult to uniformly mix the jet fluid and the suction fluid in the mixing portion of the ejector.

Thus, in the diffuser portion of the ejector, the kinetic energy of the fluid is converted to the pressure energy in an inhomogeneous state, and thereby the ejector efficiency η_e cannot be sufficiently improved. Here, the inhomogeneous state means a state other than a homogeneous state that includes a complete gas state, a complete liquid state and a homogeneously mixed state in which the gas fluid and the liquid fluid are homogeneously mixed with approximately the same flow velocity. In an example of the inhomogeneously mixed state of the gas fluid and the liquid fluid, the flow velocity of the gas fluid is different from the flow velocity of the liquid fluid.

Furthermore, in JP 2003-14318A or JP 2004-116807A, the ejector is configured to improve the ejector efficiency η_e , in a case where the fluid of the homogeneous state passes through the mixing portion and the diffuser portion of the ejector. Actually, it is difficult for the gas-liquid two-phase fluid passing through the mixing portion and the diffuser portion of the ejector to be in the homogeneous state. Accordingly, when gas-liquid two-phase refrigerant passes through the mixing portion

and the diffuser portion in the ejector, it is difficult to sufficiently improve the ejector efficiency η_e .

SUMMARY OF THE INVENTION

5 In view of the foregoing problems, it is an object of the present invention to sufficiently improve the ejector efficiency η_e in an ejector having a mixing and pressurizing portion in which the kinetic energy of a gas-liquid two-phase fluid is converted to the pressure energy thereof.

10 It is another object of the present invention to provide an ejector provided with a suction passage which is configured to improve the ejector efficiency η_e .

15 The following aspects of the present invention are devised by the inventors of the present application based on the following experiments and studies. An ejector recovers the energy lost in decompression and expansion by decompressing and expanding a fluid in iso-entropy at a nozzle, and converts the recovered energy (recovery energy) to the pressure energy, so as to improve the ejector efficiency η_e .

20 If it is possible for all the recovery energy to be converted to the pressure energy, the ejector efficiency η_e will be made maximum. The inventors of the present application examined and studied in detail regarding the recovery energy used actually in the ejector. That is, the energy capable of being used for pressurizing the fluid, among the recovery energy, is studied.

25 Fig. 28 shows results examined and studied by the inventors of the present application. In an ejector of a comparative example having a mixing portion and a diffuser portion shown in Fig. 29, a total recovery energy at an inlet of a mixing portion can be divided into E1 to E4 as shown in Fig. 28. In Fig. 28, E1 indicates the energy used for pressurizing, E2 indicates a remaining kinetic energy without being used, E3 indicates energy transmission loss, and E4 indicates the other loss. As shown from Fig. 28, the energy E1 used for pressurizing is about 20% of the total recovery energy, and the other energy E2, E3, E4 is not used for pressurizing. The remaining kinetic energy E2 is remained as a flow velocity of the fluid flowing out of the diffuser portion of the ejector without being converted to the pressure energy.

30 The energy transmission loss E3 includes the energy transmission loss

caused by transmitting the kinetic energy of the liquid fluid to the gas fluid, while the liquid fluid and the gas refrigerant pass through the diffuser portion of the ejector, for example. As shown in Fig. 28, the ratio of the energy transmission loss E3, in the energy E2, E3 and E4 without being used for pressurizing, is relatively large, as compared with the energy E1 used for pressurizing.

The inventors of the present application studied regarding the reduction of the energy transmission loss E3 between the gas fluid and the liquid fluid. When the energy transmission loss E3 between the gas fluid and the liquid fluid is reduced and is used for the pressurizing, the ejector efficiency η_e can be effectively improved. Thus, the inventors performed experiments for effectively transmitting energy from the liquid fluid having a high flow velocity than that of the gas fluid, to the gas fluid.

In a case of a free fall rigid body, the flow velocity in a vertical downward direction is increased by acceleration of gravity. Then, the flow velocity of the free fall rigid body is reached to a certain terminal velocity in accordance with a balance with resistance received from the circumference air.

That is, the flow velocity of the free fall rigid body is not increased more than the terminal velocity after reaching the terminal velocity. Therefore, the flow velocity of the free fall rigid body becomes maximum when reaching to the terminal velocity. It means that the kinetic energy of the rigid body can be rapidly transmitted to the circumference air when the rigid body rapidly reaches to the terminal velocity. In Fig. 29, the liquid fluid grain (i.e., virtual liquid particle) passing through the diffuser portion is supposed as the rigid body, and the gas fluid passing through the diffuser portion is supposed as the circumference air. In the supposed state of Fig. 29, the inventors of the present application studied regarding an effective energy transmission between the liquid fluid and the gas fluid passing through the diffuser portion.

The upper part of Fig. 29 is a graph showing variation in velocity of the gas fluid and velocity of the liquid fluid within an ejector. The solid line LA1 shows a variation in the liquid fluid (e.g., liquid refrigerant) in the ejector of the comparison example, and the solid line GA1 shows a variation in the gas fluid (e.g., gas refrigerant) in the ejector of the comparison example. As shown in the solid lines LA1 and GA1 of Fig. 29, the flow velocity of the gas fluid is greatly faster than that of

the liquid fluid in a nozzle of the ejector in the comparison example by the difference in the inertial force due to the density difference between the gas fluid and the liquid fluid. Thus, in the mixed fluid of the jet fluid and the suction fluid flowing into a mixing portion of the ejector, the flow velocity of the gas fluid becomes faster than the flow velocity of the liquid refrigerant.

The grains of the liquid fluid flowing into the mixing portion are accelerated together with the circumferential gas fluid, and then the flow velocity of the grains of the liquid fluid becomes equal to the flow velocity of the gas fluid. After the flow velocity of the grains of the liquid fluid becomes equal to the flow velocity of the gas fluid, the flow velocity of the liquid fluid is not more accelerated, and reaches to the terminal velocity.

The flow velocity of the grain of the liquid fluid after reaching to the terminal velocity is reduced while applying a force corresponding to the resistance force to the circumferential gas fluid as a reaction force. At this time, the kinetic amount is transmitted from the grains of the liquid fluid to the gas fluid, and the total value of impulses applied from the grains of the liquid fluid to the gas fluid becomes the pressurizing amount (pressure energy) of the gas fluid.

Accordingly, if the grains of the liquid fluid flowing into the mixing portion of the ejector are rapidly reached to the terminal velocity, the kinetic energy included in the liquid fluid can be rapidly transmitted to the gas fluid. Thus, after the flow velocity of the liquid fluid reaches to the terminal velocity, the kinetic energy of the liquid fluid can be effectively transmitted to the gas fluid. Furthermore, when the terminal velocity itself of the grains of the liquid fluid is increased, the pressurizing amount of the gas fluid can be increased, thereby improving the ejector efficiency η_e .

In Fig. 29, the chain line LA2 indicates a variation in the flow velocity of the liquid fluid of an ejector according to an example of the present invention, and the chain line GA2 indicates a variation in the flow velocity of the gas fluid of the ejector according to the example of the present invention. As shown by the chain lines LA2 and GA2 in Fig. 29, when the flow velocity of the gas fluid flowing into the mixing portion is increased, the terminal velocity of the grains of the liquid fluid can be increased, as compared with the comparison example shown by the solid lines LA1

and GA1. Thus, in the example of the present invention shown by the chain lines LA2 and GA2 in Fig. 29, because a large amount of the kinetic energy can be converted to the pressure energy, the energy transmission loss between the gas fluid and the liquid fluid can be effectively reduced, thereby significantly improving the ejector efficiency η_e .

According to an aspect of the present invention, an ejector includes a nozzle configured to decompress and expand a fluid in any one state of a gas-liquid two-phase state, a liquid state and a super-critical state, and a body portion in which the nozzle is disposed. The body portion has a fluid suction port from which a fluid is drawn by a jet flow of the fluid jetted from a jet port of the nozzle, and a mixing and pressurizing portion in which the fluid jetted from the jet port of the nozzle and the fluid drawn from the fluid suction port are mixed and kinetic energy of the mixed fluid in a gas-liquid two-phase state is converted to pressure energy thereof. The ejector is provided with a suction passage through which the fluid drawn from the fluid suction port flows into an inlet of the mixing and pressurizing portion, and a fluid passage area of the suction passage is configured to be changed such that the fluid drawn from the fluid suction port is decompressed in the suction passage substantially in iso-entropy.

Accordingly, the energy loss while the suction fluid passes through the suction passage can be reduced. Thus, the flow velocity of the fluid flowing into the mixing and pressurizing portion from the suction passage can be increased, thereby increasing the flow velocity of the gas fluid flowing into the mixing and pressurizing portion. As a result, the terminal velocity of grains of the liquid fluid flowing into the mixing and pressurizing portion can be increased, and the pressurizing amount in the gas fluid can be increased in the ejector even when the kinetic energy of the gas-liquid two-phase fluid is converted to the pressure energy thereof in the mixing and pressurizing portion. Therefore, the ejector efficiency can be effectively improved.

According to another aspect of the present invention, an ejector includes a nozzle configured to decompress and expand a fluid in any one state of a gas-liquid two-phase state, a liquid state and a super-critical state, and a body portion in which the nozzle is disposed. The body portion has a fluid suction port from which a fluid is drawn by a jet flow of the fluid jetted from a jet port of the nozzle, and a mixing and

pressurizing portion in which the fluid jetted from the jet port of the nozzle and the fluid drawn from the fluid suction port are mixed and kinetic energy of the mixed fluid in a gas-liquid two-phase state is converted to pressure energy thereof. The ejector is provided with a suction passage through which the fluid drawn from the fluid suction port flows into an inlet of the mixing and pressurizing portion. In the ejector, a fluid passage area of the suction passage is configured to be changed such that a flow velocity of the fluid flowing into the mixing and pressurizing portion from the suction passage is substantially equal to a flow velocity of the fluid flowing from the jet port of the nozzle into the mixing and pressurizing portion. As a result, the terminal velocity of grains of the liquid fluid flowing into the mixing and pressurizing portion can be increased, and the pressurizing amount in the gas fluid can be increased in the ejector even when the kinetic energy of the gas-liquid two-phase fluid is converted to the pressure energy thereof in the mixing and pressurizing portion. Therefore, the ejector efficiency can be effectively improved. Here, the meaning of "substantially equal" includes that the flow velocity of the fluid flowing into the mixing and pressurizing portion from the suction passage completely corresponds to or slightly different from the flow velocity of the fluid flowing from the jet port of the nozzle into the mixing and pressurizing portion.

According to another aspect of the present invention, an ejector includes a nozzle configured to decompress and expand a fluid in any one state of a gas-liquid two-phase state, a liquid state and a super-critical state, and a body portion in which the nozzle is disposed. The body portion has a fluid suction port from which a fluid is drawn by a jet flow of the fluid jetted from a jet port of the nozzle, and a mixing and pressurizing portion in which the fluid jetted from the jet port of the nozzle and the fluid drawn from the fluid suction port are mixed and kinetic energy of the mixed fluid in a gas-liquid two-phase state is converted to pressure energy thereof. The ejector is provided with a suction passage through which the fluid drawn from the fluid suction port flows into an inlet of the mixing and pressurizing portion. Furthermore, a fluid passage area of the suction passage is configured to be changed such that a flow velocity of the fluid flowing into the mixing and pressurizing portion from the suction passage is equal to or larger than a sound velocity. As a result, the terminal velocity

of grains of the liquid fluid flowing into the mixing and pressurizing portion can be increased, and the pressurizing amount in the gas fluid can be increased in the ejector even when the kinetic energy of the gas-liquid two-phase fluid is converted to the pressure energy thereof in the mixing and pressurizing portion.

5 In any one aspect of the present invention, the fluid passage area of the suction passage may be gradually reduced toward downstream in a flow direction of the fluid flowing in the suction passage. In this case, a reduce degree of the fluid passage area at an inlet side of the suction passage may be larger than a reduce degree of the fluid passage area at an outlet side of the suction passage.

10 Alternatively, the fluid passage area of the suction passage at an inlet side of the suction passage may be gradually reduced toward downstream in the flow direction of the fluid flowing in the suction passage, and the fluid passage area of the suction passage at an outlet side of the suction passage may be gradually increased toward downstream in the flow direction of the fluid flowing in the suction passage.

15 The suction passage may be provided between an outer peripheral surface of the nozzle and an inner peripheral surface of the body portion, or may be configured by another nozzle to be provided therein. Alternatively, the nozzle and the suction passage may be configured, such that an enthalpy difference (ΔH) between enthalpy of the fluid at an inlet of the nozzle and enthalpy of the fluid at the jet port of the nozzle is equal to or larger than an enthalpy difference (Δh) between enthalpy of the fluid at the inlet of the suction passage and enthalpy of the fluid at the outlet of the suction passage.

20 According to another aspect of the present invention, an ejector includes a nozzle configured to decompress and expand a fluid in any one state of a gas-liquid two-phase state, a liquid state and a super-critical state, and a body portion in which the nozzle is disposed. The body portion has a fluid suction port from which a fluid is drawn by a jet flow of the fluid jetted from a jet port of the nozzle, and a mixing and pressurizing portion in which the fluid jetted from the jet port of the nozzle and the fluid drawn from the fluid suction port are mixed and kinetic energy of the mixed fluid in a gas-liquid two-phase state is converted to pressure energy thereof. The mixing and pressurizing portion is configured by a straight portion extending from the inlet of the

5 mixing and pressurizing portion in a range, and an expanding portion extending from a downstream end of the straight portion to the outlet of the mixing and pressurizing portion. The straight portion is cylindrical passage having a constant passage area in its entire range, and the expanding portion is configured such that a passage sectional area of the expanding portion is gradually increased toward downstream in a flow direction of the fluid. As a result, the terminal velocity of grains of the liquid fluid flowing into the mixing and pressurizing portion can be increased, and the pressurizing amount in the gas fluid can be increased in the ejector even when the kinetic energy of the gas-liquid two-phase fluid is converted to the pressure energy thereof in the mixing and pressurizing portion.

10 For example, the range of the straight portion may be set such that the flow velocities of gas fluid and liquid fluid within the fluid flowing into the mixing and pressurizing portion become equal to each other in the range. Alternatively, when a length of the straight portion in an axial direction of the nozzle is L1 and a length from the inlet of the mixing and pressurizing portion to the outlet of the mixing and pressurizing portion in the axial direction is L2, the mixing and pressurizing portion is configured such that $0 < L1/L2 \leq 0.4$. Furthermore, the mixing and pressurizing portion may be configured such that the fluid is pressurized in iso-entropy in the mixing and pressurizing portion.

20 In the ejector, a sectional shape of a wall surface of the expanding portion in a section including an axial line of the nozzle may be a straight line or a curved line. Alternatively, the sectional shape of the wall surface of the expanding portion in a section including the axial line of the nozzle may be formed by combining plural straight lines or may be formed by combining at least a straight line and a curved line. Alternatively, an expanding degree of the expanding portion at an inlet side of the expanding portion may be larger than an expanding degree of the expanding portion at an outlet side of the expanding portion.

BRIEF DESCRIPTION OF THE DRAWINGS

30 Additional objects and advantages of the present invention will be more readily apparent from the following detailed description of preferred embodiments

when taken together with the accompanying drawings. In which:

Fig. 1 is a schematic diagram showing a refrigeration cycle device having an ejector according to a first embodiment of the present invention;

Fig. 2A is an axial sectional view of the ejector including an axial line of a nozzle according to the first embodiment, Fig. 2B is a cross-sectional view taken along the line IIB-IIB of Fig. 2A, and Fig. 2C is a cross-sectional view taken along the line IIC-IIC of Fig. 2A;

Fig. 3 is a graph showing a variation in a ratio of a refrigerant passage sectional area of a suction passage to a refrigerant passage sectional area at an inlet of the suction passage, in the ejector according to the first embodiment;

Fig. 4 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion of the ejector according to the first embodiment;

Fig. 5A is a Mollier diagram showing a refrigerant state in a refrigerant cycle of the refrigeration cycle device according to the first embodiment, and Fig. 5B is an enlarged view showing the part VB in Fig. 5A;

Fig. 6 is a graph showing variations in the flow velocity of gas refrigerant and the flow velocity of liquid refrigerant in the ejector of the first embodiment and in an ejector of a comparison example;

Fig. 7A is a graph showing variations in a flow velocity of refrigerant and a pressurizing amount (ΔP) in the ejector according to the first embodiment, and Fig. 7B is a graph showing variations in a flow velocity of refrigerant and a pressurizing amount (ΔP) in the ejector according to a comparison example;

Fig. 8 is a graph showing an energy amount (E1) to be used for pressurizing, a remain kinetic energy (E2), an energy transmission loss (E3) and the other loss (E4), according to the first embodiment and the comparison example;

Fig. 9 is a graph showing a variation in a ratio of a refrigerant passage sectional area of a suction passage to a refrigerant passage sectional area at an inlet of the suction passage, in an ejector according to a second embodiment of the present invention;

Fig. 10 is a graph showing a variation in a ratio of a refrigerant passage sectional area of a suction passage to a refrigerant passage sectional area at an inlet

of the suction passage, in an ejector according to a third embodiment of the present invention;

Fig. 11 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion in an ejector according to a fourth embodiment of the present invention;

Fig. 12 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion in an ejector according to a fifth embodiment of the present invention;

Fig. 13 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion in an ejector according to a sixth embodiment of the present invention;

Fig. 14 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion in an ejector according to a seventh embodiment of the present invention;

Fig. 15 is a schematic diagram showing a passage configuration of a mixing and pressurizing portion in an ejector according to an eighth embodiment of the present invention;

Fig. 16 is an axial sectional view showing an ejector according to a ninth embodiment of the present invention;

Fig. 17 is an axial sectional view showing an ejector according to a tenth embodiment of the present invention;

Fig. 18 is a Mollier diagram showing a refrigerant state in a refrigerant cycle of a refrigeration cycle device according to an eleventh embodiment of the present invention;

Fig. 19 is a Mollier diagram showing a refrigerant state in a refrigerant cycle of a refrigeration cycle device according to a twelfth embodiment of the present invention;

Fig. 20 is a Mollier diagram showing a refrigerant state in a refrigerant cycle of a refrigeration cycle device according to a thirteenth embodiment of the present invention;

Fig. 21 is a Mollier diagram showing another refrigerant state in the refrigerant

cycle of the refrigeration cycle device according to the thirteenth embodiment of the present invention;

Fig. 22 is a schematic diagram showing a refrigeration cycle device having an ejector according to a fourteenth embodiment of the present invention;

Fig. 23 is a Mollier diagram showing a refrigerant state in a refrigerant cycle of the refrigeration cycle device according to the fourteenth embodiment of the present invention;

Fig. 24 is a schematic diagram showing a refrigeration cycle device having an ejector according to a fifteenth embodiment of the present invention;

Fig. 25 is a Mollier diagram showing a refrigerant state in a refrigerant cycle of the refrigeration cycle device according to the fifteenth embodiment of the present invention;

Fig. 26 is a schematic diagram showing a refrigeration cycle device having an ejector according to another embodiment of the present invention;

Fig. 27A is a Mollier diagram showing a refrigerant state in a refrigerant cycle of a refrigeration cycle device according to another embodiment of the present invention, and Fig. 27B is a Mollier diagram showing a refrigerant state in a refrigerant cycle of a refrigeration cycle device according to another embodiment of the present invention;

Fig. 28 is a graph showing energy division in a recovery energy at an inlet of a mixing portion of an ejector in a comparison example; and

Fig. 29 is a graph showing experimental results in velocity distribution of gas fluid and liquid fluid in an ejector.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

(First Embodiment)

A first embodiment of the present invention will be described with reference to Figs. 1 to 8. In the first embodiment, an ejector 16 of the present invention is typically used for a refrigeration cycle device 10 shown in Fig. 1. The refrigeration cycle apparatus 10 shown in Fig. 1 can be used for a vehicle air conditioner, for example.

In the refrigeration cycle device 10, a compressor 11 is configured to draw refrigerant, to compress the drawn refrigerant, and to discharge the compressed high-pressure and high-temperature refrigerant. The compressor 11 is driven and rotated by a vehicle engine (not shown) via an electromagnetic clutch and a belt, or the like, as an example.

The compressor 11 may be a variable displacement compressor in which a discharge capacity of the refrigerant can be continuously adjustable, or may be a fixed displacement compressor in which the discharge capacity of the refrigerant can be adjusted by changing a compressor operation ratio. For example, in the fixed displacement compressor, the compressor operation ratio is changed by interruption of the electromagnetic clutch. Alternatively, an electrical compressor may be used as the compressor 11 such that the refrigerant discharge capacity of the compressor 11 can be adjusted by adjusting a rotation speed of an electrical motor.

A refrigerant radiator 12 used as a heat exchanger for heat radiation such as a refrigerant cooler is disposed at a refrigerant discharge side of the compressor 11. The radiator 12 is configured to perform a heat exchange between the high-pressure refrigerant discharged from the compressor 11 and outside air (i.e., air outside a vehicle compartment) blown by a blower fan 12a, thereby cooling the high-pressure refrigerant in the radiator 12.

As the refrigerant used in a refrigerant cycle of the refrigeration cycle apparatus 10, a Freon-based refrigerant such as HFC134a may be used so that a refrigerant pressure on a high-pressure side in the refrigerant cycle does not exceed the critical pressure of the refrigerant. In this case, the radiator 12 is used as a condenser in which the refrigerant is cooled and condensed therein.

A receiver 12b is located at a refrigerant outlet side of the radiator 12. The receiver 12b is a gas-liquid separator with a vertically elongated tank. The receiver 12b is configured to separate the refrigerant flowing therein into gas refrigerant and liquid refrigerant, and to store therein surplus liquid refrigerant in the refrigerant cycle. The receiver 12b has a liquid refrigerant outlet at a lower side of the tank so that the liquid refrigerant flows out of the receiver 12 from the liquid refrigerant outlet. As an example of the present embodiment, the receiver 12b is formed integrally with the

radiator 12, as shown in Fig. 1. However, the receiver 12b may be located separately from the radiator 12, or may be omitted.

A branch portion 13 is connected to the liquid refrigerant outlet of the receiver 12b, and is configured to divide the liquid refrigerant from the receiver 12b into two streams. For example, the branch portion 13 is a three-way joint member having one refrigerant inlet and first and second refrigerant outlets. The three-way joint member used as the branch portion 13 may be configured by bonding pipes having different pipe diameters, or may be configured by providing plural refrigerant passages in a metal block member or a resin block member.

One of the refrigerant streams branched at the branch portion 13 flows into a first refrigerant passage 14a (nozzle-side refrigerant passage), and the other one of the refrigerant streams branched at the branch portion 13 flows into a second refrigerant passage 14b (suction-side refrigerant passage). One end of the first refrigerant passage 14a is connected to the first refrigerant outlet of the branch portion 13, and the other end of the first refrigerant passage 14a is connected to an inlet of a nozzle 16a of the ejector 16, so that one of the refrigerant streams branched at the branch portion 13 flows into the nozzle 16a through the first refrigerant passage 14a. One end of the second refrigerant passage 14b is connected to the second refrigerant outlet of the branch portion 13, and the other end of the second refrigerant passage 14b is connected to a refrigerant suction port 16d of the ejector 16, so that the other one of the refrigerant streams branched at the branch portion 13 flows into the refrigerant suction port 16d through the second refrigerant passage 14b.

An expansion valve 15 is located in the first refrigerant passage 14a at an upstream side of the nozzle 16a of the ejector 16 in a refrigerant flow of the first refrigerant passage 14a. The expansion valve 15 is used as a decompression portion configured to decompress high-pressure liquid refrigerant flowing from the receiver 12b into the first refrigerant passage 14a to be in a gas-liquid two-phase state having a middle pressure. The expansion valve 15 is also used as a flow amount adjusting portion for adjusting a flow amount of the refrigerant flowing into the nozzle 16a.

In the present embodiment, the expansion valve 15 is a thermal expansion

valve having a temperature sensing portion 15a that is located at a refrigerant suction passage of the compressor 11 so as to detect a super-heat degree of the refrigerant to be drawn into a refrigerant suction side of the compressor 11. In this embodiment, the refrigerant on the refrigerant suction side of the compressor 11 corresponds to the refrigerant at a refrigerant outlet side of a first evaporator 17. That is, the temperature sensing portion 15a detects the super-heat degree of the refrigerant at a refrigerant outlet side of the first evaporator 17 based on at least one of temperature and pressure of the refrigerant at the refrigerant outlet side of the first evaporator 17, and a valve open degree of the expansion valve 15 is adjusted using a mechanical mechanism or an electrical mechanism so that the super-heat degree of the refrigerant at the refrigerant outlet of the first evaporator 17 is approached to a predetermined value. Thus, the flow amount of the refrigerant flowing to downstream of the expansion valve 15 can be adjusted.

The other throttle structure or decompression device may be used instead of the thermal expansion valve 15. For example, a decompression device such as an electrical variable throttle device or a fixed throttle device, or the other type expansion valve may be used instead of the thermal expansion device 15.

The ejector 16 is located at a refrigerant outlet side of the expansion valve 15. The ejector 14 is adapted as a decompression portion for further decompressing the refrigerant flowing from the expansion valve 15, and as a refrigerant circulation portion for circulating the refrigerant by the suction action of a high-speed refrigerant flow jetted from the nozzle 16a. The structure of the ejector 16 will be now described with reference to Figs. 2A to 4.

Fig. 2A is an axial sectional view of the ejector 16 taken along a section including an axial line, Fig. 2B is a cross-sectional view taken along the line IIB-IIB in Fig. 2A at an inlet of a suction passage 16i of the ejector 16, and Fig. 2C is a cross-sectional view taken along the line IIC-IIC in Fig. 2A at an outlet of the suction passage 16i of the ejector 16.

As shown in Fig. 2A, the ejector 16 is configured by the nozzle 16a and a body portion 16b. The nozzle 16a is made of a metal such as a stainless alloy, and is formed into an approximately cylindrical shape having a taper end portion tapered

toward downstream in the refrigerant flow. The refrigerant passage sectional area of the nozzle 16a is changed in the refrigerant flow direction so that the refrigerant flowing into the nozzle 16a is decompressed and expanded in iso-entropy.

5 A refrigerant jet port 16c, from which the refrigerant is jetted from the nozzle 16a, is formed at a tip end of the taper end portion of the nozzle 16a. The nozzle 16a is disposed in the body portion 16b and is attached into the body portion 16b such that the refrigerant is prevented from being leaked from a fixing portion of the nozzle 16a and the body portion 16b. For example, the nozzle 16a may be air-tightly fitted into the body portion 16b, or may be air-tightly bonded to the body portion 16b by using a
10 bonding means such as welding, pressing or brazing, or the like.

For example, the nozzle 16a may be a Laval nozzle having a throat portion at which the passage sectional area becomes smallest within the refrigerant passage inside of the nozzle 16a. The nozzle 16a is configured such that the flow velocity of the refrigerant jetted from the refrigerant jet port 16c of the nozzle 16a becomes equal
15 to or larger than the sound velocity. Alternatively, the nozzle 16a may be configured by a taper nozzle so that the flow velocity of the refrigerant jetted from the refrigerant jet port 16c of the nozzle 16a becomes equal to or larger than the sound velocity.

The body portion 16b can be made of a metal, for example, aluminum or an aluminum alloy, or may be made of a material other than the metal such as resin.
20 The body portion 16b is provided with the refrigerant suction port 16b penetrating through the interior and the exterior of the body portion 16b in a radial direction perpendicular to the axial direction of the nozzle 16a. The refrigerant suction port 16b is open in the body portion 16b at a portion radially outside of the nozzle 16a. The body portion 16b has therein a mixing and pressurizing portion 16e extending in
25 the axial direction (longitudinal direction) from a position of the refrigerant jet port 16c to the refrigerant outlet (downstream end).

The refrigerant suction port 16d is coupled to a refrigerant outlet side of a second evaporator 19 so that the refrigerant from the second evaporator 19 is drawn into the suction passage 16i from the refrigerant suction port 16d. The refrigerant
30 suction port 16d is provided at an outer peripheral side of the nozzle 16a, and communicates with a space at the refrigerant jet port 16c within the body portion 16b

through the suction passage 16i.

An inlet space, into which refrigerant from the refrigerant suction port 16d flows, is provided within the body portion 16b around the refrigerant suction port 16d, and the suction passage 16i is provided between the outer wall surface of the taper end portion of the nozzle 16a and an inner wall surface of the body portion 16b. Therefore, the refrigerant flowing from the refrigerant suction port 16d into the inlet space of the body portion 16b is introduced into an inlet of the mixing and pressurizing portion 16e via the suction passage 16i. Here, the inlet of the mixing and pressurizing portion 16e in the body portion 16b substantially corresponds to the position of the refrigerant jet port 16c of the nozzle 16a in the axial direction.

Fig. 2B shows a refrigerant passage sectional area A_{in} at the inlet of the suction passage 16i, and Fig. 2C shows a refrigerant passage sectional area A_{out} at the outlet of the suction passage 16i. As shown in Figs. 2B and 2C, refrigerant passage area A_{out} at the outlet of the suction passage 16i is smaller than the refrigerant passage area A_{in} at the inlet of the suction passage 16i.

Fig. 3 shows a variation in a ratio (passage area ratio) of a refrigerant passage sectional area of the suction passage 16i in the refrigerant flow direction, to the refrigerant passage sectional area at the inlet of the suction passage 16i. As shown by the solid line in Fig. 3, the passage area ratio of the suction passage 16i is gradually reduced from the inlet to the outlet of the suction passage 16i in the refrigerant flow of the suction passage 16i. As shown in Fig. 3, a reduce degree of the passage sectional area on a side of the inlet of the suction passage 16i is larger than a reduce degree of the passage sectional area on a side of the outlet of the suction passage 16i.

Specifically, as shown by the solid-line graph of Fig. 3, the passage sectional area of the suction passage 16i is rapidly reduced in a range from the inlet of the suction passage 16i approximately to a middle position of the suction passage 16i, and the passage sectional area of the suction passage 16i is slowly reduced approximately from the middle position of the suction passage 16i to the outlet of the suction passage 16i. Thus, as compared with the comparative chain line straightly connecting the inlet and the outlet of the suction passage 16i, the variation line (i.e.,

solid line in Fig. 3) of the passage area ratio of the suction passage 16i is positioned under the comparative chain line and is made convex downwardly.

In the present embodiment, the passage sectional area of the suction passage 16i is changed as described above so that the flow velocity of the refrigerant passing through the suction passage 16i becomes equal to or greater than the sound velocity. Thus, the flow velocity of the suction refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i can be made approximately equal to the flow velocity of the jet flow jetted from the jet port 16c of the nozzle 16a into the mixing and pressurizing portion 16e. Accordingly, it is possible for the suction refrigerant can be decompressed in iso-entropy in the suction passage 16i.

As shown in Fig. 2A, the mixing and pressurizing portion 16e is positioned just downstream of the nozzle 16a and the suction passage 16i, so that the kinetic energy of gas-liquid two-phase refrigerant is converted to the pressure energy thereof in the mixing and pressurizing portion 16e while the jet refrigerant jetted from the nozzle 16a and the suction refrigerant drawn from the refrigerant suction port 16d are mixed in the mixing and pressurizing portion 16e.

The mixing and pressurizing portion 16e is configured by a straight portion 16g from the inlet of the mixing and pressurizing portion 16e to a predetermined range, and an expanding portion 16h from the downstream side of the straight portion 16g to the outlet of the ejector 16. The straight portion 16g of the mixing and pressurizing portion 16e is a cylindrical passage having a constant passage sectional area. The expanding portion 16h is gradually enlarged in the passage sectional area from its inlet toward downstream.

The straight portion 16g is provided in a range from the inlet of the mixing and pressurizing portion 16e, such that the flow velocity of the gas refrigerant and the flow velocity of the liquid refrigerant in the refrigerant flowing into the mixing and pressurizing portion 16e become approximately equal to each other in the straight portion 16e. When the length of the straight portion 16g in the axial direction of the nozzle 16a is $L1$ and when the length of the mixing and pressurizing portion 16e in the axial direction of the nozzle 16a is $L2$, a ratio $L1/L2$ is set about 0.2, as an example.

The refrigerant passage shape of the expanding portion 16h in the section

including the center line (axial line) is changed in a smoothly curved line as shown in Fig. 4. An increase degree in the refrigerant passage sectional area of the expanding portion 16h is changed as shown in Fig. 4. The increase degree on an inlet side of the refrigerant passage sectional area of the expanding portion 16h is larger than the increase degree on an outlet side of the refrigerant passage sectional area of the expanding portion 16h. That is, the increase degree on the inlet side of the refrigerant passage sectional area of the expanding portion 16h is relatively rapidly increased, and the increase degree on the outlet side of the refrigerant passage sectional area of the expanding portion 16h is relatively slowly increased, as compared with the mean increase degree from the inlet to the outlet of the expanding portion 16h.

As shown in Fig. 4, the passage wall surface of the expanding portion 16h on the section including the axial line of the expanding portion 16h, the sectional shape at the inlet side of the passage wall surface of the expanding portion 16h is formed into a curved line 101 with a slight convex toward an inner peripheral side, and the sectional shape at the outlet side of the passage wall surface of the expanding portion 16h is formed into a curved line 102 with a slight convex toward an outer peripheral side. The straight portion 16g and the expanding portion 16h of the mixing and pressurizing portion 16e are continuously extended, and are configured such that the refrigerant is substantially pressurized in iso-entropy in the mixing and pressurizing portion 16e while the refrigerant is prevented from being separated from the passage wall surface of the mixing and pressurizing portion 16e at the outlet of the mixing and pressurizing portion 16e.

Thus, the energy loss of the refrigerant while passing through the mixing and pressurizing portion 16e can be reduced, and the energy loss of the refrigerant when flowing out of the mixing and pressurizing portion 16e can be reduced. Fig. 4 is a schematic diagram for only explaining the sectional shape of the inner wall surface of the mixing and pressurizing portion 16e, and the black points in Fig. 4 are indicated only for explaining the positions of the straight portion 16g, the curved line 101 and the curved line 102 in the section shape of the mixing and pressurizing portion 16e.

As shown in Fig. 1, the first evaporator 17 is connected to the downstream

side of the mixing and pressurizing portion 16e of the ejector 16, that is, the outlet side of the expanding portion 16h of the mixing and pressurizing portion 16e. The first evaporator 17 is a heat exchanger, in which the refrigerant flowing out of the mixing and pressurizing portion 16e of the ejector 16 is heat-exchanged with air blown by a blower fan 17a, and is evaporated by absorbing heat from air passing through the first evaporator 17.

The blower fan 17a may be an electrical blower in which a fan rotation speed is controlled by a control voltage output from an air conditioning controller (not shown) so as to control an air blowing amount. A refrigerant outlet of the first evaporator 17 is coupled to the refrigerant suction port of the compressor 11.

In contrast, the second passage 14b has the one end branched from the first passage 14a at the branch portion 13, and the other end connected to the refrigerant suction port 16d of the ejector 16. A throttle unit 18 and the second evaporator 19 are located in the second passage 14b between the branch portion 13 and the refrigerant suction port 16d of the ejector 16. The throttle unit 18 is configured to function as a decompression means for decompressing the refrigerant flowing into the second evaporator 19 via the second passage 14b as well as a flow amount adjusting means for adjusting a flow amount of the refrigerant flowing into the second evaporator 19. As the throttle unit 18, a fixed throttle such as a capillary tube an orifice or the like may be used, or a variable throttle may be used.

The second evaporator 19 is located in the second passage 14b at a downstream side of the throttle unit 18, so that the refrigerant decompressed in the throttle unit 18 flows into the second evaporator 19. The second evaporator 19 is a heat exchanger, in which the refrigerant flowing out of the throttle unit 18 is heat-exchanged with air blown by a blower fan 19a, and is evaporated by absorbing heat from air passing through the second evaporator 19. The blower fan 19a may be an electrical blower, similarly to the blower fan 17a.

Fig. 5A shows a Mollier diagram showing refrigerant states in the refrigerant cycle of the refrigeration cycle device 10 with the above structure according to the first embodiment, and Fig. 5B is an enlarged view showing the part VB in Fig. 5A. When the compressor 11 is driven and is operated by a power source such as a vehicle

engine, the high-temperature and high-pressure refrigerant is discharged from the compressor 100 (point 201 in Fig. 5A), and flows into the radiator 12. The high-temperature and high-pressure refrigerant is cooled and condensed in the radiator 12 (from point 201 to point 202 in Fig. 5A).

5 The high-pressure refrigerant flowing out of the radiator 12 flows into the receiver 12b, and is separated into gas refrigerant and liquid refrigerant. The separated liquid refrigerant flowing out of the receiver 12b flows into the branch portion 13 (from point 202 to point 203 in Fig. 5A), and is branched into a refrigerant stream flowing into the first passage 14a so as to flow toward the nozzle 16a and a refrigerant stream flowing into the second passage 14b so as to flow toward the refrigerant suction port 16d.

10 A flow amount ratio G_e/G_{noz} of the flow amount G_e of the refrigerant flowing through the second passage 14b to the flow amount G_{noz} of the refrigerant flowing through the first passage 14a is determined based on flow characteristics (decompression characteristics) of the expansion valve 15, the nozzle 16a of the ejector 16 and the throttle unit 18.

15 The refrigerant flowing into the expansion valve 15 through the branched first passage 14a is decompressed and expanded in the expansion valve 15 while the flow amount of the refrigerant to flow into the nozzle 16a of the ejector 16 is adjusted by the expansion valve 15 (from point 203 to point 204 in Fig. 5A). Here, the flow amount of the refrigerant is adjusted by the expansion valve 15 so that the super-heat degree of the refrigerant at the refrigerant outlet side (the point 208 of Fig. 5A) of the first evaporator 17 is approached to a predetermined value. As shown in Fig. 5A from point 203 to point 204, the refrigerant is decompressed in iso-enthalpy in the expansion valve 15.

20 The refrigerant after being decompressed in the expansion valve 15 is further decompressed in the nozzle 16a substantially in iso-entropy while the enthalpy of the refrigerant is reduced (from point 204 to point 205 in Fig. 5A). The pressure energy of the refrigerant is converted to the speed energy of the refrigerant in the nozzle 16a so that the refrigerant is jetted from the refrigerant jet port 16c of the nozzle 16a by a high speed.

By the high-speed refrigerant stream from the refrigerant jet port 16c of the nozzle 16a, the refrigerant evaporated in the second evaporator 19 is drawn into the ejector 16 from the refrigerant suction port 16d. In Fig. 5A, ΔH indicates a reduction part of the enthalpy while the refrigerant is decompressed and expanded in iso-entropy at the nozzle 16a.

The refrigerant jetted from the nozzle 16a and the refrigerant drawn from the refrigerant suction port 16d flows into the mixing and pressurizing portion 16e positioned downstream of the nozzle 16a. The refrigerant jetted from the nozzle 16a and the refrigerant drawn from the refrigerant suction port 16d are mixed in the mixing and pressurizing portion 16e, and the speed energy of the refrigerant is converted to the pressure energy, thereby increasing the refrigerant pressure in the mixing and pressurizing portion 16e (point 205 $\rightarrow$ to point 206 $\rightarrow$ point 207 in Fig. 5A).

The refrigerant flowing out of the mixing and pressurizing portion 16e of the ejector 16 flows into the first evaporator 17. In the first evaporator 17, low-pressure refrigerant is evaporated by absorbing heat from air blown by the blower fan 17a, so that the enthalpy of the refrigerant is increased (from point 207 to point 208 in Fig. 5A). Thus, air passing through the first evaporator 17 is cooled and the cooled air can be blown into a compartment to be cooled (e.g., a vehicle compartment). The gas refrigerant flowing out of the first evaporator 17 is drawn into the compressor 11 to be compressed again by the compressor 11 (from point 208 to point 201 in Fig. 5A).

In contrast, the refrigerant stream flowing into the second passage 14b from the branch portion 13 is decompressed and expanded by the throttle unit 18 (from point 203 to point 209 in Fig. 5A), and low-pressure refrigerant decompressed by the throttle unit 18 flows into the second evaporator 19. In the second evaporator 19, low-pressure refrigerant is evaporated by absorbing heat from air blown by the blower fan 19a, so that the enthalpy of the refrigerant is increased (from point 209 to point 210 in Fig. 5A). Thus, air passing through the second evaporator 19 is cooled and the cooled air can be blown into a compartment to be cooled (e.g., the vehicle compartment).

The refrigerant after passing through the second evaporator 19 is drawn into the ejector 16 from the refrigerant suction port 16d. The refrigerant drawn from the

refrigerant suction port 16d flows into the mixing and pressurizing portion 16e of the ejector 16 through the suction passage 16i. In the present embodiment, the flow velocity of the refrigerant flowing through the suction passage 16i is greater than the sound velocity, and the refrigerant passing through the suction passage 16i is decompressed in iso-entropy as shown in Fig. 5B from point 210 to point 210'. While the refrigerant is decompressed in the suction passage 16i in iso-entropy, the enthalpy of the refrigerant is reduced by Δh .

The refrigerant flowing from the refrigerant suction port 16d into the mixing and pressurizing portion 16e through the suction passage 16i is mixed with the refrigerant jetted from the nozzle 16a in the mixing and pressurizing portion 16e (from point 210' to point 206 in Fig. 5A). Then, the refrigerant is pressurized in the mixing and pressurizing portion 16e (from point 206 to point 207 in Fig. 5A), and is supplied to the first evaporator 17 after passing through the mixing and pressurizing portion 16e.

In the refrigeration cycle device 10 having the ejector 16, the refrigerant flowing out of the mixing and pressurizing portion 16e of the ejector 16 can be supplied to the first evaporator 17 while the refrigerant decompressed by the throttle unit 18 in the second passage 14b can be supplied to the second evaporator 19 through the throttle unit 18. Thus, both the first evaporator 17 and the second evaporator 19 can be operated simultaneously to have cooling functions.

Because a refrigerant downstream side of the first evaporator 17 is connected to the refrigerant suction side of the compressor 11, the refrigerant pressurized in the mixing and pressurizing portion 16e of the ejector 16 is drawn into the compressor 11. Therefore, the suction pressure of the compressor 11 can be increased, and the driving power of the compressor 11 can be reduced. As a result, the coefficient of performance (COP) in the refrigerant cycle of the refrigeration cycle device 10 can be effectively improved.

In the ejector 16 of the first embodiment, the suction passage 16i is provided to decompress the refrigerant in iso-entropy, such that the flow velocity of the refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i is equal to or larger than the sound velocity. Therefore, the flow velocity

of the refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i can be made substantially equal to the flow velocity of the refrigerant jetted from the refrigerant jet port 16c of the nozzle 16a into the mixing and pressurizing portion 16e. Therefore, the flow velocity of the refrigerant drawn from the refrigerant suction port 16d can be increased while the energy loss of the refrigerant passing through the suction passage 16i can be reduced.

Accordingly, the flow velocity of the gas refrigerant flowing into the straight portion 16g of the mixing and pressurizing portion 16e can be increased, and thereby the terminal velocity of grains of the liquid refrigerant can be increased.

Thus, even when gas-liquid two-phase refrigerant passes through the mixing and pressurizing portion 16e in the ejector 16, the pressurizing amount of gas refrigerant can be increased in the mixing and pressurizing portion 16e, thereby improving the ejector efficiency η_e . That is, even in the ejector 16 in which the kinetic energy of gas-liquid two-phase refrigerant is converted to the pressure energy thereof, the pressurizing amount of the gas refrigerant can be effectively increased in the mixing and pressurizing portion 16e.

In the present embodiment, because the refrigerant from the refrigerant suction port 16d is decompressed in the suction passage 16i in iso-entropy as shown in Fig. 5B, the energy to be used for pressurizing can be increased by the Δh , as compared with a case where the refrigerant is decompressed in iso-enthalpy. Thus, the pressurizing amount in the mixing and pressurizing portion 16e can be increased by an amount corresponding to the Δh .

The ejector efficiency η_e' of the present embodiment can be defined as in the following formula (F2) which is different from the formula (F1).

$$\eta_e' = ((G_{noz} + G_e) \times (\Delta P / \rho)) / (G_{noz} \times \Delta i + G_e \times \Delta h) \quad \dots \dots (F2)$$

Here, G_e is the flow amount of the suction refrigerant in the suction passage 16i, G_{noz} is the flow amount of the jet refrigerant jetted from the nozzle 16a, ΔP is the pressurizing amount in the mixing and pressurizing portion 16e, ρ is the density of the suction fluid, Δi is the enthalpy difference between the inlet and the outlet of the nozzle 16a, and Δh is the energy to be used for pressurizing. As compared with the above formula (F1), the expansion energy item ($G_e \times \Delta h$) in the suction passage 16i

can be added in the denominator item (recovery energy item) in the formula (F2).

Thus, in the present embodiment, if the various configurations of the ejector 16 are set such that the same ejector efficiency η_e in formula F1 is obtained, the pressurizing amount ΔP can be increased by the recovery energy, thereby effectively improving the ejector efficiency.

In the present embodiment, the enthalpy reduction part Δh of the refrigerant while being decompressed and expanded in iso-entropy in the suction passage 16i, and the enthalpy reduction part ΔH of the refrigerant while being decompressed and expanded in iso-enthalpy in the nozzle 16a have the following relationship in the formula F3.

$$\Delta H \geq \Delta h \quad \dots\dots (F3)$$

That is, in the present embodiment, respective configurations of the ejector 16 are set to satisfy the above formula F3. That is, ΔH is the enthalpy difference between the enthalpy of the refrigerant at the inlet of the nozzle 16a and the enthalpy of the refrigerant at the refrigerant jet port 16c of the nozzle 16a, and Δh is the enthalpy difference between the enthalpy of the refrigerant at the inlet of the suction passage 16i and the enthalpy of the refrigerant at the outlet of the suction passage 16i.

According to the first embodiment of the present invention, because the refrigerant is decompressed and expanded in the suction passage 16i in iso-entropy, the flow velocity of the refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i can be increased. If the iso-entropy decompression amount of the refrigerant in the suction passage 16i is increased more than a necessary amount, the flow velocity of the refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e is unnecessarily increased as compared with the flow velocity of the refrigerant jetted from the nozzle 16a into the mixing and pressurizing portion 16e. Therefore, the energy loss may be increased while the gas refrigerant and the liquid refrigerant having different flow velocities are mixed in the mixing and pressurizing portion 16e, and thereby the ejector efficiency may be decreased.

That is, the unnecessary increased flow velocity of the refrigerant in the

suction passage 16i causes the gas refrigerants having different flow speeds to be mixed in the mixing and pressurizing portion 16e, thereby increasing the energy loss and decreasing the ejector efficiency.

Fig. 6 shows variations in the flow velocities of the gas refrigerant and liquid refrigerants within the ejector 16 when $\Delta H \geq \Delta h$ and when $\Delta H < \Delta h$. In the graphs of Fig. 6, the horizontal axis indicates axial positions in the ejector 16 from the inlet of the nozzle 16a to the outlet of the ejector 16. The upper side graph of Fig. 6 indicates the present embodiment where $\Delta H \geq \Delta h$, in which GJ2 indicates variations in the flow velocity of gas refrigerant in the jet refrigerant jetted from the nozzle 16a, GS2 indicates variations in the flow velocity of gas refrigerant in the suction refrigerant drawn from the refrigerant suction port 16d, the chain line of LIQUID indicates variations in the flow velocity of liquid refrigerant. The lower side graph of Fig. 6 indicates a comparison example where $\Delta H < \Delta h$, in which GJ1 indicates variations in the flow velocity of gas refrigerant in the jet refrigerant jetted from the nozzle 16a, GS1 indicates variations in the flow velocity of gas refrigerant in the suction refrigerant drawn from the refrigerant suction port 16d, the chain line of LIQUID indicates variations in the flow velocity of liquid refrigerant.

More specifically, the upper side graph in Fig. 6 shows the first embodiment of the present invention in which the flow velocity GS2 of the gas refrigerant in the suction refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i is approximately equal to the flow velocity GJ2 of gas refrigerant in the jet refrigerant flowing into the mixing and pressurizing portion 16e from the nozzle 16a, and thereby $\Delta H \geq \Delta h$.

In contrast, the lower side graph in Fig. 6 shows the comparison example in which the flow velocity GS1 of the gas refrigerant in the suction refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i is greatly faster than the flow velocity GJ1 of gas refrigerant in the jet refrigerant flowing into the mixing and pressurizing portion 16e from the nozzle 16a, and thereby $\Delta H < \Delta h$.

As shown in the graphs of Fig. 6, if the flow velocity of the gas refrigerant in the suction refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i is greatly faster than the flow velocity of gas refrigerant in the jet

refrigerant flowing into the mixing and pressurizing portion 16e from the nozzle 16, the flow of the gas refrigerant in the jet refrigerant is accelerated by the flow of the gas refrigerant in the suction refrigerant. When the flow velocity of the gas refrigerant in the jet refrigerant becomes equal to the flow velocity of the gas refrigerant in the suction refrigerant, the gas refrigerant in the jet refrigerant and the gas refrigerant in the suction refrigerant are joined with the same flow velocity. Then, after the gas refrigerant in the jet refrigerant and the gas refrigerant in the suction refrigerant are joined with the same flow velocity, the grains of the liquid refrigerant in the jet refrigerant are accelerated by the joined gas refrigerant.

Accordingly, the terminal velocity, at which the flow velocity of the liquid refrigerant reaches to the joined flow velocity of the gas refrigerant in the jet refrigerant and the gas refrigerant in the suction refrigerant, is positioned on a downstream side in the mixing and pressurizing portion 16e, and thereby the moving distance of the liquid refrigerant from the inlet to a position corresponding to the terminal velocity in the mixing and pressurizing portion 16e is increased. As a result, the distance from the position corresponding to the terminal velocity to the outlet of the mixing and pressurizing portion 16e, in which the kinetic energy is transmitted between the gas refrigerant and the liquid refrigerant after the grains of the liquid refrigerant reaches to the terminal velocity in the mixing and pressurizing portion 16e, becomes shorter, and thereby the refrigerant can not be sufficiently pressurized in the mixing and pressurizing portion 16e.

In contrast, according to the first embodiment of the present invention, the configurations including dimensions in the respective portions of the ejector 16 are set so that $\Delta H \geq \Delta h$. Thus, it can prevent the flow velocity of the suction refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e from being excessively increased.

More specifically, the tilt of the iso-entropy line of the gas refrigerant from the inlet to the outlet of the suction passage 16i (point 210 to point 210' of Figs. 5A, 5B) relative to a horizontal line is smaller than the tilt of the iso-entropy line of the gas refrigerant from the inlet to the outlet of the nozzle (point 204 to point 205 of Fig. 5A) relative to the horizontal line. Therefore, the decompression amount of the

refrigerant in the suction passage 16i can be accurately set smaller than the decompression amount of the refrigerant in the nozzle 16a. Thus, the refrigerant can be decompressed in the suction passage 16i by a suitable decompression amount.

As a result, the energy loss, generated while gas refrigerants having different flow velocities are mixed, can be reduced, and the refrigerant can be sufficiently pressurized in the mixing and pressurizing portion 16e, thereby effectively improving the ejector efficiency.

According to the first embodiment of the present invention, because the straight portion 16g is provided in a suitable range at a refrigerant inlet side of the mixing and pressurizing portion 16e, the energy force of the gas refrigerant can be effectively applied to the grains of the liquid refrigerant in the straight portion 16g, and thereby the flow velocity of the grains of the liquid refrigerant can rapidly reach to the terminal velocity in the straight portion 16g.

Furthermore, the kinetic energy of the liquid refrigerant having being reached to the terminal velocity can be effectively transmitted to the gas refrigerant in the expanding portion 16h. As a result, the energy transmission loss between the gas refrigerant and the liquid refrigerant in the expanding portion 16h can be reduced, and thereby the ejector efficiency can be sufficiently improved.

Fig. 7A shows the flow velocity of the gas refrigerant, the flow velocity of the liquid refrigerant, the pressurizing amount ΔP shown by the line P1, at respective positions from the inlet to the outlet of the mixing and pressurizing portion 16e of the ejector 16 according to the present embodiment. On the other hand, Fig. 7B shows the flow velocity of the gas refrigerant, the flow velocity of the liquid refrigerant, the pressurizing amount ΔP shown by the line P2, at respective positions from the inlet of a mixing portion to the outlet of a diffuser portion of an ejector of a comparative example.

As shown in Fig. 7A, the straight portion 16g is provided in a range of the mixing and pressurizing portion 16e from the inlet of the mixing and pressurizing portion 16e, such that the flow velocities of the gas refrigerant and the liquid refrigerant in the refrigerant flowing into the mixing and pressurizing portion 16e become equal at the downstream end of the straight portion 16g. That is, the

terminal velocity is caused at the downstream end of the straight portion 16g. Therefore, the kinetic energy of the refrigerant immediately after reaching to the terminal velocity can be converted to the pressure energy in the expanding portion 16h.

5 Because the flow velocity of the liquid refrigerant reaches to the terminal velocity at the inlet side of the expanding portion 16h, the energy transmission loss between the gas refrigerant and the liquid refrigerant can be effectively reduced. Thus, the flow velocity of the liquid refrigerant and the gas refrigerant at the outlet of the expanding portion 16h can be sufficiently reduced, and the ratio of energy to be
10 used actually for pressurizing can be increased.

 As a result, the pressurizing amount ΔP of the refrigerant in the mixing and pressurizing portion 16e can be increased in the present embodiment shown by the graph P1 in Fig. 7A, as compared with the comparative example shown by the graph P2 in Figs. 7A and 7B.

15 According to experiments by the inventors of the present application, when the ratio $L1/L2$ of the length $L1$ of the straight portion 16g to the length $L2$ of the mixing and pressurizing portion 16e from the inlet to the outlet of the mixing and pressurizing portion 16e is set about at 0.2, the pressurizing amount ΔP of the refrigerant in the mixing and pressurizing portion 16e can be made maximum.

20 When the ratio $L1/L2$ is set about at 0.2, the flow velocities of the gas refrigerant and the liquid refrigerant flowing out of the outlet of the straight portion 16g can be made approximately equal, and the pressurizing amount ΔP of the refrigerant in the mixing and pressurizing portion 16e can be made maximum. If the manufacturing error of the ejector 16 and the variation in the flow amount of the
25 refrigerant circulating in the refrigerant cycle of the refrigeration cycle device 10 are considered, the ejector efficiency can be sufficiently increased when $0 < L1/L2 \leq 0.4$. More preferably, the ratio $L1/L2$ is set such that $0.1 \leq L1/L2 \leq 0.3$.

 In a case where $0 < L1/L2 \leq 0.4$, the ejector efficiency can be sufficiently improved even when the gas-liquid density difference of the gas-liquid two-phase
30 refrigerant passing through the mixing and pressurizing portion 16e is changed in a wider range of 0.9 – 600 kg/m³.

In the first embodiment of the present invention, the refrigerant can be pressurized substantially in iso-entropy in the entire range of the mixing and pressurizing portion 16e, and the sectional shape of the mixing and pressurizing portion 16e is changed so as to reduce a separation from of the refrigerant at the outlet of the mixing and pressurizing portion 16e. Therefore, the energy loss of the refrigerant passing through the mixing and pressurizing portion 16e can be reduced, thereby reducing energy loss of the refrigerant flowing out of the mixing and pressurizing portion 16e.

As a result, the ratio of the energy to be actually used for pressuring can be increased among the recovery energy in the ejector 16. Fig. 8 shows energy distribution in the mixing and pressurizing portion 16e of the ejector 16 according to the first embodiment and the comparison example. In Fig. 8, E1 indicates the energy to be used for pressurizing, E2 indicates the remaining energy of the refrigerant, E3 indicates the energy transmission loss, and E4 indicates the other loss. As shown in Fig. 8, according to the first embodiment, the energy to be used for pressurizing in the mixing and pressurizing portion 16e can be greatly increased as compared with the comparison example.

(Second Embodiment)

A second embodiment of the present invention will be described with reference to Fig. 9. Fig. 9 is a diagram corresponding to Fig. 3 of the above-described first embodiment. In the second embodiment, the passage sectional area of the suction passage 16i is changed such that the ratio (passage area ratio) of the passage sectional area of the suction passage 16i to the passage sectional area at the inlet of the suction passage 16i is changed as in the straight line graph shown in Fig. 9. As shown in Fig. 9, the passage sectional area of the suction passage 16i is changed from the inlet to the outlet of the suction passage 16i by a constant degree. In the second embodiment, the other parts of the ejector 16 are similar to those in the ejector 16 of the above-described first embodiment.

According to the second embodiment of the present invention, the suction passage 16i of the ejector 16 can be configured, such that the flow velocity of the suction refrigerant flowing from the suction passage 16i into the straight portion 16g of

the mixing and pressurizing portion 16e becomes equal to or greater than the sound velocity and the suction refrigerant is decompressed in iso-entropy. Thus, the terminal velocity of the grains of the liquid refrigerant flowing into the straight portion 16g in the mixing and pressurizing portion 16e can be increased, thereby improving the ejector efficiency. In the second embodiment, the other parts of the ejector 16 are similar to those in the ejector 16 of the above-described first embodiment.

(Third Embodiment)

A third embodiment of the present invention will be described with reference to Fig. 10. Fig. 10 is a diagram corresponding to Fig. 3 of the above-described first embodiment. In the third embodiment, the passage sectional area of the suction passage 16i is changed, such that the passage sectional area at the inlet side of the suction passage 16i is gradually reduced toward downstream in the refrigerant flow direction from the inlet of the suction passage 16i, and the passage sectional area at the outlet side of the suction passage 16i is gradually increased toward downstream in the refrigerant flow direction. That is, at a predetermined portion between the inlet and the outlet of the suction passage 16i, the passage sectional area of the suction passage 16i becomes smallest, as shown in Fig. 10. A reduction ratio of the passage sectional area at the inlet side of the suction passage 16i is larger than an increase ratio of the passage sectional area at the outlet side of the suction passage 16i. At the outlet side of the suction passage 16i, the passage sectional area of the suction passage 16i is gradually increased, but is not increased more than the passage sectional area at the inlet of the suction passage 16i.

In the third embodiment, the other parts of the ejector 16 are similar to those in the ejector 16 of the above-described first embodiment.

According to the third embodiment of the present invention, the suction passage 16i of the ejector 16 is configured such that the flow velocity of the suction refrigerant flowing through the suction passage 16i becomes equal to or greater than the sound velocity at a contraction position where the refrigerant passage area becomes smallest in the suction passage 16i. Thus, the flow velocity of the suction refrigerant can be increased downstream of the contraction position in the suction passage 16i. Therefore, the terminal velocity of the grains of the liquid refrigerant

flowing into the straight portion 16g in the mixing and pressurizing portion 16e can be increased, thereby improving the ejector efficiency.

(Fourth Embodiment)

5 A fourth embodiment of the present invention will be described with reference to Fig. 11. Fig. 11 is a schematic diagram corresponding to Fig. 4 of the above-described first embodiment, showing a refrigerant passage sectional shape of the expending portion 16h in a section including the center axis of the nozzle 16a of the ejector 16. As shown in Fig. 11, the passage wall surface of the expending portion 16h is configured by combining plural straight line portions 103, 104, 105, 106,
10 107. That is, the expanding portion 16h is formed by plural cylindrical passage portions (103 to 107) each of which has a taper surface. The taper surfaces of the plural cylindrical passage portions (103 to 107) are suitably combined so as to form the expending portion 16h in the mixing and pressurizing portion 16e.

15 In the fourth embodiment, the other parts of the ejector 16 are similar to those in the ejector 16 of the above-described first embodiment.

In the structure of the expanding portion 16h according to the fourth embodiment, the energy transmission loss between the gas refrigerant and the liquid refrigerant can be reduced, thereby sufficiently improving the ejector efficiency. In the above example of the fourth embodiment, the structure of the expanding portion
20 16h is used for the ejector 16 according to the first embodiment. However, the structure of the expanding portion 16h of the fourth embodiment can be used for the ejector 16 according to any one of the second and third embodiments of the present invention.

(Fifth Embodiment)

25 A fifth embodiment of the present invention will be described with reference to Fig. 12. Fig. 12 is a schematic diagram corresponding to Fig. 4 of the above-described first embodiment, showing a refrigerant passage sectional shape of the expending portion 16h in a section including the center axis of the nozzle 16a of the ejector 16. As shown in Fig. 12, the passage wall surface of the expending portion 16h is configured by combining plural straight line portions 103, 104, 105, and
30 the curved line portion 102. That is, the expanding portion 16h is formed, by plural

cylindrical passage portions (103 to 105) each of which has a taper surface and by the cylindrical passage portion (102) having a curved surface (102).

In the fifth embodiment, the other parts of the ejector 16 are similar to those in the ejector 16 of the above-described first embodiment.

5 In the structure of the expanding portion 16h according to the fifth embodiment, the energy transmission loss between the gas refrigerant and the liquid refrigerant can be reduced, thereby sufficiently improving the ejector efficiency. In the above example of the fifth embodiment, the structure of the expanding portion 16h is used for the ejector 16 according to the first embodiment. However, the structure of the expanding portion 16h of the fifth embodiment can be used for the ejector 16 according to any one of the second and third embodiments of the present invention.

(Sixth to Eighth Embodiments)

15 A sixth embodiment of the present invention will be described with reference to Fig. 13. Fig. 13 is a schematic diagram corresponding to Fig. 4 of the above-described first embodiment, showing a refrigerant passage sectional shape of the expanding portion 16h in a section including the center axis of the nozzle 16a of the ejector 16. As shown in Fig. 13, the passage wall surface of the expanding portion 16h is configured by a single straight line portion 108 having a constant taper angle. That is, the refrigerant passage sectional area of the expanding portion 16h in the mixing and pressurizing portion 16e is gradually increased toward downstream by a constant expanding degree in the entire length of the expanding portion 16h.

25 A seventh embodiment of the present invention will be described with reference to Fig. 14. Fig. 14 is a schematic diagram corresponding to Fig. 4 of the above-described first embodiment, showing a refrigerant passage sectional shape of the expanding portion 16h in a section including the center axis of the nozzle 16a of the ejector 16. As shown in Fig. 14, the passage wall surface of the expanding portion 16h is configured by combining plural straight line portions 103, 104, 105, 106, 109. The plural straight line portions 103, 104, 105, 106, 109 are suitably combined to configure the expanding portion 16h such that the expanding degree of the refrigerant passage sectional area of the expanding portion 16h is gradually increased.

An eighth embodiment of the present invention will be described with reference to Fig. 15. Fig. 15 is a schematic diagram corresponding to Fig. 4 of the above-described first embodiment, showing a refrigerant passage sectional shape of the expanding portion 16h in a section including the center axis of the nozzle 16a of the ejector 16. As shown in Fig. 15, the passage wall surface of the expanding portion 16h is configured by a single curved line portion 110 in which the expanding angle is gradually increased as toward downstream.

In the sixth to eighth embodiments of the present invention, the other parts of the ejector 16 can be made similar to those of the ejector 16 according to the first embodiment, and the ejector efficiency can be increased. In the above examples of the sixth to eighth embodiments of the present invention, the structure of the expanding portion 16h is used for the ejector 16 according to the first embodiment. However, the structure of the expanding portion 16h according to any one of the sixth to eighth embodiments can be used for the ejector 16 of any one of the second and third embodiments of the present invention. That is, the expanding portion 16h of the mixing and pressurizing portion 16e according to any one of the fourth to eighth embodiments can be suitably used for the ejector 16 according to any one of the first to third embodiments.

(Ninth Embodiment)

In the above-described embodiments of the present invention, the ejector 16 includes the mixing and pressurizing portion 16e that is configured by the straight portion 16g and the expanding portion 16h. However, in a ninth embodiment of the present invention, an ejector 16 is configured without using the straight portion 16g, so that the mixing and pressurizing portion 16e is configured only by the expanding portion 16h as shown in Fig. 16.

Fig. 16 is an axial sectional view of the ejector 16 according to the ninth embodiment of the present invention, which corresponds to Fig. 2A of the first embodiment. That is, the inlet of the expanding portion 16h is located at the position corresponding to the refrigerant jet port 16c of the nozzle 16a in the axial direction of the nozzle 16a. The passage wall surface of the expanding portion 16h of the ejector 16 shown in Fig. 16 has the same passage sectional shape of the expanding portion

16h in the mixing and pressurizing portion 16e of the above-described first embodiment shown in Fig. 4. Thus, the inner peripheral surface of the expanding portion 16h is curved to be convex toward radially inside at the inlet side of the expanding portion 16h, and is curved to be convex toward radially outside at the outlet side of the expanding portion 16h.

Thus, even when the straight portion 16g is omitted in the mixing and pressurizing portion 16e, it is possible for the inlet side of the mixing and pressurizing portion 16e to have the same function as the straight portion 16g, thereby improving the ejector efficiency. The mixing and pressurizing portion 16e configured by only the expanding portion 16h according to the ninth embodiment can be used for the ejector 16 according to the second or the third embodiment.

Furthermore, in a case where the ejector efficiency can be sufficiently increased by increasing the flow velocity of the suction refrigerant in the suction passage 16i, the straight portion 16g described in any one of the above embodiments may be omitted from the mixing and pressurizing portion 16e.

For example, in the above-described fourth to eighth embodiments shown in Figs. 11 to 15, the mixing and pressurizing portion 16e may be configured by only the expanding portion 16h without using the straight portion 16g. In this case, the inlet of the expanding portion 16h is located at a position corresponding to the refrigerant jet port 16c of the nozzle 16a in the ejector 16 according to any one of the fourth to eighth embodiments.

(Tenth Embodiment)

In the ejector 16 according to any one of the above-described embodiments, the suction passage 16i is provided between the outer peripheral surface of the tip end portion of the nozzle 16a and the inner peripheral surface of body portion 16b. In the ejector 16 of the tenth embodiment, the nozzle 16a is used as a first nozzle 16a, and a second nozzle 16j is provided for forming a suction passage 16i through which the refrigerant drawn from a refrigerant suction port 16d flows into the mixing and pressurizing portion 16e, as shown in Fig. 17. That is, the suction passage 16i is defined by the second nozzle 16j and the refrigerant suction port 16d is provided at the inlet of the second nozzle 16j, so that the refrigerant drawn from the suction port

16d flows into the mixing and pressurizing portion 16e through the suction passage 16i.

As an example of the second nozzle 16j in the tenth embodiment, a Laval nozzle may be used. The refrigerant passage sectional area of the suction passage 16i of the second nozzle 16j can be changed similar to that of the suction passage 16i of the third embodiment. In this case, the advantages of the suction passage 16i described in the third embodiment can be obtained.

Alternatively, the second nozzle 16j can be configured by a taper nozzle such that the refrigerant passage sectional area of the suction passage 16i of the second nozzle 16j is changed similar to that of the suction passage 16i of the first or second embodiment described above. In this case, the advantages of the suction passage 16i described in the first or second embodiment can be obtained.

(Eleventh Embodiment)

In the above-described embodiments, the ejector 16 is typically used for the refrigeration cycle device 10 that is provided with the radiator 12 and the receiver 12b, for example, as shown in Fig. 1. In the refrigeration cycle device 10, the radiator 12 provided with the receiver 12b is an example of a super-cooled type condenser in which the refrigerant is cooled and condensed.

In the eleventh embodiment, the ejector 16 according to any one of the above-described embodiments is used for a refrigeration cycle device having a super-cooled type condenser that is configured by a condensation heat exchanging portion, a receiver portion and a super-cooling heat exchanging portion. Here, the condensation heat exchanging portion is configured to cool and condense the high-pressure refrigerant from the compressor 11, the receiver portion is configured to separate the refrigerant flowing from the condensation heat exchanging portion into gas refrigerant and liquid refrigerant, and the super-cooling heat exchanging portion is configured to super-cool the saturated liquid refrigerant from the receiver portion. Even in this case, the liquid refrigerant super-cooled in the super-cooling heat exchanging portion can be introduced into the branch portion 13 to be branched at the branch portion 13. The other parts of the refrigerant cycle structure in the refrigeration cycle device of the eleventh embodiment may be similar to those of the

refrigeration cycle device 10 shown in Fig. 1.

Fig. 18 is a Mollier diagram showing refrigerant states in a refrigerant cycle of the refrigeration cycle device according to the eleventh embodiment, in which the super-cooled type condenser configured by the condensation heat exchanging portion, the receiver portion and the super-cooling heat exchanging portion is used instead of the receiver 12 provided with the receiver 12b. In this case, as shown in Fig. 18, the liquid refrigerant of a super-cooled state (point 203' of Fig. 18) is branched at the branch portion 13.

Thus, the refrigerant state flowing from the expansion valve 15 into the nozzle 16a of the ejector 16 may become in a gas-liquid two-phase state (point 204 of Fig. 18) or in a liquid state (point 204' of Fig. 18). In Fig. 18, the parts corresponding to or similar to those in Fig. 5A are indicated by the same reference numbers, and the detail explanation thereof is omitted.

Even in the refrigeration cycle device with the Mollier diagram shown in Fig. 18, the ejector 16 is configured such that, the flow velocity of the suction refrigerant passing through the suction passage 16i of the ejector 16 is increased, the flow velocity of the grains of the liquid refrigerant can be rapidly reached to the terminal velocity by the straight portion 16g of the mixing and pressurizing portion 16e, and the flow velocity of the refrigerant can be sufficiently reduced in the expanding portion 16h. Thus, the ejector efficiency can be improved.

Thus, even in a case where gas-liquid two-phase refrigerant flows into the nozzle 16a or only the liquid refrigerant flows into the nozzle 16a in the refrigerant cycle, when the mixed refrigerant, in which the jet refrigerant jetted from the nozzle 16a and the suction refrigerant flowing from the suction passage 16i are mixed, is in the gas-liquid two-phase state in the ejector 16, the ejector efficiency can be effectively improved.

(Twelfth Embodiment)

In the above-described embodiments, the ejector 16 is used for the refrigerant cycle in which the expansion valve 15 is provided in the first passage 14a at an upstream side of the nozzle 16a of the ejector 16, for example, as shown in Fig. 1. However, in the twelfth embodiment, the ejector 16 is used for a refrigerant cycle of a

refrigeration cycle device in which the expansion valve 15 is omitted from the refrigeration cycle device 10 shown in Fig. 1. The other parts of the refrigeration cycle device of the twelfth embodiment are similar to those of the refrigeration cycle device 10 shown in Fig. 1. The refrigerant state of the refrigerant cycle is changed as
5 in the Mollier diagram of Fig. 19 when the refrigeration cycle device according to the twelfth embodiment is operated.

Because the expansion valve 15 is not provided in the refrigeration cycle device shown in Fig. 1, the refrigerant branched at the branch portion 13 flows into the nozzle 16a of the ejector 16 through the first passage 14a, and is decompressed and
10 expanded substantially in iso-entropy in the nozzle 16a (from point 203 to point 205 of Fig. 19). Even when the ejector 16 of the present invention is used for the refrigeration cycle device in which the refrigerant from the branch portion 13 without being decompressed is firstly decompressed and expanded in the nozzle 16a of the
15 ejector 16, the ejector efficiency can be improved similarly to that in the above-described first embodiment.

Alternatively, both the receiver 12b and the expansion valve 15 can be omitted from the refrigeration cycle device 10 shown in Fig. 1. In this case, gas-liquid two-phase refrigerant flowing out of the radiator 12 is directly branched at the branch portion 13 (point 202 of Fig. 19), and flows into the nozzle 16a of the ejector 16
20 through the first passage 14a to be decompressed and expanded substantially in iso-entropy in the nozzle 16a of the ejector 16. Alternatively, a super-cooled type condenser can be used as the radiator 12 similarly to the above described in the eleventh embodiment while the expansion valve 15 is omitted in the refrigeration cycle device 10 shown in Fig. 1. In this case, a super-cooled liquid refrigerant (point 203' of Fig. 19) flowing out of the radiator 12 is branched at the branch portion 13, and a
25 part of the branched refrigerant flows into the nozzle 16a of the ejector 16 through the first passage 14a to be decompressed substantially in iso-entropy in the nozzle 16a. (Thirteenth Embodiment)

In the above-described embodiments, the ejector 16 is used for a refrigerant cycle in which the refrigerant state flowing from the branch portion 13 into the first
30 passage 14a and the refrigerant state flowing from the branch portion 13 into the

second passage 14b are made equal. However, the ejector 16 can be used for a refrigerant cycle in which the refrigerant state flowing from the branch portion 13 into the first passage 14a and the refrigerant state flowing from the branch portion 13 into the second passage 14b are made different from each other.

5 As an example of a refrigeration cycle device of a thirteenth embodiment, the receiver 12b shown in Fig. 1 is omitted, the expansion valve 15 is located upstream of the branch portion 13, and the branch portion 13 is configured so as to change the refrigerant states (e.g., dryness) flowing into the first and second passages 14a, 14b.

10 For example, the branch portion 13 may be configured to have an interior space in which a scroll flow of the refrigerant is generated so that the dryness distributions of the refrigerant are caused in the interior space of the branch portion 13 by centrifugal force due to the scroll flow of the refrigerant.

15 The first passage 14a and the second passage 14b are connected to the branch portion 13 so that refrigerant having a predetermined dryness can be respectively introduced into the first passage 14a and the second passage 14b. Thus, the dryness of the refrigerant flowing into the first passage 14a from the branch portion 13 and the dryness of the refrigerant flowing into the second passage 14b from the branch portion 13 can be suitably changed. As the structure of the branch portion 13, the structure described in US 2007/028630 (corresponding to JP 2007-46806) can be incorporated herein by reference.

20 When the refrigeration cycle device according to the thirteenth embodiment is operated, refrigerant states circulated in the refrigerant cycle can be set to be changed as in the Mollier diagram shown in Fig. 20 or Fig. 21. In the diagram of Fig. 20, the refrigerant flowing into the nozzle 16a of the ejector 16 from the branch portion 13 through the first passage 14a is in a gas-liquid two-phase state (point 203" in Fig. 20). On the other hand, in the diagram of Fig. 21, the refrigerant flowing into the nozzle 16a of the ejector 16 from the branch portion 13 through the first passage 14a is in a liquid state (point 203' in Fig. 21).

25 Even in the refrigerant cycle with the operation states shown in Fig. 20 or Fig. 21, the ejector efficiency can be effectively improved by using the ejector 16 according to any one of the first to tenth embodiments.

(Fourteenth Embodiment)

In the above-described embodiments, the ejector 16 of the present invention is used for a sub-critical refrigerant cycle in which the pressure of refrigerant on a high-pressure side before being decompressed is lower than the critical pressure of the refrigerant. However, in a fourteenth embodiment of the present invention, the ejector 16 is used for a super-critical refrigerant cycle in which the pressure of refrigerant on the high-pressure side before being decompressed is higher than the critical pressure of the refrigerant. For example, carbon dioxide is used as the refrigerant so that the refrigerant pressure discharged from the compressor 11 becomes higher than the critical pressure of the refrigerant.

Fig. 22 shows an example of a refrigeration cycle device 10 according to the fourteenth embodiment of the present invention. In the refrigeration cycle device 10 of Fig. 22, the receiver 12b and the expansion valve 15 are omitted from the refrigeration cycle device 10 shown in Fig. 1, and a pressure control valve is used as the throttle unit 18 as compared with the refrigerant cycle device 10 shown in Fig. 1. A valve open degree of the throttle unit 18 is adjusted such that the refrigerant pressure on the high-pressure side of the refrigerant cycle of the refrigeration cycle device 10 is approached to a target pressure that is determined in accordance with a temperature of the refrigerant at a refrigerant outlet side of the radiator 12.

For example, the throttle unit 18 is provided with a temperature sensing portion 18a located at the refrigerant outlet side of the radiator 12. The temperature sensing portion 18a is configured to generate therein an inner pressure corresponding to the temperature of the refrigerant on the refrigerant outlet side of the radiator 12, so that the valve open degree of the throttle unit 18 is adjusted by a balance between the inner pressure of the temperature sensing portion 18a and the pressure of the refrigerant on the refrigerant outlet side of the radiator 12. Thus, the refrigerant pressure on the high-pressure side of the refrigerant cycle can be adjusted to the target pressure, and thereby the COP of the refrigerant cycle can be made maximum.

In the fourteenth embodiment, as shown in Fig. 22, an accumulator 20 as a low-pressure side gas-liquid separator is located at a refrigerant outlet side of the first evaporator 17 so that surplus refrigerant in the refrigerant cycle is stored in the

accumulator 20. A gas refrigerant outlet is provided in the accumulator 20 and is coupled to the refrigerant suction side of the compressor 11 so that the gas refrigerant separated from the liquid refrigerant in the accumulator 20 is supplied to the compressor 11. In the components of the refrigeration cycle device 10 shown in Fig. 22, the other parts are similar to those of the refrigeration cycle device 10 shown in Fig. 1.

When the refrigeration cycle device 10 of the present embodiment is operated, the refrigerant state is charged as in the Mollier diagram shown in Fig. 23. As shown in Fig. 23, the refrigerant is compressed in the compressor 11 to have a pressure higher than the critical pressure of the refrigerant (point 201 of Fig. 23), and is discharged to the radiator 12.

The refrigerant is cooled in the radiator 12 by performing heat exchange with outside air while keeping the refrigerant pressure at the pressure higher than the critical pressure (from point 201 to point 202 of Fig. 23). The high-pressure refrigerant flowing out of the radiator 12 is branched at the branch portion 13 into a refrigerant stream flowing into the first passage 14a and a refrigerant stream flowing into the second passage 14b.

The refrigerant flowing into the first passage 14a from the branch passage 13 flows through the nozzle 16a, the first evaporator 17 and the accumulator 20 in this order (point 202 → point 205 → point 206 → point 207 → point 208 in Fig. 23). The gas refrigerant separated at the accumulator 20 is drawn into the compressor 11.

On the other hand, the refrigerant flowing into the second passage 14b flows through the throttle unit 18 (i.e., high-pressure control valve) and the second evaporator 19 in this order, and is drawn into the ejector 16 from the refrigerant suction port 16d (point 202 → point 209 → point 210 → point 210' → point 206 in Fig. 23). The throttle unit 18 is adjusted so as to adjust the refrigerant pressure on the high pressure side from the refrigerant discharge side of the compressor 11 to the inlet of the nozzle 16a of the ejector 16 and the inlet of the throttle unit 18, such that the COP of the refrigerant cycle becomes the target pressure.

Thus, even in the refrigerant cycle of the refrigeration cycle device 10 in which the super-critical refrigerant flows into the nozzle 16a of the ejector 16, the ejector

efficiency can be improved.

Even in a case where the super-critical refrigerant flows into the nozzle 16a of the ejector 16, when the mixed refrigerant, in which the jet refrigerant jetted from the nozzle 16a and the suction refrigerant drawn from the refrigerant suction port 16d are mixed, is in a gas-liquid two-phase state in the ejector 16, the ejector efficiency can be significantly improved.

That is, when the ejector 16 is used for a super-critical refrigerant cycle in which at least the jet refrigerant jetted from nozzle 16d is in a gas-liquid two-phase state or the refrigerant downstream of the throat portion of the nozzle 16d is in a gas-liquid two-phase state, the ejector efficiency can be more significantly improved. (Fifteenth Embodiment)

A fifteenth embodiment of the present invention will be described with reference to Figs. 24 and 25. As shown in Fig. 24, in a refrigeration cycle device 10 of the fifteenth embodiment, the compressor 11 is used as a first compressor 11, and a second compressor 21 is added in the second passage 14b between the refrigerant outlet of the second evaporator 19 and the refrigerant suction port 16d of the ejector 16. Therefore, the second compressor 21 compresses the refrigerant flowing out of the second evaporator 19 and discharges the compressed refrigerant to the refrigerant suction port 16d of the ejector 16. The other components of the refrigeration cycle device 10 of the fifteenth embodiment are similar to those of the refrigeration cycle device 10 shown in Fig. 1.

For example, in the fifteenth embodiment of the present invention, the first evaporator 17 can be used for cooling the interior of a passenger compartment of a vehicle, and the second evaporator 19 can be used for cooling a cooler box (refrigerator) mounted in the vehicle. That is, the space to be cooled by the first evaporator 17 is the passenger compartment of the vehicle, and the space to be cooled by the second evaporator 19 is the interior space of the cooler box.

The basic structure of the second compressor 21 may be similar to that of the first compressor 11, and a generally known compressor may be used as the second compressor 21.

Fig. 25 is a Mollier diagram showing the refrigerant operation state of the

refrigerant cycle of the refrigeration cycle device 10, according to the fifteenth embodiment. As shown in Fig. 25, the refrigerant is compressed in the first compressor 11 to be in a high-pressure and high-temperature state (point 201 of Fig. 25), and is discharged to the radiator 12. The high-pressure and high-temperature refrigerant is cooled in the radiator 12 by performing heat exchange with outside air (from point 201 to point 202 of Fig. 25). The high-pressure refrigerant flowing out of the radiator 12 is separated into gas refrigerant and liquid refrigerant in the receiver 12b, and the separated liquid refrigerant flows into the branch portion 13 (from point 202 to point 203 of Fig. 25), similarly to Fig. 5A of the first embodiment. Then, the refrigerant is branched at the branch portion 13 into a refrigerant stream flowing into the first passage 14a and a refrigerant stream flowing into the second passage 14b.

The refrigerant flowing into the expansion valve 15 through the branched first passage 14a is decompressed and expanded in iso-enthalpy by the expansion valve 15 (from point 203 to point 204 in Fig. 25). Then, the refrigerant after being decompressed at the expansion valve 15 is further decompressed and expanded in the nozzle 16a substantially in iso-entropy while the enthalpy of the refrigerant is reduced (from point 204 to point 205 in Fig. 25). The pressure energy of the refrigerant is converted to the speed energy of the refrigerant in the nozzle 16a so that the refrigerant is jetted from the refrigerant jet port 16c by a high speed. Then, the refrigerant jetted from the refrigerant jet port 16c of the nozzle 16 is mixed in the mixing and pressurizing portion 16e with the refrigerant drawn from the refrigerant suction port 16d, so that the mixed refrigerant is pressurized in the mixing and pressurizing portion 16e (from point 206 to point 207 in Fig. 25).

The refrigerant flowing out of the mixing and pressurizing portion 16e of the ejector 16 flows into the first evaporator 17. In the first evaporator 17, low-pressure refrigerant is evaporated by absorbing heat from air blown by the blower fan 17a, so that the enthalpy of the refrigerant is increased (from point 207 to point 208 in Fig. 25). Thus, air passing through the first evaporator 17 is cooled and the cooled air can be blown into the passenger compartment. The gas refrigerant flowing out of the first evaporator 17 is drawn into the first compressor 11 to be compressed again by the first compressor 11 (from point 208 to point 201 in Fig. 25).

In contrast, the refrigerant stream flowing into the second passage 14b from the branch portion 13 is decompressed and expanded in iso-enthalpy by the throttle unit 18 (from point 203 to point 209 in Fig. 25), and low-pressure refrigerant decompressed by the throttle unit 18 flows into the second evaporator 19. In the second evaporator 19, low-pressure refrigerant is evaporated by absorbing heat from air blown by the blower fan 19a, so that the enthalpy of the refrigerant is increased (from point 209 to point 210 in Fig. 25). Thus, air passing through the second evaporator 19 is cooled so as to cool the interior of the cooler box.

In the fifteenth embodiment of the present invention, the throttled passage area of the throttle unit 18 can be set smaller than that of the first embodiment, thereby increasing the refrigerant decompression amount at the throttle unit 18. Therefore, the refrigerant evaporation pressure (refrigerant evaporation temperature) in the second evaporator 19 can be set lower as compared with the first embodiment.

As shown in Fig. 24, the refrigerant flowing out of the second evaporator 19 is drawn into the second compressor 21, and is compressed in the second compressor 21 (from point 210 to point 211 in Fig. 25). Then, the compressed refrigerant is discharged from the second compressor 21 into the refrigerant suction port 16d of the ejector 16, and is drawn into the mixing and pressurizing portion 16e of the ejector 16 from the refrigerant suction port 16d. Similarly to the first embodiment, the refrigerant is decompressed in iso-entropy while passing through the suction passage 16i (from point 211 to point 210' in Fig. 25). The other operations of the refrigeration cycle device 10 are similar to those of the above-described first embodiment.

In the refrigeration cycle device 10 having the ejector 16 according to the fifteenth embodiment, the refrigerant flowing out of the mixing and pressurizing portion 16e of the ejector 16 can be supplied to the first evaporator 17 while the refrigerant greatly decompressed by the throttle unit 18 in the second passage 14b can be supplied to the second evaporator 19 through the throttle unit 18. Thus, both the first evaporator 17 and the second evaporator 19 can be operated simultaneously to have greatly different cooling capacities, and thereby the second evaporator 19 can be used to cool the interior of the cooler box that needs a cooling temperature lower than that in the passenger compartment.

At a low outside air temperature, a pressure difference between the refrigerant pressure on the high-pressure side and the refrigerant pressure on the low-pressure side becomes smaller in the refrigerant cycle of the refrigerant cycle device 10. In this case, the flow amount of the refrigerant passing through the nozzle 16a of the ejector 16 may be decreased, and thereby the suction capacity of the ejector 13 may be decreased. Even in this case, because the second compressor 21 is located in the refrigeration cycle device 10 of the fifteenth embodiment, the suction capacity of the refrigerant into the ejector 16 from the refrigerant suction port 16d can be increased, so that the refrigerant cycle can be stably operated.

Furthermore, because the refrigerant is pressurized by using both the first and second compressors 11, 21, a pressure difference between the suction pressure and the discharge pressure in respective compressors 11, 21 can be reduced. Thus, compression efficiency of each of the first and second compressors 11, 21 can be improved, thereby improving the COP in the refrigerant cycle of the refrigeration cycle device 10.

The compression efficiency in the compressor 11, 21 is a ratio $\Delta E1 / \Delta E2$ of an increase amount $\Delta E1$ of the enthalpy of the refrigerant while being compressed in iso-entropy in the compressor 11, 21 to an increase amount $\Delta E2$ of the enthalpy of the refrigerant while being actually compressed in the compressor 11, 21. For example, when the rotational speed or the pressurizing amount of the compressor 11, 21 is increased, the refrigerant temperature is increased by the friction force, and thereby the increase amount $\Delta E2$ is increased and the compression efficiency is reduced.

Thus, in the refrigeration cycle device 10, if the pressure difference between the refrigerant pressure on the high-pressure side and the refrigerant pressure on the low-pressure side needs to be increased, the improving effect of the COP in the refrigerant cycle can be made significantly.

According to the fifteenth embodiment of the present invention, even when the ejector 16 is used for the refrigerant cycle device 10 provided with the first compressor 11 and the second compressor 21, the ejector efficiency can be sufficiently improved. Furthermore, the refrigerant suction capacity of the ejector 16 can be suitably increased by using the second compressor 21, and thereby the configuration of the

ejector 16 can be easily set.

Thus, in the present embodiment, the ejector 16 can be easily configured so as to prevent the flow velocity of the suction refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e from being unnecessarily increased. That is, in the present embodiment, because the flow velocity of the refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e can be changed by not only the decompression characteristics in the suction passage 16i but also the discharge refrigerant pressure of the second compressor 21, the suction passage 16i of the ejector 16 can be easily formed. Therefore, the flow velocity of the suction refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e can be easily adjusted by adjusting the refrigerant pressure at the refrigerant suction port 16d of the ejector 16.

According to the fifteenth embodiment, by adjusting the refrigerant discharge capacity of the second compressor 21, the flow velocity of the suction refrigerant flowing from the suction passage 16i into the mixing and pressurizing portion 16e can be easily adjusted at a suitable velocity, relative to the flow velocity of the jet refrigerant jetted from the refrigerant jet port 16c of the nozzle 16. As a result, the configurations of respective parts in the ejector 16 can be easily set, and thereby the ejector 16 can be easily formed.

(The Other Embodiments)

Although the present invention has been fully described in connection with the preferred embodiments thereof with reference to the accompanying drawings, it is to be noted that various changes and modifications will become apparent to those skilled in the art.

According to any one embodiment of the present invention, the ejector 16 is provided with the suction passage 16i through which the refrigerant (fluid) drawn from the refrigerant suction port 16d flows into the inlet of the mixing and pressurizing portion 16e. The passage sectional area of the suction passage 16i is configured to be changed such that the refrigerant (fluid) drawn from the refrigerant suction port 16d is decompressed in the suction passage 16i substantially in iso-entropy. Alternatively, the passage area of the suction passage 16i is configured to be changed such that the

flow velocity of the refrigerant flowing into the mixing and pressurizing portion 16e from the suction passage 16i is substantially equal to the flow velocity of the refrigerant (fluid) flowing from the jet port 16c of the nozzle 16a into the mixing and pressurizing portion 16e. Alternatively, the passage sectional area of the suction passage 16i is configured to be changed such that the flow velocity of the fluid flowing into the mixing and pressurizing portion 16e from the suction passage 16i is equal to or larger than the sound velocity. In this case, the ejector efficiency can be effectively improved. The other configurations in the ejector 16 may be suitably changed or combined without being limited to the above-described embodiments.

According to any one embodiment of the present invention, the mixing and pressurizing portion 16e is configured by the straight portion 16g extending from the inlet of the mixing and pressurizing portion 16e in a range in the axial direction, and the expanding portion 16h extending continuously from a downstream end of the straight portion 16g to the outlet of the mixing and pressurizing portion 16e. The straight portion 16g is a cylindrical passage having a constant passage area in its entire range, and the expanding portion 16h is configured such that a passage sectional area of the expanding portion 16h is gradually increased toward downstream in the flow direction of the refrigerant. In the ejector 16, the other configurations may be suitably changed or combined without being limited to the above-described embodiments. For example, the range of the straight portion 16g in the axial direction of the nozzle 16a is set such that the flow velocities of gas refrigerant and liquid refrigerant within the refrigerant flowing into the mixing and pressurizing portion 16e become equal to each other in the range. Alternatively, when the length of the straight portion 16g in the axial direction of the nozzle is $L1$ and the length from the inlet of the mixing and pressurizing portion 16e to the outlet of the mixing and pressurizing portion 16e in the axial direction is $L2$, the mixing and pressurizing portion 16e is configured such that $0 < L1/L2 \leq 0.4$. Alternatively, the mixing and pressurizing portion 16e may be configured such that the refrigerant is pressurized in iso-entropy therein.

In the above-described embodiments of the present invention, the ejector 16 is used for the refrigeration cycle device 10 in which the refrigerant is branched at the

branch portion 13 on an upstream side of the nozzle 16a in the refrigerant flow from the radiator 12. However, the ejector 16 of the present invention can be used for a refrigeration cycle device without being limited to the examples of the above-described embodiments.

5 For example, the ejector 16 of the present invention can be used for a refrigeration cycle device shown in Fig. 26. The refrigeration cycle device shown in Fig. 26, an accumulator 20 is located downstream of the outlet of the ejector 16 so that the refrigerant flowing out of the ejector 16 can directly flow into the accumulator 20. The accumulator 20 has a gas refrigerant outlet coupled to the refrigerant suction side of the compressor 11, and a liquid refrigerant outlet connected to a refrigerant inlet of an evaporator 19 so that the liquid refrigerant separated from the gas refrigerant in the accumulator 20 flows into the evaporator 19. The gas refrigerant evaporated in the evaporator 19 is drawn into the refrigerant suction port 16d of the ejector 16. In the refrigeration cycle device shown in Fig. 26, the refrigerant flowing from the radiator 12 is decompressed in the nozzle 16a and the gas refrigerant from the evaporator 19 is drawn into the ejector 16 from the refrigerant suction port 16d by the high-speed jet flow from the nozzle 16a. Even when the ejector 16 according to any one of the first to tenth embodiments is used for the refrigeration cycle device shown in Fig. 26, the ejector efficiency can be improved.

20 In the example shown in Figs. 5A and 5B, the suction gas refrigerant is decompressed in iso-entropy in the suction passage 16i; however, the suction gas refrigerant is not limited to be decompressed in iso-entropy.

Figs. 27A and 27B are modified examples of Fig. 5B. X and Y in Fig. 27A and Fig. 27B correspond to the enlarged part VB in Fig. 5A. As shown in Fig. 27A, gas-liquid two-phase refrigerant can be drawn from the refrigerant suction port 16b and can be decompressed in iso-entropy in the suction passage 16i of the ejector 16. Alternatively, as shown in Fig. 27B, the gas refrigerant is drawn from the refrigerant suction port 16d, and can be decompressed in iso-entropy into a gas-liquid two-phase state.

30 In the above-described embodiments, the Freon-based refrigerant or the carbon dioxide is typically used as the refrigerant. However, as the refrigerant, a

generally-known refrigerant or a generally known fluid may be used. For example, carbon-hydride based refrigerant may be used as the refrigerant.

5 In the above-described embodiments, the refrigerant cycle device is used for a vehicle air conditioner or for a vehicle refrigerator. However, the refrigeration cycle device may be used for a fixed cooler, a fixed refrigerator, a box having a cooling function, a cooling device for a coin machine or the like.

10 In the above-described embodiments, the first and second evaporators 17, 19 are used as an interior heat exchanger for cooling air, and the radiator 12 is used as an exterior heat exchanger for radiating heat to outside air. However, the first and second evaporators 17, 19 may be used as an exterior heat exchanger for absorbing heat from outside air, and the radiator 12 may be used as an interior heat exchanger for heating a fluid to be heated such as water or air. That is, the ejector 16 of the present invention can be used for a heat pump cycle system with a heating function or/and a cooling function.

15 Such changes and modifications are to be understood as being within the scope of the present invention as defined by the appended claims.

What is claimed is:

1. An ejector comprising:

5 a nozzle configured to decompress and expand a fluid in any one state of a gas-liquid two-phase state, a liquid state and a super-critical state; and

a body portion in which the nozzle is disposed, the body portion having a fluid suction port from which a fluid is drawn by a jet flow of the fluid jetted from a jet port of the nozzle, and a mixing and pressurizing portion in which the fluid
10 jetted from the jet port of the nozzle and the fluid drawn from the fluid suction port are mixed and kinetic energy of the mixed fluid in a gas-liquid two-phase state is converted to pressure energy thereof, wherein

the mixing and pressurizing portion is configured by a straight portion extending from the inlet of the mixing and pressurizing portion in a range, and an
15 expanding portion extending from a downstream end of the straight portion to the outlet of the mixing and pressurizing portion,

the straight portion is a cylindrical passage having a constant passage area in its entire range, and

the expanding portion is configured such that a passage sectional area of
20 the expanding portion is gradually increased toward downstream in a flow direction of the fluid.

2. The ejector according to claim 1, wherein the range of the straight
25 portion is set such that the flow velocities of gas fluid and liquid fluid within the fluid flowing into the mixing and pressurizing portion become equal to each other in the range.

3. The ejector according to claim 1 or 2, wherein when a length of the straight portion in an axial direction of the nozzle is L1 and a length from the inlet of the mixing and pressurizing portion to the outlet of the mixing and pressurizing portion in the axial direction is L2, the mixing and pressurizing portion is configured such that $0 < L1/L2 < 0.4$.

4. The ejector according to any one of claims 1 to 3, wherein the mixing and pressurizing portion is configured such that the fluid is pressurized in iso-entropy in the mixing and pressurizing portion.

5. The ejector according to any one of claims 1 to 4, wherein a sectional shape of a wall surface of the expanding portion in a section including an axial line of the nozzle is a straight line.

6. The ejector according to any one of claims 1 to 4, wherein a sectional shape of a wall surface of the expanding portion in a section including an axial line of the nozzle is a curved line.

7. The ejector according to any one of claims 1 to 4, wherein a sectional shape of a wall surface of the expanding portion in a section including an axial line of the nozzle is formed by combining plural straight lines that continuously extend in order.

8. The ejector according to any one of claims 1 to 4, wherein a sectional shape of a wall surface of the expanding portion in a section including an axial line of the nozzle is formed by combining at least a straight line and a curved line that continuously extend in order.

5

9. The ejector according to any one of claims 1 to 4, wherein an expanding degree of the expanding portion at an inlet side of the expanding portion is larger than an expanding degree of the expanding portion at an outlet side of the expanding portion.

10

10. The ejector according to claim 1, further comprising
a suction passage through which the fluid drawn from the fluid suction port
flows into an inlet of the mixing and pressurizing portion,
wherein a fluid passage area of the suction passage is configured to be
changed such that the fluid drawn from the fluid suction port is decompressed in
the suction passage substantially in iso-entropy.

20

11. The ejector according to claim 1, further comprising
a suction passage through which the fluid drawn from the fluid suction port
flows into an inlet of the mixing and pressurizing portion,
wherein a fluid passage area of the suction passage is configured to be
changed such that a flow velocity of the fluid flowing into the mixing and
pressurizing portion from the suction passage is substantially equal to a flow
velocity of the fluid flowing from the jet port of the nozzle into the mixing and
pressurizing portion.

25

12. The ejector according to claim 1, further comprising
a suction passage through which the fluid drawn from the fluid suction port
flows into an inlet of the mixing and pressurizing portion,

5 wherein a fluid passage area of the suction passage is configured to be
changed such that a flow velocity of the fluid flowing into the mixing and
pressurizing portion from the suction passage is equal to or larger than a sound
velocity.