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Ejector decompression device with throttle controllable nozzle
Ejektor zur Druckverminderung mit verstellbarer Drosseldüse
Ejecteur à decompression avec soupape d’étranglement controllable

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Description

[0001] The present invention relates to an ejector decompression device for a vapor compression refrigerant cycle. More specifically, the present invention relates to an ejector with a throttle controllable nozzle in which a throttle degree can be controlled.

[0002] In an ejector cycle, pressure of refrigerant to be sucked into a compressor is increased by converting expansion energy to pressure energy in a nozzle of an ejector, thereby reducing motive power consumed by the compressor. Further, refrigerant is circulated into an evaporator by using a pumping function of the ejector. However, when energy converting efficiency of the ejector, that is, ejector efficiency \( \eta_e \) is reduced, the pressure of refrigerant to be sucked to the compressor cannot be sufficiently increased by the ejector. In this case, the motive power consumed by the compressor cannot be satisfactorily reduced. On the other hand, a throttle degree (passage opening degree) of the nozzle of the ejector is generally fixed. Therefore, when an amount of refrigerant flowing into the nozzle changes, the ejector efficiency \( \eta_e \) is changed in accordance with the change of the refrigerant flowing amount. Further, according to experiments by the inventors of the present invention, if the throttle degree of the nozzle is simply changed, the ejector efficiency \( \eta_e \) may be greatly reduced due to a refrigerant flow loss of a control mechanism for controlling the throttle degree.


[0004] In view of the foregoing problems, it is a first object of the present invention to provide an ejector decompression device having a throttle controllable nozzle with an improved structure.

[0005] It is a second object of the present invention to variably control a throttle degree of a nozzle of the ejector decompression device without largely reducing ejector efficiency \( \eta_e \) of the ejector decompression device.

[0006] These objects are achieved by the features in claim 1.

[0007] According to the present invention, an ejector decompression device for a refrigerant cycle includes a nozzle for decompressing and expanding refrigerant flowing from a radiator by converting pressure energy of refrigerant to speed energy of the refrigerant, a pressure-increasing portion that is disposed to increase a pressure of refrigerant by converting the speed energy of refrigerant to the pressure energy of refrigerant while mixing refrigerant injected from the nozzle and refrigerant sucked from an evaporator of the refrigerant cycle, and a needle valve disposed to be displaced in a refrigerant passage of the nozzle in an axial direction of the nozzle for adjusting an opening degree of the refrigerant passage of the nozzle. Here, the refrigerant passage is defined by an inner wall of the nozzle. Further, the nozzle includes a throat portion having a cross-sectional area that is smallest in the refrigerant passage of the nozzle, and an expansion portion in which the cross-sectional area is increased from the throat toward downstream in a refrigerant flow. In the ejector decompression device, the needle valve and the inner wall of the nozzle are provided to have predetermined shapes so that refrigerant flowing into the nozzle is decompressed to a gas-liquid two-phase state at upstream from the throat portion in the refrigerant flow. In the present invention, because refrigerant is decompressed to the gas-liquid state at upstream from the throat portion, refrigerant bubbles are generated, and a mass density of the refrigerant is reduced. Accordingly, the cross-sectional area of the refrigerant passage is relatively reduced in the nozzle. Thus, the flow amount of refrigerant can be adjusted, and the refrigerant passage can be prevented from being throttled more than a necessary degree. As a result, ejector efficiency \( \eta_e \) can be prevented from being largely reduced in the ejector decompression device having the nozzle where the opening degree of the refrigerant passage can be variably controlled.

[0008] Alternatively, the needle valve is disposed in the refrigerant passage of the nozzle to define a throttle portion having a cross-sectional area that is smallest in a space between the needle valve and the inner wall of the nozzle, and the throttle portion is positioned upstream from the throat portion in the refrigerant flow. Therefore, rectified refrigerant with a small disturbance can pass through the throat portion, and is sufficiently accelerated more than the sound speed while flowing through the extension portion. Because the refrigerant can be accurately sufficiently accelerated in the nozzle, the ejector efficiency can be effectively improved.

[0009] The needle valve has a downstream portion that is tapered toward a downstream end of the needle valve so that a cross-sectional area of the downstream portion of the needle valve is reduced toward the downstream end, and the inner wall of the nozzle is formed into an approximate cone shape having at least two different taper angles, upstream from the throat portion. Further, the inner wall of the nozzle has a radial dimension that is reduced toward the throat portion. Alternatively, the inner wall of the nozzle has a radial dimension that is reduced from an upstream end of the nozzle toward the throat portion and is increased from the throat portion toward a downstream end of the nozzle.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010] 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 an ejector cycle according to a first preferred embodiment of
the present invention;
FIG. 2 is a schematic diagram showing an ejector according to the first embodiment;
FIG. 3A is an enlarged schematic diagram showing a refrigerant flow in a nozzle of the ejector according to the first embodiment, and FIG. 3B is an enlarged schematic diagram showing an inner wall shape of the nozzle shown in FIG. 3A;
FIG. 4 is an enlarged schematic diagram for explaining an operational effect of the nozzle of the ejector according to the first embodiment;
FIG. 5 is a bar graph showing a comparison between efficiency of the ejector according to the first embodiment and efficiency of a reference ejector;
FIG. 6 is an enlarged schematic diagram for explaining a trouble in a nozzle of a reference ejector;
FIG. 7 is an enlarged schematic diagram for explaining a trouble in a nozzle of another reference ejector;
FIG. 8 is an enlarged schematic diagram showing a nozzle not within the scope of the claimed invention; and
FIG. 9A is an enlarged schematic diagram showing a nozzle not within the scope of the claimed invention, and FIG. 9B is a graph showing a sectional area change in a refrigerant passage of the nozzle shown in FIG. 9A and in a mixing portion and a diffuser shown in FIG. 2 in an axial direction of the nozzle.

DETAILED DESCRIPTION OF THE PRESENTLY PREFERRED EMBODIMENTS

[0011] Preferred embodiments of an ejector decompression device will be described hereinafter with reference to the appended drawings.

(First Embodiment)

[0012] In the first embodiment, as shown in FIG. 1, an ejector for an ejector cycle is typically used for a heat pump cycle for a water heater. In the ejector cycle, the ejector is used as a decompression device for decompressing refrigerant. In the heat pump cycle shown in FIG. 1, a compressor 10 sucks and compresses refrigerant, and a radiator 20 cools the refrigerant discharged from the compressor 10. Specifically, the radiator 20 is a high-pressure heat exchanger that heats water for the water heater by heat-exchange between the refrigerant flowing from the compressor 10 and the water. The compressor 10 is driven by an electric motor (not shown), and a rotation speed of the compressor 10 can be controlled. A flow amount of refrigerant discharged from the compressor 10 is increased by increasing the rotational speed of the compressor 10, thereby increasing heating performance of the water in the radiator 20. On the contrary, the flow amount from the compressor 10 is reduced by reducing the rotational speed of the compressor 10, thereby reducing the heating performance of the water in the radiator 20.

[0013] In the first embodiment, since freon is used as refrigerant, refrigerant pressure in the radiator 20 is equal to or lower than the critical pressure of the refrigerant, and the refrigerant is condensed in the radiator 20. However, the other refrigerant such as carbon dioxide may be used as the refrigerant. When carbon dioxide is used as the refrigerant, the refrigerant pressure in the radiator 20 becomes equal to or higher than the critical pressure of refrigerant, and the refrigerant is cooled without being condensed in the radiator 20. In this case, a temperature of refrigerant is reduced from an inlet of the radiator 20 toward an outlet of the radiator 20. An evaporator 30 evaporates liquid refrigerant. Specifically, the evaporator 30 is a low-pressure heat exchanger that evaporates the liquid refrigerant by absorbing heat from outside air in heat-exchange operation between the outside air and the liquid refrigerant. An ejector 40 sucks refrigerant evaporated in the evaporator 30 while decompressing and expanding refrigerant flowing from the radiator 20, and increases pressure of refrigerant to be sucked into the compressor 10 by converting expansion energy to pressure energy.

[0014] A gas-liquid separator 50 separates the refrigerant from the ejector 40 into gas refrigerant and liquid refrigerant, and stores the separated refrigerant therein. The gas-liquid separator 50 includes a gas-liquid separator outlet connected to a suction port of the compressor 10, and a liquid-liquid separator outlet connected to an inlet of the evaporator 30. Accordingly, in the ejector cycle (heat pump cycle), liquid refrigerant flows into the evaporator 30 while refrigerant from the radiator 20 is decompressed in a nozzle 41 of the ejector 40.

[0015] Next, the structure of the ejector 40 will be described in detail with reference to FIGS. 2, 3A, 3B. As shown in FIG. 2, the ejector 40 includes the nozzle 41, a mixing portion 42 and a diffuser 43. The nozzle 41 decompresses and expands high-pressure refrigerant from the radiator 20 by converting pressure energy of the high-pressure refrigerant to speed energy. Gas refrigerant from the evaporator 30 is sucked into the mixing portion 42 by a high speed stream of refrigerant injected from the nozzle 41, and the sucked gas refrigerant and the injected refrigerant are mixed in the mixing portion 42. The diffuser 43 increases refrigerant pressure by converting the speed energy of refrigerant to the pressure energy of the refrigerant while mixing the gas refrigerant sucked from the evaporator 30 and the refrigerant injected from the nozzle 41.

[0016] In the mixing portion 42, the refrigerant jetted from the nozzle 41 and the refrigerant sucked from the evaporator 30 are mixed so that the sum of their momentum of two-kind refrigerant flows is conserved. Therefore, static pressure of refrigerant is increased also in the mixing portion 42. Because a sectional area of a refrigerant passage in the diffuser 43 is gradually increased, dynamic pressure of refrigerant is converted to static pressure of refrigerant in the diffuser 43. Thus, refrigerant pressure is increased in both of the mixing portion 42 and the dif-
fuser 43. Accordingly, in the first embodiment, the mixing portion 42 and the diffuser 43 define a pressure-increasing portion. Theoretically, in the ejector 40, refrigerant pressure is increased in the mixing portion 42 so that the total momentum of two-kind refrigerants flows are conserved in the mixing portion 42, and refrigerant pressure is increased in the diffuser 43 so that total energy of refrigerant is conserved in the diffuser 43.

The nozzle 41 is a laburl nozzle (refer to Fluid Engineering published by Tokyo University Publication) having a throat portion 41a and an expansion portion 41b. Here, a cross-sectional area of the throat portion 41a is smallest in a refrigerant passage of the nozzle 41. As shown in FIG. 3A, an inner radial dimension d2 of the expansion portion 41b is gradually increased from the throat portion 41a toward a downstream end of the nozzle 41. As shown in FIG. 2, a needle valve 44 is displaced by an actuator 45 in an axial direction of the nozzle 41, so that an open degree of the throat portion 41a is adjusted. That is, the throttle degree of the refrigerant passage in the nozzle 41 is adjusted by the displacement of the needle valve 44. In the first embodiment, an electric actuator such as a linear solenoid motor and a stepping motor including a screw mechanism is used as the actuator 45, and pressure of high-pressure refrigerant is detected with a pressure sensor (not shown). Then, the open degree of the throat portion 41a is adjusted so as to control the detected pressure at a predetermined pressure.

The needle valve 44 is disposed upstream of the throat portion 41a in the refrigerant passage of the ejector 40. Further, as shown in FIG. 3A, a taper portion of the needle valve 44 and an inner wall surface of the nozzle 41 are formed so that a throttle portion 41c is formed upstream from the throat portion 41a, so that refrigerant from the radiator 20 is decompressed into a gas-liquid two-phase state at the upstream of the throat portion 41a. Here, a cross-sectional area of the throat portion 41c is determined by the needle valve 44 and the nozzle 41, and is smallest in the refrigerant passage of the nozzle 41. Specifically, as shown FIG. 3B, the inner wall surface of the nozzle 41 has at least two taper angles α1, α2 (refer to Japanese Industrial Standards B 0612), and is formed in a two-step taper shape so that an inner radial dimension d1 is reduced toward the throat portion 41a. Further, a top end portion of the needle valve 44 is formed in an approximate cone shape so that a cross-sectional area of the needle valve 44 is reduced toward the top end thereof.

Next, operational effects of the ejector 40 according to the first embodiment will be now described. As shown in FIGS. 3A, 3B, the sectional area of the refrigerant passage, defined by the nozzle 41 and the needle valve 44, reduces toward the throttle portion 41c. Therefore, a flow speed of refrigerant, flowing from the radiator 20 into the nozzle 41, increases toward the throttle portion 41c while a flow amount of the refrigerant becomes a flow amount determined by the open degree of the nozzle 41. On the other hand, the sectional area of the refrigerant passage is slightly increased from the throttle portion 41c to the downstream end of the needle valve 44. However, an increase rate of the sectional area is a little in the refrigerant passage from the throttle portion 41c to the downstream end of the needle valve 44, as compared with the expansion portion 41b. Therefore, in the refrigerant passage between throttle portion 41c and the downstream end of the needle valve 44, refrigerant flow acceleration due to expansion and evaporation of refrigerant is not caused, and large turbulence is not generated in speed boundary layers of refrigerant flowing on and around a surface of the needle valve 44.

Next, operational effects of the ejector 40 according to the first embodiment will be now described. As shown in FIGS. 3A, 3B, the sectional area of the refrigerant passage, defined by the nozzle 41 and the needle valve 44, reduces toward the throttle portion 41c. Therefore, a flow speed of refrigerant, flowing from the radiator 20 into the nozzle 41, increases toward the throttle portion 41c while a flow amount of the refrigerant becomes a flow amount determined by the open degree of the nozzle 41. On the other hand, the sectional area of the refrigerant passage is slightly increased from the throttle portion 41c to the downstream end of the needle valve 44. However, an increase rate of the sectional area is a little in the refrigerant passage from the throttle portion 41c to the downstream end of the needle valve 44, as compared with the expansion portion 41b. Therefore, in the refrigerant passage between throttle portion 41c and the downstream end of the needle valve 44, refrigerant flow acceleration due to expansion and evaporation of refrigerant is not caused, and large turbulence is not generated in speed boundary layers of refrigerant flowing on and around a surface of the needle valve 44.

Further, the sectional area of the refrigerant passage in the nozzle 41 reduces again from the top end of the needle valve 44 to the throat portion 41a. Therefore, between the top end of the needle valve 44 and the throat portion 41a, refrigerant flow is throttled and accelerated while a little turbulence, generated between throttle portion 41c and the top end of the needle valve 44, is rectified. Further, the rectified refrigerant passes through the throat portion 41a, and flows into the expansion portion 41b. Then, in the expansion portion 41b, the refrigerant is expanded, and is accelerated to a speed equal to or higher than the sound speed. At this time, since the refrigerant, passing through the throat portion 41a, has a little turbulence, eddy loss generated due to the turbulence can be restricted in the expansion portion 41b.

The refrigerant from the radiator 20 is decompressed in the ejector 41 at an upstream portion from the throat portion 41a to be gas-liquid two-phase refrigerant. Therefore, as shown in FIG. 4, refrigerant bubbles, generated upstream of the throat portion 41a, are more compressed as toward the throat portion 41a. Then, the number of the refrigerant bubbles is reduced, and boiling cores are generated at the throat portion 41a. When the refrigerant flows into the expansion portion 41b through the throat portion 41a, the boiling cores are again boiled, thereby facilitating refrigerant boiling in the expansion portion 41b, and accelerating the refrigerant to be equal to or higher than the sound speed. In the first embodiment, a flow amount of refrigerant is not adjusted by directly changing the cross-sectional area of the refrigerant passage in the throat portion 41a. Actually, refrigerant is decompressed to the gas-liquid two-phase refrigerant in the refrigerant passage upstream from the throat portion 41a, and refrigerant bubbles are generated in the gas-liquid refrigerant, so that a mass density of refrigerant is reduced. Accordingly, the cross-sectional area of the refrigerant passage in the nozzle 41 is relatively reduced. Thus, the flow amount of refrigerant can be adjusted, and the refrigerant passage can be prevented from being throttled more than a necessary degree. Accordingly, as shown at the right side (present-invention test result) in FIG. 5, ejector efficiency η can be prevented from being largely reduced.

In FIG. 5, "FIXED" represents a nozzle having a fixed shape most suitable for a flow amount of refrigerant
erant, and "CONTROL" represents a nozzle having a refrigerant passage throttled by the needle valve 44. In the present invention, since refrigerant can be accurately and sufficiently accelerated by the nozzle 41, the ejector efficiency η can be improved. As a result, the throttle degree of the nozzle 41 can be controlled in accordance with a refrigerant flow amount while the ejector efficiency η can be maintained at a high level.

0023 Further, a reference test result is shown at the left side in FIG. 5, and the ejector efficiency η of a refrigerant ejector is largely reduced as compared with the present embodiment. The reference test was performed by using a nozzle 410 shown FIGS. 6, 7. As shown FIG. 6, the inventors of the present invention studied a reference ejector 410 including a needle valve 440 for adjusting a throttle degree of the nozzle 410. The needle valve 440 has a cone-shaped top end, and is displaced in the nozzle 410 to adjust the throttle degree. In this case, refrigerant, flowing on and around the surface of the needle valve 440, flows along the surface of the cone-shaped top end of the needle valve 440. Therefore, the refrigerant streams along the surface of the cone-shaped top end collide with each other on a downstream side of the top end of the needle valve 440. Thus, an eddy loss due to refrigerant turbulence is generated in refrigerant streams and speed boundary layers of the refrigerant passage at a downstream side from the needle valve 440. Accordingly, a refrigerant flow speed is reduced even on a center axial line of the nozzle 410 in an expansion portion 410b of the nozzle 410. Originally, the refrigerant flow speed on the center axial line becomes highest. Therefore, refrigerant cannot be sufficiently accelerated by the nozzle 410, and the ejector efficiency η is reduced.

0024 On the other hand, as shown in FIG. 7, if the cross-sectional area of the refrigerant passage is simply controlled at the throat portion 410a so that the cross-sectional area of a space around the nozzle 410 is smallest at the throat portion 410a, refrigerant bubbles due to refrigerant boiling are readily generated downstream from the throat portion 410a. When refrigerant bubbles are generated in the refrigerant passage downstream from the throat portion 410a, the cross-sectional area of the refrigerant passage on the downstream side of the throat portion 410a is substantially reduced due to the refrigerant bubbles. Thus, the refrigerant passage is throttled more than a necessary level, and the ejector efficiency η is largely reduced as compared with the ejector having a fixed nozzle. Here, refrigerant can be decompressed to a pressure higher than saturation vapor pressure of refrigerant in the nozzle 410, in order to prevent the bubbles from being generated. However, an adiabatic heat fall (enthalpy change amount) due to the decompression around the saturation vapor pressure, is small. Therefore, it is difficult for the ejector 400 to recover a sufficient amount of energy. Furthermore, since the pumping function of the ejector 400 is small, a sufficient amount of refrigerant cannot be circulated to the evaporator 30.

0025 According to the first embodiment of the present invention, the refrigerant is decompressed to the gas-liquid two-phase refrigerant at an upstream side of the throat portion 41a. Therefore, it can prevent the refrigerant from being throttled more than a necessary degree while the ejector efficiency can be effectively improved.

(Second Embodiment)

0026 In the above-described first embodiment, as shown FIG. 3B, the inner wall surface of the nozzle 41 is formed into the two-step taper shape to have two taper angles α1, α2, so that the inner radial dimension d1 is reduced toward the throat portion 41a. However, in the second embodiment which is not within the scope of the claimed invention, as shown in FIG. 8, the inner wall surface has a taper angle gradually reduced toward the throat portion 41a, and is formed in a non-step taper shape so that the inner radial dimension d1 is reduced toward the throat portion 41a. Accordingly, the cross-sectional area of the refrigerant passage is smoothly and continuously changed in the nozzle 41, and turbulence can be further restricted from being generated in the refrigerant stream.

0027 In the second embodiment, the other parts are similar to those of the above-described first embodiment. Accordingly, similarly to the first embodiment, the refrigerant is decompressed to the gas-liquid two-phase state at an upstream side of the throat portion 41a.

(Third Embodiment)

0028 In the third embodiment which is not within the scope of the claimed invention, as shown in FIGS. 9A, 9B, the inner wall surface of the nozzle 41 is formed as a smoothly curved surface so that refrigerant is decompressed to the gas-liquid phase state at upstream from the throat portion 41a. In FIGS. 9A, 9B, 41d indicates an upstream area portion of the throat portion 41a, where the inner radial dimension d1 is reduced toward the throat portion 41a. Further, the nozzle 41, the mixing portion 42 and the diffuser 43 are set in the ejector 40 to have the sectional areas shown in FIG. 9B.

0029 In the third embodiment, the other parts are similar to those of the above-described first embodiment. Accordingly, similarly to the first embodiment, the refrigerant is decompressed to the gas-liquid two-phase state at an upstream side of the throat portion 41a.

0030 Although the present invention has been fully described in connection with the first embodiment 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.

0031 For example, in the above-described embodiments of the present invention, the top end shape of the needle valve 44 and the inner wall shape of the nozzle 41 are set so that the throttle portion 41c is formed upstream from the throat portion 41a, and refrigerant is de-
compressed to the gas-liquid refrigerant at the upstream of the throat portion 41a. However, without being limited to this manner, the top end shape of the needle valve 44 and the inner wall shape of the nozzle 41 may be determined only so that refrigerant is decompressed to the gas-liquid two-phase refrigerant at upstream from the throat portion 41a. In the above embodiments, the pressure of high-pressure refrigerant is detected as a physical value corresponding to refrigerant pressure in the refrigerant cycle, and the actuator 45 is controlled based on the detected refrigerant pressure. However, in the present invention, the actuator 45 may be controlled based on a physical value relative to the refrigerant pressure, such as a temperature of high-pressure refrigerant, a temperature of water for the water heater and an amount of refrigerant flowing into the nozzle 41.

[0032] In the above embodiments, the throttle degree of the nozzle 41 is controlled so that the high-pressure refrigerant is set at the predetermined pressure. However, for example, the throttle degree may be controlled so that a ratio of heating performance of the radiator 20 to motive power consumed by the compressor 10, that is, a performance coefficient of the ejector cycle, is set higher than the predetermined value. In the above-described embodiments, the present invention is typically applied to the water heater. However, without being limited to the water heater, the present invention can be applied to another ejector cycle such as a refrigerator, a freezer and a water heater, the present invention can be applied to any other ejector cycle.

[0033] Such changes and modifications are to be understood as being within the scope of the present invention as defined by the appended claims.

Claims

1. An ejector decompression device for a refrigerant cycle that includes a radiator (20) for radiating heat of refrigerant compressed by a compressor (10), and an evaporator (30) for evaporating refrigerant after being decompressed, the ejector decompression device comprising:

- a nozzle (41) having an inner wall defining a refrigerant passage, for decompressing and expanding refrigerant flowing from the radiator by converting pressure energy of refrigerant to speed energy of the refrigerant, a needle valve (44) disposed to be displaced in the refrigerant passage of the nozzle (41) in an axial direction of the nozzle, for adjusting an opening degree of the refrigerant passage of the nozzle, characterized in that, the nozzle includes a throat portion (41a) having a cross-sectional area that is smallest in the refrigerant passage of the nozzle, and an expansion portion (41b) in which the cross-sectional area is increased toward downstream in a refrigerant flow;
- a pressure-increasing portion (42, 43) which increases the pressure of refrigerant by converting the speed energy of refrigerant to the pressure energy of refrigerant while mixing refrigerant injected from the nozzle and refrigerant sucked from the evaporator; and
- the needle valve (44) and the inner wall of the nozzle (41) are provided to have predetermined shapes so that refrigerant flowing into the nozzle is decompressed to a gas-liquid two-phase state at upstream from the throat portion (41a) in the refrigerant flow, and
- the needle valve (44) has a downstream portion that is tapered toward a downstream end of the nozzle so that a cross-sectional area of the downstream portion of the needle valve is reduced toward the downstream end; the inner wall of the nozzle is formed into an approximate cone shape having at least two different taper angles (α1, α2), upstream from the throat portion (41a); and
- the inner wall has a radial dimension that is reduced toward the throat portion (41a).

2. The ejector decompression device according to claim 1, wherein the needle valve (44) has a downstream end that is disposed to be displaced in the refrigerant passage of the nozzle, in an area upstream from the throat portion (41a).

3. The ejector decompression device according to any one of claims 1 and 2, wherein:

- the needle valve (44) is disposed in the refrigerant passage of the nozzle (41) to define a throat portion (41c) having a cross-sectional area that is smallest in a space between the needle valve and the inner wall of the nozzle; and
- the needle valve and the inner wall of the nozzle are provided such that the throat portion (41c) is positioned upstream from the throat portion (41a) in the refrigerant flow.

4. The ejector decompression device according to any one of claims 1-3, wherein:

- the needle valve (44) has a downstream portion that is tapered toward a downstream end of the needle valve so that a cross-sectional area of the downstream portion of the needle valve is reduced toward the downstream end; and
- the inner wall of the nozzle (41) has a radial dimension that is reduced from an upstream end of the nozzle toward the throat portion (41a) and
is increased from the throat portion toward a downstream end of the nozzle.

5. The ejector decompression device according to any one of claims 1-4, further comprising an electric actuator (45) for displacing the needle valve (44) in the refrigerant passage of the nozzle.

6. The ejector decompression device according to claim 5, further comprising:

a detecting unit for detecting a physical value relative to a refrigerant pressure in the refrigerant cycle; and

a controller for controlling operation of the electric actuator (45) based on the physical value detected by the detecting unit.

7. The ejector decompression device according to claim 5, wherein the electric actuator (45) is a stepping motor.

8. The ejector decompression device according to claim 5, wherein the electric actuator (45) is a linear solenoid motor.

9. The ejector decompression device according to any one of claims 1-8, wherein a pressure of refrigerant in the radiator (20) becomes equal to or higher than the critical pressure of the refrigerant.

10. The ejector decompression device according to any one of claims 1-9, wherein the refrigerant is carbon dioxide.

Patentansprüche

1. Ejektorpumpen-Dekompressionsvorrichtung für einen Kühlkreis, der einen Kühlern (20) zum Kühlen eines durch einen Kompressor (10) komprimierten Kältemittels und einen Verdampfapparat (30) zum Verdampfen des Kältemittels nach der Dekompression enthält, wobei die Ejektorpumpen-Dekompressionsvorrichtung aufweist:

   eine Düse (41) mit einer eingespritzte Kältemittel und das vom Verdampfapparat angesaugte Kältemittel vermischten; und ein Nadeldieter (44), das in dem Kältemittel durchgang der Düse (41) in einer axialen Richtung der Düse verschiebbar angeordnet ist, um einen Öffnungsgrad des Kältemittel durchgangs der Düse einzu stellen.

   dass die Düse einen Verengungsabschnitt (41 a) mit einer Querschnittsfläche, die im Kältemittel durchgang der Düse am kleinsten ist, und einem Erweiterungsabschnitt (41 b), in dem die Querschnittsfläche in stromabwärtiger Richtung eines Kältemittelstroms größer wird, enthält;

   das Nadeldieter (44) und die Innenwand der Düse (41) mit vorbestimmten Formen derart vorge sehen sind, dass das in die Düse strömende Kältemittel stromabwärts des Verengungsabschnitts (41 a) im Kältemittelstrom in einen Gas/Flüssigkeit-Zweiphasen zustand dekomprimiert wird; und

   das Nadeldieter (44) einen stromabwärtsen Abschnitt hat, der zu einem stromabwärtsen Ende des Nadeldiventils konisch verjüngt ist, sodass eine Querschnittsfläche des stromabwärtsen Bereichs des Nadeldiventils zum stromabwärtigen Ende hin kleiner wird, die Innenwand der Düse in eine unge fähr Konusform mit wenigstens zwei konischen Winkeln a1, a2 stromauf de Verengungsabschnitts (41 a) ausgebildet ist und die Innenwand ein Radi almaß besitzt, das zum Verengungsabschnitt (41 a) hin kleiner wird.

2. Ejektorpumpen-Dekompressionsvorrichtung nach Anspruch 1, bei welcher das Nadeldieter (44) ein stromabwärtiges Ende aufweist, das in dem Kältemittel durchgang der Düse in einem Bereich stromauf de Verengungsabschnitts (41 a) verschiebbar angeordnet ist.

3. Ejektorpumpen-Dekompressionsvorrichtung nach einem der Ansprüche 1 und 2, bei welcher das Nadeldieter (44) im Kältemittel durchgang der Düse (41) so angeordnet ist, dass es einen Druckschlag (41 c) mit einer Querschnittsfläche, die in einem Raum zwischen dem Nadeldieter und der In nenwand der Düse am kleinsten ist, definiert; und das Nadeldieter und die Innenwand der Düse so vorge sehen sind, dass der Druckschlagabschnitt (41 c) im Kältemittelstrom stromauf de Verengungsabschnitts (41 a) positioniert ist.

4. Ejektorpumpen-Dekompressionsvorrichtung nach einer der Ansprüche 1 bis 3, bei welcher das Nadeldieter (44) einen stromabwärtigen Abschnitt besitzt, der zu einem stromabwärtigen Ende des Nadeldiventils konisch verjüngt ist, sodass ei-
Dispositif éjecteur à décompression pour un cycle réfrigérant

Revendications

1. Dispositif éjecteur à décompression pour un cycle réfrigérant qui inclut une soupape (44) pour éjecter un réfrigérant comprimé dans un compresseur (10), et un évaporateur (30) pour faire éjecter un réfrigérant après avoir été décompressé, le dispositif éjecteur à décompression comprenant:

une buse (41) ayant une paroi intérieure définissant un passage de réfrigérant, pour décomposer et dilater un réfrigérant s’écoutant depuis le radiateur en convertissant une énergie de pression d’un réfrigérant en une énergie de vitesse du réfrigérant, une soupape à pointeau (44) disposée pour être déplacée dans le passage de réfrigérant de la buse (41), dans une direction axiale de la buse, pour ajuster un degré d’ouverture du passage de réfrigérant de la buse, caractérisé en ce que, la buse inculc une portion de gorgue (41 a) ayant une superficie transversale qui est la plus petite dans le passage de réfrigérant de la buse, et une portion d’expansion (41 b) dans laquelle la superficie transversale est augmentée vers l’aval dans un écoulement de réfrigérant;

une portion d’augmentation de pression (42, 43) qui augmente la pression de réfrigérant en convertissant l’énergie de vitesse de réfrigérant en l’énergie de pression de réfrigérant tout en délocalant du réfrigérant injecté depuis la buse et du réfrigérant aspiré depuis l’évaporateur; et

la soupape à pointeau (44) et la paroi intérieure de la buse (41) sont fournies de sorte à avoir des formes prédéterminées de manière à ce qu’un réfrigérant s’écoulant dans la buse soit décompressé à un état à deux phases gaz-liquide en amont depuis la portion de gorgue (41 a), dans l’écoulement de réfrigérant, et la soupape à pointeau (44) a une portion aval qui est conique vers une extrémité aval de la soupape à pointeau de manière à ce qu’une superficie transversale de la portion aval de la soupape à pointeau soit réduite vers l’extrémité aval; et

la paroi intérieure de la buse est formée en une forme de cône approximative ayant au moins deux angles de conicité différents (α1, α2), en amont depuis la portion de gorgue (41 a); et

la paroi intérieure a une dimension radiale qui est réduite vers la portion de gorgue (41 a).

2. Dispositif éjecteur à décompression selon la revendication 1, dans lequel la soupape à pointeau (44), a une extrémité aval qui est disposé pour être déplacée dans le passage de réfrigérant de la buse, dans une zone en amont depuis la portion de gorgue (41 a).

3. Dispositif éjecteur à décompression selon l’une quelconque des revendications 1 et 2, dans lequel:

la soupape à pointeau (44) est disposée dans le passage de réfrigérant de la buse (41) pour définir une portion d’étranglement (41 c) ayant une superficie transversale qui est la plus petite dans un espace entre la soupape à pointeau et la paroi intérieure de la buse; et

la soupape à pointeau et la paroi intérieure de la buse sont fournies de sorte que la portion d’étranglement (41 c) soit positionnée en amont depuis la portion de gorgue (41 a) dans l’écoulement de réfrigérant.

4. Dispositif éjecteur à décompression selon l’une quelconque des revendications 1-3, dans lequel:
la soupape à pointeau \((44)\) a une portion aval qui est conique vers une extrémité aval de la soupape à pointeau de manière à ce qu’une superficie transversale de la portion aval de la soupape à pointeau soit réduite vers l’extrémité aval; et

la paroi intérieure de la buse \((41)\) a une dimension radiale qui est réduite depuis une extrémité amont de la buse vers la portion de gorge \((41\ a)\) et est augmentée depuis la portion de gorge vers une extrémité aval de la buse.

5. Dispositif éjecteur à décompression selon l’une quelconque des revendications 1-4, comprenant en plus un actionneur électrique \((45)\) pour déplacer la soupape à pointeau \((44)\) dans le passage de réfrigérant de la buse.

6. Dispositif éjecteur à décompression selon la revendication 5, comprenant en plus:

   une unité de détection pour détecter une valeur physique se rapportant à une pression de réfrigérant dans le cycle frigorifique; et 

   une unité de commande pour commander un fonctionnement de l’actionneur électrique \((45)\) sur la base de la valeur physique détectée par l’unité de détection.

7. Dispositif éjecteur à décompression selon la revendication 5, dans lequel l’actionneur électrique \((45)\) est un moteur pas à pas.

8. Dispositif éjecteur à décompression selon la revendication 5, dans lequel l’actionneur électrique \((45)\) est un moteur à solénoïde linéaire.

9. Dispositif éjecteur à décompression selon l’une quelconque des revendications 1-8, dans lequel une pression de réfrigérant dans le radiateur \((20)\) devient égale à ou supérieure à la pression critique du réfrigérant.

10. Dispositif éjecteur à décompression selon l’une quelconque des revendications 1-9, dans lequel le réfrigérant est du dioxyde de carbone.
FIG. 4

FIG. 5

100
90
80
70
60
50

REFERENCE

PRESENT INVENTION

7% REDUCED IN VARIABLE CONTROL

FIXED VARIABLE CONTROL (IN MINIMUM FLOW)

90% 92% 80%

60%
REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

• FR 1575202 A [0003]