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(54) HEAT EXCHANGER

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(73) Proprietor: **Daikin Industries, Ltd.**
Osaka-shi, Osaka 530-8323 (JP)

(72) Inventors:
- **JINDOU, Masanori**
  Osaka-shi, Osaka 530-8323 (JP)
- **ORITANI, Yoshio**
  Osaka-shi, Osaka 530-8323 (JP)
- **KIKUCHI, Yoshimasa**
  Osaka-shi, Osaka 530-8323 (JP)
- **SASAI, Yasuhiro**
  Osaka-shi, Osaka 530-8323 (JP)
- **KUBO, Hiroharu**
  Osaka-shi, Osaka 530-8323 (JP)
- **INOUE, Satoshi**
  Osaka-shi, Osaka 530-8323 (JP)
- **KIM, Hyunyoung**
  Osaka-shi, Osaka 530-8323 (JP)
- **FUJINO, Hirokazu**
  Osaka-shi, Osaka 530-8323 (JP)

(74) Representative: **Hoffmann Eitle**
Patent- und Rechtsanwälte PartmbB
Arabellastraße 30
81925 München (DE)

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Description

TECHNICAL FIELD

[0001] The present invention relates to a refrigerant circuit configured to perform a refrigerating cycle and comprising a heat exchanger including a plurality of flat tubes and header-collecting pipes connected to the plurality of flat tubes.

BACKGROUND ART

[0002] Document FR 2 664 371 A1 discloses a refrigerant circuit having the features in the preamble of claim 1. Heat exchangers including a plurality of flat tubes and header-collecting pipes connected to the plurality of flat tubes have conventionally been known. Japanese Unexamined Patent Publication No. 2006-105545 and Japanese Unexamined Patent Publication No. 2005-003223 each disclose a heat exchanger of this type. Specifically, the heat exchanger of each of the patent documents includes first and second header-collecting pipes which are installed in an upright position on the right and left sides of the heat exchanger, respectively, and a plurality of flat tubes which extend from the first header-collecting pipe to the second header-collecting pipe. The heat exchanger of each of the patent documents causes a refrigerant flowing inside the flat tubes to exchange heat with air flowing outside the flat tubes.

[0003] The heat exchanger described in JP 2006-105545 functions as a condenser. In this heat exchanger, the gaseous refrigerant having flowed into an upper end portion of the first header-collecting pipe is distributed to all of the flat tubes. The refrigerant flowing through the flat tubes dissipates heat into air and condensates, and then, flows into the second header-collecting pipe. Thereafter, the refrigerant flows out from a lower end portion of the second header-collecting pipe to the outside of the heat exchanger.

[0004] The heat exchanger described in JP 2005-003223 also functions as a condenser. In this heat exchanger, the refrigerant having flowed into an upper end portion of the first header-collecting pipe makes one and a half round-trips between the two header-collecting pipes, and then, flows out from a lower end portion of the second header-collecting pipe. Further examples can be seen in document US 2010/031698 A1.

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0005] Meanwhile, a heat exchanger as described in the patent documents can be used as an evaporator. When the heat exchanger functions as an evaporator, the refrigerant being in a gas-liquid two-phase state and having been supplied to the heat exchanger flows into one of the header-collecting pipes, and then, diverges and separately flows into the plurality of flat tubes. In this case, it is desirable to make flow rates at which the refrigerant flows into the flat tubes as uniform as possible, for the following reason. If the flow rates at which the refrigerant flows into the flat tubes are nonuniform, at a midpoint in the flat tubes into which the refrigerant has flowed at low flow rates, the refrigerant enters a single-phase gas state, which disadvantageously causes the heat exchanger to provide insufficient performance.

[0006] Conventionally, a prerequisite for uniformly distributing a refrigerant being in a gas-liquid two-phase state from a header-collecting pipe to a plurality of flat tubes has not been found out to a sufficient extent. Consequently, it has been difficult to cause a heat exchanger functioning as an evaporator to provide sufficient performance.

[0007] It is therefore an object of the present invention to cause a heat exchanger including a plurality of flat tubes and header-collecting pipes to provide the highest performance when the heat exchanger functions as an evaporator.

SOLUTION TO THE PROBLEM

[0008] A first aspect of the present invention relates to a refrigerant circuit configured to perform a refrigerating cycle and comprising a heat exchanger, the heat exchanger comprising: a plurality of flat tubes (31); a first header-collecting pipe (60) having inserted therein an end portion of each of the flat tubes (31); a second header-collecting pipe (70) having inserted therein the other end portion of each of the flat tubes (31); and a plurality of fins (36) joined to the flat tubes (31), wherein the second header-collecting pipe (70) forms flow spaces (71a-71c) which communicate with the plurality of flat tubes (31) and in which a refrigerant being in a gas-liquid two-phase state flows upwardly when the heat exchanger functions as an evaporator, an effective cross-sectional area of the flow spaces (71a-71c) is an area obtained by subtracting a projected area which corresponds to a portion of each flat tube (31) located in a corresponding one of the flow spaces (71a-71c) and which is projected onto a plane perpendicular to an axial direction of the second header-collecting pipe (70), from an area of a cross section of the corresponding one of the flow spaces (71a-71c) which is perpendicular to the axial direction of the second header-collecting pipe (70), and the effective cross-sectional area of the flow spaces (71a-71c) is set based on a mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator.

[0009] In the heat exchanger (23), the second header-collecting pipe (70) forms the flow spaces (71a-71c). Here, a flow of the refrigerant in the heat exchanger (23) functioning as the evaporator is described. The refrigerant being in a gas-liquid two-phase state flows into the flow spaces (71a-71c) of the second header-collecting pipe (70). The refrigerant having flowed into the flow spaces (71a-71c) upwardly flows in the flow spaces (71a-71c). While flowing in the flow spaces (71a-71c), the refrigerant diverges and separately enters the plurality of flat tubes (31) communicating with the flow spaces (71a-71c). The refrigerant having flowed from the flow spaces (71a-71c) into the flat tubes (31) passes through the flat tubes (31) and enters the first header-collecting pipe (60).

[0010] According to the first aspect, the effective cross-sectional area A of the flow spaces (71a-71c) of the second header-collecting pipe (70) is an area obtained by subtracting a projected area A_1 which corresponds to a portion of each flat tube (31) located in a corresponding one of the flow spaces (71a-71c) and which is projected onto a plane perpendicular to an axial direction of the second header-collecting pipe (70) from the area A_0 of a cross section of the flow spaces (71a-71c) which is perpendicular to the axial direction of the second header-collecting pipe (70) ($A = A_0 - A_1$). In the heat exchanger (23) of the present invention, the effective cross-sectional area A of the flow spaces (71a-71c) is set based on the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger (23) functions as the evaporator.

[0011] According to the first aspect, a value included in a variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator is determined as a reference mass flow rate M_R [kg/h], and the effective cross-sectional area A [mm^2] of the flow spaces (71a-71c) is equal to or greater than ($1.91M_R$ - 22.7) and equal to or smaller than ($1.96M_R$ + 30.8).

[0012] According to the first aspect, a value included in a variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator is determined as a reference mass flow rate M_R [kg/h], and the effective cross-sectional area A [mm^2] of the flow spaces (71a-71c) is equal to or greater than ($1.96M_R$ - 25.0) and equal to or smaller than ($1.96M_R$ + 30.0).

[0013] According to the first aspect, a mass flow rate at which the refrigerant flows into the heat exchanger (23) functioning as the evaporator varies according to operational states of the refrigerant circuit (20) in which the heat exchanger (23) is provided. The mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger (23) functions as the evaporator varies within a predetermined variation range. A value included in the variation range of the mass flow rate of the refrigerant is determined as the reference mass flow rate M_R. The reference mass flow rate M_R may be the upper limit value, the lower limit value, or a median value of the variation range, for example. In the first aspect, the reference mass flow rate M_R is expressed in units of "kg/h" and the effective cross-sectional area A is expressed in units of "mm^2."

[0014] According to the first aspect, the effective cross-sectional area A of the flow spaces (71a-71c) of the second header-collecting pipe (70) is a value satisfying the following formula:

$$1.91M_R - 22.7 \leq A \leq 1.96M_R + 30.8$$

[0015] On the other hand, according to the first aspect, the effective cross-sectional area A of the flow spaces (71a-71c) of the second header-collecting pipe (70) is a value satisfying the following formula:

$$1.96M_R - 25.0 \leq A \leq 1.96M_R + 30.0$$

[0016] A second aspect of the present invention relates to the refrigerant circuit of the first aspect, wherein the reference mass flow rate M_R [kg/h] is the upper limit value of the variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator.

[0017] According to the second aspect, the upper limit value of the variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger (23) functions as the evaporator is determined as the reference mass flow rate M_R.

[0018] A third aspect of the present invention relates to the refrigerant circuit of any one of the first or second aspect, wherein the first header-collecting pipe (60) and the second header-collecting pipe (70) are in an upright position, and the heat exchanger is configured such that the refrigerant flows into a lower end portion of each of the flow spaces (71a-71c) when the heat exchanger functions as the evaporator.

[0019] According to the third aspect, the first header-collecting pipe (60) and the second header-collecting pipe (70) are in an upright position. When the heat exchanger (23) according to this aspect functions as the evaporator, the refrigerant having flowed into the lower end portion of each of the flow spaces (71a-71c) of the second header-collecting pipe (70) upwardly flows in the flow spaces (71a-71c).

ADVANTAGES OF THE INVENTION

[0020] In respect of the heat exchanger (23) functioning as the evaporator, in order to distribute uniformly the refrigerant being in a gas-liquid two-phase state and flowing upwardly in the flow spaces (71a-71c) of the second header-collecting pipe (70) to the plurality of flat tubes (31), it is required to set the flow velocity of the refrigerant flowing in the flow spaces (71a-71c) to an appropriate value. The inventors of the present invention have made this finding through analyses of results of various experiments.

[0021] Specifically, when the flow velocity at which the refrigerant upwardly flows in each of the flow spaces (71a-71c) is excessive low, since almost none of the liquid refrigerant can reach upper ones of the flat tubes (31), the refrigerant flowing into the upper ones of the flat tubes (31) increases in dryness fraction. Consequently, the amount of heat which the refrigerant flowing through the upper ones of the flat tubes (31) communicating with the flow spaces (71a-71c) absorbs is reduced, thereby causing the heat exchanger (23) to provide insufficient performance. On the other hand, when the flow velocity at which the refrigerant upwardly flows in each of the flow spaces (71a-71c) is excessive high, since a large proportion of the liquid refrigerant having been raised with great force flows into upper ones of the flat tubes (31) and a large proportion of the gaseous refrigerant flows into lower ones of the flat tubes (31), the refrigerant flowing into the lower ones of the flat tubes (31) increases in degree of dryness. Consequently, the amount of heat which the refrigerant flowing through the lower ones of the flat tubes (31) communicating with the flow spaces (71a-71c) absorbs is reduced, thereby causing the heat exchanger (23) to provide insufficient performance. Thus, the performance provided by the heat exchanger (23) becomes insufficient when the flow velocity at which the refrigerant upwardly flows in the flow spaces (71a-71c) is excessively low as well as when the flow velocity is excessively high.

[0022] To address this problem, the heat exchanger (23) of the present invention is configured such that the effective cross-sectional area A of the flow spaces (71a-71c) of the second header-collecting pipe (70) is set based on the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger (23) functions as the evaporator. The effective cross-sectional area A of the flow spaces (71a-71c) and the mass flow rate of the refrigerant flowing into the flow spaces (71a-71c) are physical quantities which have influence on the flow velocity at which the refrigerant flows in the flow spaces (71a-71c). Accordingly, setting of the effective cross-sectional area A of the flow spaces (71a-71c) on the basis of the mass flow rate of the refrigerant flowing into the flow spaces (71a-71c) makes it possible to set the flow velocity of the refrigerant flowing in the flow spaces (71a-71c) to an appropriate value at which the refrigerant is uniformly distributed from the flow spaces (71a-71c) to the plurality of flat tubes (31). Thus, according to the present invention, it is possible to cause the heat exchanger (23) to provide sufficient performance.

[0023] Here, the assumption is made that the refrigerant in the heat exchanger (23) functioning as the evaporator absorbs an insufficient amount of heat. In such a case, if the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) is smaller than the upper limit value of the variation range, it is possible to increase the amount of heat which the refrigerant absorbs by increasing the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c). However, when the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) has already reached the upper limit value of the variation range, no further increase in the mass flow rate of the refrigerant is possible.

[0024] To address this, according to the second aspect of the present invention, the reference mass flow rate M_R, based on which the effective cross-sectional area A of the flow spaces (71a-71c) is set, is equal to the upper limit value of the variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger (23) functions as the evaporator. Therefore, according to this aspect, it is possible to cause the heat exchanger (23) to provide the highest performance when no increase in the mass flow rate of the refrigerant flowing into the flow spaces (71a~71c) is possible.

BRIEF DESCRIPTION OF THE DRAWINGS

[0025]

[FIG. 1] FIG. 1 is a refrigerant circuit diagram schematically illustrating a configuration of an air conditioner including an outdoor heat exchanger according to an embodiment of the present invention.

[FIG. 2] FIG. 2 is a front view schematically illustrating a configuration of the outdoor heat exchanger of the embodiment.

[FIG. 3] FIG. 3 is a cross-sectional view illustrating a portion of the outdoor heat exchanger of the embodiment, viewed from front.

[FIG. 4] FIG. 4 is an enlarged cross-sectional view illustrating a portion of the cross section taken along the line A-A in FIG. 3.

[FIG. 5] FIGS. 5A-5D are enlarged cross-sectional views of a main portion of the cross section taken along the line B-B in FIG. 3. Specifically, FIG. 5A illustrates the dimensions of parts. FIG. 5B illustrates a cross-sectional area A_0 of a subspace in a second header-collecting pipe. FIG. 5C illustrates a projected area A_1 of a flat tube. FIG. 5D illustrates an effective cross-sectional area A of the subspace in the second header-collecting pipe.

[FIG. 6] FIG. 6 is a graph illustrating relations between a flow velocity V of a refrigerant in the subspace of the second header-collecting pipe and a performance rate of the outdoor heat exchanger.

[FIG. 7] FIG. 7 is a graph illustrating a range of the effective cross-sectional area A of the outdoor heat exchanger of the embodiment.

[FIG. 8] FIG. 8 is a graph illustrating a range of the effective cross-sectional area A of the outdoor heat exchanger of Variation 2 of the embodiment.

DESCRIPTION OF EMBODIMENTS

[0026] An embodiment of the present invention will be descried below in detail with reference to the drawings. The following embodiment and variations are merely preferred examples in nature, and are not intended to limit the scope, applications, and use of the present disclosure.

[0027] A heat exchanger of this embodiment is an outdoor heat exchanger (23) provided in an air conditioner (10). The air conditioner (10) is described first, and thereafter, a detailed description of the outdoor heat exchanger (23) will be given.

- Air Conditioner -

[0028] First, the air conditioner (10) is described with reference to FIG. 1.

<Configuration of Air Conditioner>

[0029] The air conditioner (10) includes an outdoor unit (11) and an indoor unit (12). The outdoor unit (11) and the indoor unit (12) are connected to each other via a liquid communication pipe (13) and a gas communication pipe (14). In the air conditioner (10), the outdoor unit (11), the indoor unit (12), the liquid communication pipe (13), and the gas communication pipe (14) form a refrigerant circuit (20).

[0030] The refrigerant circuit (20) includes a compressor (21), a four-way switching valve (22), the outdoor heat exchanger (23), an expansion valve (24), and an indoor heat exchanger (25). The compressor (21), the four-way switching valve (22), the outdoor heat exchanger (23), and the expansion valve (24) are housed in the outdoor unit (11). The outdoor unit (11) is provided with an outdoor fan (15) configured to supply outdoor air to the outdoor heat exchanger (23). On the other hand, the indoor heat exchanger (25) is housed in the indoor unit (12). The indoor unit (12) is provided with an indoor fan configured to supply indoor air to the indoor heat exchanger (25).

[0031] The refrigerant circuit (20) is a closed circuit filled with a refrigerant. In the refrigerant circuit (20), a discharge side and a suction side of the compressor (21) are in connection to a first port and a second port of the four-way switching valve (22), respectively. Further, in the refrigerant circuit (20), a third port of the four-way switching valve (22), the outdoor heat exchanger (23), the expansion valve (24), the indoor heat exchanger (25), and a fourth port of the four-way switching valve (22) are sequentially arranged.

[0032] The compressor (21) is a scroll-type or rotary-type hermetic compressor. Rotation speed of the compressor (21) is variable. Varying the rotation speed of the compressor (21) causes operation capacity of the compressor (21) to vary. The four-way switching valve (22) is switchable between a first state and a second state. In the first state (indicated by the broken lines in FIG. 1), the first port communicates with the third port and the second port communicates with the fourth port. In the second state (indicated by the solid lines in FIG. 1), the first port communicates with the fourth port and the second port communicates with the third port. The expansion valve (24) is a so-called electronic expansion valve.

[0033] The outdoor heat exchanger (23) causes outdoor air to exchange heat with the refrigerant. The outdoor heat exchanger (23) will be detailed later. On the other hand, the indoor heat exchanger (25) causes indoor air to exchange heat with the refrigerant. The indoor heat exchanger (25) is a so-called cross-fin type fin-and-tube heat exchanger including circular heat transfer tubes.

<Operation of Air Conditioner>

[0034] The air conditioner (10) selectively performs cooling operation and heating operation.

[0035] During the cooling operation, the refrigerant circuit (20) performs a refrigerating cycle with the four-way switching valve (22) maintained in the first state. In this state, the refrigerant circulates by passing through the outdoor heat exchanger (23), the expansion valve (24), and the indoor heat exchanger (25) in this order, and the outdoor heat exchanger (23) functions as a condenser whereas the indoor heat exchanger (25) functions as an evaporator. In the outdoor heat exchanger (23), the gaseous refrigerant having flowed from the compressor (21) dissipates heat into outdoor air to become condensed, and the condensed refrigerant flows out of the heat exchanger (23) toward the expansion valve (24).

[0036] During the heating operation, the refrigerant circuit (20) performs a refrigerating cycle with the four-way switching valve (22) maintained in the second state. In this state, the refrigerant circulates by passing through the indoor heat exchanger (25), the expansion valve (24), and the outdoor heat exchanger (23) in this order, and the indoor heat exchanger (25) functions as a condenser whereas the outdoor heat exchanger (23) functions as an evaporator. The refrigerant having expanded upon passing through the expansion valve (24) and being in a gas-liquid two-phase state flows into the outdoor heat exchanger (23). In the outdoor heat exchanger (23), the refrigerant absorbs heat from outdoor air and evaporates, and then, flows out of the outdoor heat exchanger (23) toward the compressor (21).

- Outdoor Heat Exchanger -

[0037] The outdoor heat exchanger (23) is now described with reference to FIGS. 2-7 as appropriate. Note that the number of flat tubes (31, 32), the number of principal heat exchange sections (51a-51c), and the number of auxiliary heat exchange sections (52a-52c) will be described below as a mere example.

<Configuration of Outdoor Heat Exchanger>

[0038] As illustrated in FIGS. 2 and 3, the outdoor heat exchanger (23) includes a first header-collecting pipe (60), a second header-collecting pipe (70), and a large number of the flat tubes (31, 32), and a large number of fins (36). The first header-collecting pipe (60), the second header-collecting pipe (70), the flat tubes (31, 32), and the fins (36) are each an aluminum alloy member and brazed to one another.

[0039] As will be detailed later, the outdoor heat exchanger (23) is divided into a principal heat exchange region (51) and an auxiliary heat exchange region (52). In the outdoor heat exchanger (23), the flat tubes (32) constitute the auxiliary heat exchange region (52) and the flat tubes (31) constitute the principal heat exchange region (51).

[0040] Each of the first header-collecting pipe (60) and the second header-collecting pipe (70) has a long narrow cylindrical shape with both ends closed. In FIGS. 2 and 3, the first header-collecting pipe (60) stands in an upright position and forms the left edge of the outdoor heat exchanger (23), and the second header-collecting pipe (70) stands in an upright position and forms the right edge of the outdoor heat exchanger (23). Thus, the first and second header-collecting pipes (60, 70) are disposed such that the axial direction of the header-collecting pipes (60, 70) corresponds to the vertical direction of the outdoor heat exchanger (23).

[0041] As illustrated in FIG. 4, each of the flat tubes (31, 32) is a heat exchanger tube having a flat oval cross-section. As illustrated in FIG. 3, in the outdoor heat exchanger (23), the direction in which the plurality of flat tubes (31, 32) extend corresponds to the lateral direction, and the flat tubes (31, 32) are arranged such that flat faces of the adjacent ones of the flat tubes (31, 32) face each other. The plurality of flat tubes (31, 32) are arranged one above the other at regular intervals and substantially in parallel with one another. Each of the flat tubes (31, 32) has an end portion inserted in the first header-collecting pipe (60) and the other end portion inserted in the second header-collecting pipe (70). The axial direction of each of the flat tubes (31, 32) is substantially perpendicular to the axial directions of the header-collecting pipes (60, 70). The flat faces (in this embodiment, the upper and lower faces) of each of the flat tubes (31, 32) are substantially perpendicular to the axial direction of the header-collecting pipes (60, 70).

[0042] As illustrated in FIG. 4, a plurality of fluid passages (34) extend in each of the flat tubes (31, 32). The fluid passages (34) extend in the direction in which the flat tubes (31, 32) extend. In each of the flat tubes (31, 32), the plurality of fluid passages (34) are aligned in the width direction (i.e., in the direction perpendicular to the longitudinal direction) of the flat tubes (31, 32). The plurality of fluid passages (34) extending in the flat tubes (31, 32) each has an end communicating with the inner space of the first header-collecting pipe (60) and the other end communicating with the inner space of the second header-collecting pipe (70). The refrigerant supplied to the outdoor heat exchanger (23) exchanges heat with air while flowing through the fluid passages (34) extending in the flat tubes (31, 32).

[0043] As illustrated in FIG. 4, each fin (36) is a vertically oriented plate fin made by subjecting a metal plate to press work. Each fin (36) has multiple long narrow notches (45) extending from the front edge (i.e., the edge located upstream of an air flow) of the fin (36) in the width direction of the fin (36). In each fin (36), the multiple notches (45) are arranged at regular intervals in the longitudinal direction (the vertical direction). A portion of each notch (45) located downstream of the air flow serves as a tube insertion section (46). Each tube insertion section (46) has a vertical width substantially equal to the thickness of the flat tubes (31, 32) and a length substantially equal to the width of flat tubes (31, 32). The flat tubes (31, 32) are inserted into the tube insertion sections (46) of the fins (36), and brazed to circumferential portions of the tube insertion sections (46). Further, louvers (40) for promoting heat transfer are formed in each fin (36). The plurality of fins (36) are arranged across the direction in which the flat tubes (31, 32) extend, and thereby divide spaces sandwiched between adjacent ones of the flat tubes (31, 32) into a plurality of air flow paths (38).

[0044] As illustrated in FIGS. 2 and 3, the outdoor heat exchanger (23) is divided into two regions located one above the other, i.e., the heat exchange regions (51, 52). In the outdoor heat exchanger (23), the upper heat exchange region serves as the principal heat exchange region (51) and the lower heat exchange region serves as the auxiliary heat exchange region (52).

[0045] The heat exchange regions (51, 52) are each divided into three heat exchange sections (51a-51c, 52a-52c) located one above the other. That is, in the outdoor heat exchanger (23), the principal heat exchange region (51) and the auxiliary heat exchange region (52) are each divided into the same number of heat exchange sections (51a-51c, 52a-52c). The heat exchange regions (51, 52) may be divided into two heat exchange sections or four or more heat exchange sections.

[0046] Specifically, the principal heat exchange region (51) includes, in the order from bottom to top, the first principal heat exchange section (51a), the second principal heat exchange section (51b), and the third principal heat exchange section (51c). The auxiliary heat exchange region (52) includes, in the order from bottom to top, the first auxiliary heat exchange section (52a), the second auxiliary heat exchange section (52b), and the third auxiliary heat exchange section (52c). Each of the principal heat exchange sections (51a-51c) and the auxiliary heat exchange sections (52a-52c) includes two or more of the flat tubes (31 or 32). As illustrated in FIG. 3, the number of the flat tubes (31) included in each of the principal heat exchange sections (51a-51c) is greater than the number of the flat tubes (32) included in each of the auxiliary heat exchange sections (52a-52c). Accordingly, the number of the flat tubes (31) included in the principal heat exchange region (51) is greater than the number of the flat tubes (32) included in the auxiliary heat exchange region (52).

[0047] As illustrated in FIG. 3, the inner space of the first header-collecting pipe (60) is partitioned by a partition plate (39a) into portions located one above the other. Thus, the first header-collecting pipe (60) includes the upper space (61) located above the partition plate (39a) and the lower space (62) located below the partition plate (39a).

[0048] The upper space (61) serves as a principal communicating space corresponding to the principal heat exchange region (51). The upper space (61) is a continuous space communicating with all of the flat tubes (31) included in the principal heat exchange region (51). That is, the upper space (61) communicates with the flat tubes (31) of the principal heat exchange sections (51a-51c).

[0049] The lower space (62) serves as an auxiliary communicating space corresponding to the auxiliary heat exchange region (52). The lower space (62) is partitioned by two partition plates (39b) into portions located one above the other. Specifically, the lower space (62) is partitioned into the same number (three, in this embodiment) of communicating chambers (62a-62c) as the number of the auxiliary heat exchange sections (52a-52c). The first communicating chamber (62a) which is the lowermost chamber communicates with all of the flat tubes (32) included in the first auxiliary heat exchange section (52a). The second communicating chamber (62b) which is located immediately above the first communicating chamber (62a) communicates with all of the flat tubes (32) included in the second auxiliary heat exchange section (52b). The third communicating chamber (62c) which is the uppermost chamber communicates with all of the flat tubes (32) included in the third auxiliary heat exchange section (52c).

[0050] The inner space of the second header-collecting pipe (70) is divided into a principal communicating space (71) corresponding to the principal heat exchange region (51) and an auxiliary communicating space (72) corresponding to the auxiliary heat exchange region (52).

[0051] The principal communicating space (71) is partitioned by two partition plates (39c) into portions located one above the other. Specifically, the partition plates (39c) partition the principal communicating space (71) into the same number (three, in this embodiment) of subspaces (71a-71c) as the number the principal heat exchange sections (51a-51c). The first subspace (71a) which is the lowermost subspace communicates with all of the flat tubes (31) included in the first principal heat exchange section (51a). The second subspace (71b) which is located immediately above the first subspace (71a) communicates with all of the flat tubes (31) included in the second principal heat exchange section (51b). The third subspace (71c) which is the uppermost subspace communicates with all of the flat tubes (31) included in the third principal heat exchange section (51c). The subspaces (71a-71c) serve as flow spaces in which the refrigerant upwardly flows when the outdoor heat exchanger (23) functions as an evaporator.

[0052] The auxiliary communicating space (72) is partitioned by two partition plates (39d) into portions located one above the other. Specifically, the partition plates (39d) partition the auxiliary communicating space (72) into the same number (three, in this embodiment) of subspaces (72a-72c) as the number of the auxiliary heat exchange sections (52a-52c). The fourth subspace (72a) which is the lowermost subspace communicates with all of the flat tubes (32) included in the first auxiliary heat exchange section (52a). The fifth subspace (72b) which is located immediately above the fourth subspace (72a) communicates with all of the flat tubes (32) included in the second auxiliary heat exchange section (52b). The sixth subspace (72c) which is the uppermost subspace communicates with all of the flat tubes (32) included in the third auxiliary heat exchange section (52c).

[0053] Two connection pipes (76, 77) are attached to the second header-collecting pipe (70). Each of the connection pipes (76, 77) is a circular pipe.

[0054] The first connection pipe (76) has an end connected to the second subspace (71b) corresponding to the second principal heat exchange section (51b) and the other end connected to the fourth subspace (72a) corresponding to the first auxiliary heat exchange section (52a). The second connection pipe (77) has an end connected to the third subspace (71c) corresponding to the third principal heat exchange section (51c) and the other end connected to the fifth subspace (72b) corresponding to the second auxiliary heat exchange section (52b). In the second header-collecting pipe (70), the sixth subspace (72c) corresponding to the third auxiliary heat exchange section (52c) and the first subspace (71a) corresponding to the first principal heat exchange section (51a) communicate with each other and together form a continuous space.

[0055] Thus, in the outdoor heat exchanger (23) of this embodiment, the first principal heat exchange section (51a) is connected in series to the third auxiliary heat exchange section (52c), the second principal heat exchange section (51b) is connected in series to the first auxiliary heat exchange section (52a), and the third principal heat exchange section (51c) is connected in series to the second auxiliary heat exchange section (52b).

[0056] As illustrated in FIG. 2, the outdoor heat exchanger (23) is equipped with a liquid connection member (80) and a gas connection pipe (85). The liquid connection member (80) and the gas connection pipe (85) are attached to the first header-collecting pipe (60).

[0057] The liquid connection member (80) is provided with a distributor (81) and three small diameter pipes (82a-82c). A pipe (17) connecting the outdoor heat exchanger (23) to the expansion valve (24) is in connection to a lower end of the distributor (81). An upper end of the distributor (81) is in connection to an end of each of the small diameter pipes (82a-82c). In the distributor (81), the pipe connected to the lower end communicates with the small diameter pipes (82a-82c). The other end of each of the small diameter pipes (82a-82c) is in connection to the first header-collecting pipe (60) and communicates with a corresponding one of the communicating chambers (62a-62c).

[0058] As illustrated also in FIG. 3, each of the small diameter pipes (82a-82c) opens in a portion near the lower end (i.e., a portion located below the middle in the vertical direction) of the corresponding one of the communicating chambers (62a-62c). Specifically, the first small diameter pipe (82a) opens in the portion near the lower end of the firstcommunicating chamber (62a). The second small diameter pipe (82b) opens in the portion near the lower end of the second communicating chamber (62b). The third small diameter pipe (82c) opens in the portion near the lower end of the third communication chamber (62c). The length of each of the small diameter pipes (82a-82c) is individually determined such that differences between flow rates at which the refrigerant flows into the heat exchange sections (52a-52c) become as small as possible.

[0059] The gas connection pipe (85) has an end connected to an upper portion of the first header-collecting pipe (60) and communicates with the upper space (61). The other end of the gas connection pipe (85) is in connection to a pipe 18 connecting the outdoor heat exchanger (23) to the third port of the four-way switching valve (22).

<Refrigerant Flow in Outdoor Heat Exchanger (When Functioning as Condenser)>

[0060] When the air conditioner (10) is performing the cooling operation, the outdoor heat exchanger (23) is functioning as a condenser. A flow of the refrigerant through the outdoor heat exchanger (23) during the cooling operation is now described.

[0061] The gaseous refrigerant discharged from the compressor (21) is supplied to the outdoor heat exchanger (23). The gaseous refrigerant sent from the compressor (21) passes through the gas connection pipe (85) and flows into the upper space (61) of the first header-collecting pipe (60), and then, is distributed to the flat tubes (31) of the principal heat exchange region (51). In the principal heat exchange sections (51a-51c) of the principal heat exchange region (51), the refrigerant having flowed into the fluid passages (34) of the flat tubes (31) dissipates heat into outdoor air and condenses while flowing through the fluid passages (34). Thereafter, the refrigerant flows into the corresponding subspaces (71a-71c) of the second header-collecting pipe (70).

[0062] The refrigerant having flowed into the subspaces (71a-71c) of the principal communicating space (71) is sent to the corresponding subspaces (72a-72c) of the auxiliary communicating space (72). Specifically, the refrigerant having flowed into the first subspace (71a) of the principal communicating space (71) downwardly flows and enters the sixth subspace (72c) of the auxiliary communicating space (72). The refrigerant having flowed into the second subspace (71b) of the principal communicating space (71) passes through the first connection pipe (76) and enters the fourth subspace (72a) of the auxiliary communicating space (72). The refrigerant having flowed into the third subspace (71c) of the principal communicating space (71) passes through the second connection pipe (77) and enters the fifth subspace (72b) of the auxiliary communicating space (72).

[0063] The refrigerant having flowed into the subspaces (72a-72c) of the auxiliary communicating space (72) is distributed to the flat tubes (32) of the corresponding auxiliary heat exchange sections (52a-52c). The refrigerant flowing though the fluid passages (34) of the flat tubes (32) dissipates heat into outdoor air to be converted into subcooled liquid, and then, flows into the corresponding communicating chambers (62a-62c) of the lower space (62) of the first header-collecting pipe (60). The refrigerant then flows out from the communicating chambers (62a-62c), passes through the small diameter pipes (82a-82c), and enters the distributor (81), where the flows of the refrigerant merge with one another. In this manner, the refrigerant flows out of the outdoor heat exchanger (23).

<Refrigerant Flow in Outdoor Heat Exchanger (When Functioning as Evaporator)>

[0064] When the air conditioner (10) is performing the heating operation, the outdoor heat exchanger (23) is functioning as an evaporator. A flow of the refrigerant through the outdoor heat exchanger (23) during the heating operation is now described.

[0065] The refrigerant having expanded upon passing through the expansion valve (24) and being in a gas-liquid two-phase state is supplied to the outdoor heat exchanger (23). The refrigerant sent from the expansion valve (24) flows into the distributor (81) of the liquid connection member (80), and thereafter, diverges and flows into the three small diameter pipes (82a-82c) to be distributed to the heat exchange sections(52a-52c).

[0066] Specifically, the refrigerant having flowed from the distributor (81) into the small diameter pipes (82a-82c) enters the corresponding communicating chambers (62a-62c) of the first header-collecting pipe (60). In the communicating chambers (62a-62c) of the first header-collecting pipe (60), the refrigerant is distributed to the flat tubes (32) of the corresponding auxiliary heat exchange sections (52a-52c). The refrigerant then flows through the fluid passages (34) of the flat tubes (32). While flowing through the fluid passages (34), the refrigerant absorbs heat from outdoor air, and part of the liquid refrigerant evaporates. The refrigerant having passed though the fluid passages (34) of the flat tubes (32) flows into the corresponding subspaces (72a-72c) of the auxiliary communicating space (72) of the second header-collecting pipe (70). The refrigerant having entered the subspaces (72a-72c) remains in the gas-liquid two-phase state.

[0067] The refrigerant having flowed into the subspaces (72a-72c) of the auxiliary communicating space (72) is sent to the corresponding sunspaces (71a-71c) of the principal communicating space (71). Specifically, the refrigerant having flowed into the fourth subspace (72a) of the auxiliary communicating space (72) passes through the first connection.

[0068] pipe (76) and enters a lower end portion of the second subspace (71b) of the principal communicating space (71). The refrigerant having flowed into the fifth subspace (72b) of the auxiliary communicating space (72) passes through the second connection pipe (77) and enters a lower end portion of the third subspace (71c) of the principal communicating space (71). The refrigerant having flowed into the sixth subspace (72c) of the auxiliary communicating space (72) upwardly flows and enters a lower end portion of the first subspace (71a) of the principal communicating space (71).

[0069] In each of the subspaces (71a-71c) of the principal communicating space (71), the refrigerant upwardly flows. The refrigerant in the subspaces (71a-71c) is distributed to the flat tubes (31) of the corresponding principal heat exchange sections (51a-51c). The refrigerant flowing through the fluid passages (34) of the flat tubes (31) absorbs heat from outdoor air and evaporates, thereby entering a substantially single-phase gas state. Thereafter, the refrigerant flows into the upper space (61) of the first header-collecting pipe (60). The refrigerant flows out of the outdoor heat exchanger (23) and passes through the gas connection pipe (85).

<Insertion Length L of Flat Tubes>

[0070] In the outdoor heat exchanger (23) of this embodiment, an insertion length L of the flat tubes (31) in the second header-collecting pipe (70) is determined such that the subspaces (71a-71c) of the principal communicating space (71) provided in the second header-collecting pipe (70) has an effective cross-sectional area A which is equal to a predetermined design value. The insertion length L is expressed in units of "mm" and the effective cross-sectional area A is expressed in units of "mm^2." In FIG. 5, the fins (36) are not shown.

[0071] As illustrated in FIG. 5A, the insertion length L of each flat tube (31) in the second header-collecting pipe (70) refers to the length of a portion of the flat tube (31) inserted in the corresponding one of the subspaces (71a-71c). Specifically, the insertion length L corresponds to the distance from the end face of the portion of the flat tube (31) inserted in a corresponding one of the subspaces (71a-71c) to the inner surface of the second header-collecting pipe (70).

[0072] The effective cross-sectional area A of each of the subspaces (71a-71c) corresponds to the area of the region marked with dots in FIG. 5D. The effective cross-sectional area A is obtained by subtracting a projected area A_1 of each flat tube (31) from a cross-sectional area A_0 of each of the subspaces (71a-71c) ($A = A_0 - A_1$). The cross-sectional area A_0 of each of the subspaces (71a-71c) corresponds to the area of the region marked with dots in FIG. 5B. That is, the cross-sectional area A_0 of each of the subspaces (71a-71c) is the area of a cross section of each of the subspaces (71a-71c) which is perpendicular to the axial direction of the second header-collecting pipe (70). Since the cross section of each of the subspaces (71a-71c) is circular, the cross-sectional area A_0 of each of the subspaces (71a-71c) is written as $(\pi/4)d^2$. The projected area A_1 of each flat tube (31) corresponds to the area of the region marked with dots in FIG. 5C. Specifically, the projected area A_1 of each flat tube (31) is an area corresponding to a portion of the flat tube (31) which is located in the subspaces (71a-71c) and which is projected onto a plane perpendicular to the axial direction of the second header-collecting pipe (70).

[0073] A width W of the flat tubes (31) is determined according to the design value of the capability of the outdoor heat exchanger (23) for example. An inside diameter d of the second header-collecting pipe (70) is determined such that the flat tubes (31) having the width W can be inserted into the second header-collecting pipe (70). Thus, when a design for the outdoor heat exchanger (23) is developed, the width W of the flat tubes (31) and the inside diameter d of the second header-collecting pipe (70) are determined first, and then, the insertion length L of the flat tubes (31) is determined such that the effective cross-sectional area A of the subspaces (71a-71c) becomes equal to the predetermined value.

[0074] As described above, the compressor (21) of the air conditioner (10) has the variable operation capacity. When the operation capacity of the compressor (21) varies, the flow rate at which the refrigerant circulates through the refrigerant circuit (20) varies, and consequently, the mass flow rate at which the refrigerant flows into the outdoor heat exchanger (23) varies. In the air conditioner (10) performing the heating operation, the flow rate at which the refrigerant circulates through the refrigerant circuit (20) (i.e., the mass flow rate at which the refrigerant flows into the outdoor heat exchanger (23)) varies within the range approximately from 90 kg/h to 270 kg/h inclusive. On the other hand, the outdoor heat exchanger (23) includes the three principal heat exchange sections (51a-51c), and the principal communicating space (71) of the second header-collecting pipe (70) is partitioned into three subspaces (71a-71c). Accordingly, when the outdoor heat exchanger (23) functions as the evaporator, the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) varies within the range approximately from 30 kg/h to 90 kg/h inclusive.

[0075] For the outdoor heat exchanger (23) of this embodiment, the upper limit value (i.e. 90 kg/h) of the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) is determined as a reference mass flow rate M_R. On the other hand, in the outdoor heat exchanger (23) of this embodiment, the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) is set to a value which is equal to or greater than ($1.91\,M_R - 22.7$) and equal to or smaller than ($1.96M_R + 30.8$). Accordingly, in the outdoor heat exchanger (23), the insertion length L of each flat tube (31) in the second header-collecting pipe (70) is determined such that the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) becomes equal to or greater than 149 mm^2 and equal to or smaller than 207 mm^2.

[0076] In the outdoor heat exchanger (23) of this embodiment, it is most desirable to determine the insertion length L of each flat tube (31) in the second header-collecting pipe (70) such that the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) becomes equal to 188 mm^2. For example, if the flat tube (31) has the width W of 18 mm and the second header-collecting pipe (70) has the inside diameter d of 21 mm, the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) becomes 188 mm^2 by setting the insertion length L of each flat tube (31) in the second header-collecting pipe (70) to 10 mm.

<Effective Cross-sectional Area A of Principal Communicating Space>

[0077] As described above, in the outdoor heat exchanger (23) of this embodiment, the insertion length L of each flat tube (31) in the second header-collecting pipe (70) is determined such that the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (70) becomes equal to the predetermined value.

[0078] In the outdoor heat exchanger (23), the effective cross-sectional area A [mm^2] of the subspaces (71a-71c) of the principal communicating space (71) is set to a value which is equal to or greater than ($1.91M_R$ - 22.7) and equal to or smaller than ($1.96M_R$ + 30.8). The reference mass flow rate M_R [kg/h] is a desired value within the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) when the outdoor heat exchanger (23) functions as the evaporator. When the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) is within the above range and the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) is equal to the reference mass flow rate M_R, the outdoor heat exchanger (23) functioning as an evaporator provides sufficient performance. The reason for this is now described with reference to FIGS. 6 and 7.

[0079] FIG. 6 illustrates relations between a flow velocity V of the refrigerant flowing in the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) and a performance rate of the outdoor heat exchanger (23) functioning as the evaporator. Specifically, FIG. 6 illustrates the relations shown under low-temperature heating conditions, rated heating conditions, and intermediate heating conditions which are the conditions of the heating operation of the air conditioner (10), and with varying the effective cross-sectional area A of the subspaces (71a-71c). The data shown in FIG. 6 were obtained from experiments conducted using several outdoor heat exchangers (23) and R410A as the refrigerant. Specifically, the outdoor heat exchangers (23) subjected to the experiments each included the flat tubes (31) having the width W of 18 mm and the second header-collecting pipe (70) having a circular cross section and the inside diameter d of 21 mm, but were different from one another only in the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71).

[0080] Under the low-temperature heating conditions, an evaporating temperature Te of the refrigerant in the outdoor heat exchanger (23) is -7°C and a mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) is 90 kg/h. Under the rated heating conditions, the evaporating temperature Te of the refrigerant in the outdoor heat exchanger (23) is 0°C and the mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) is 80 kg/h. Under the intermediate heating conditions, the evaporating temperature Te of the refrigerant in the outdoor heat exchanger (23) is 2°C and the mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) is 40 kg/h.

[0081] In FIG. 6, the horizontal axis represents the flow velocity V [m/s] at which the refrigerant flows in the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70). The flow velocity V is calculated by dividing a volume flow rate X [m^3/s] of the refrigerant flowing into the subspaces (71a-71c) by the effective cross-sectional area A [m^2] of the subspaces (71a-71c) (V = X/A). The volume flow rate X [m^3/s] of the refrigerant flowing into the subspaces (71a-71c) is calculated by dividing the mass flow rate M [kg/h] of the refrigerant flowing into the subspaces (71a-71c) by a density D [kg/m^3] of the refrigerant flowing into the subspaces (71a-71c) (X = (M/3600)/D).

[0082] In FIG. 6, the vertical axis represents the performance rate R of the outdoor heat exchangers (23) under the above operation conditions. The performance rate R is a percentage of the performance of each outdoor heat exchanger (23) relative to predetermined reference performance. The reference performance is the performance of the outdoor heat exchanger (23) including the subspaces (71a-71c) having the effective cross-sectional area A of 188 mm^2.

[0083] The performance rate R of each outdoor heat exchanger (23) under the low-temperature heating conditions is calculated by dividing performance Q_1 of each outdoor heat exchanger (23) under the low-temperature heating conditions by performance of the outdoor heat exchanger (23) including the subspaces (71a-71c) having the effective cross-sectional area A of 188 mm^2 and operating under the low-temperature heating conditions (i.e., by the reference performance Q_{01} under the low-temperature heating conditions) ($R = 100\ (Q_1/Q_{01})$). The performance rate R of each outdoor heat exchanger (23) under the rated heating conditions is calculated by dividing performance Q_2 of each outdoor heat exchanger (23) under the rated heating conditions by performance of the outdoor heat exchanger (23) including the subspaces (71a-71c) having the effective cross-sectional area A of 188 mm^2 and operating under the rated heating conditions (i.e., by the reference performance Q_{02} under the rated heating conditions) ($R = 100\ (Q_2/Q_{02})$). The performance rate R of each outdoor heat exchanger (23) under the intermediate heating conditions is calculated by dividing performance Q_3 of each outdoor heat exchanger (23) under the intermediate heating conditions by performance of the outdoor heat exchanger (23) including the subspaces (71a-71c) having the effective cross-sectional area A of 188 mm^2 and operating under the intermediate heating conditions (i.e., by the reference performance Q_{03} under the intermediate heating conditions) ($R = 100\ (Q_3/Q_{03})$). As a matter of course, the reference performance provided under the low-temperature heating conditions, that provided under the rated heating conditions, and that provided under the intermediate heating conditions are different from one another ($Q_{01} \neq Q_{02} \neq Q_{03}$).

[0084] The performance Q of the outdoor heat exchanger (23) is calculated according to the formula: $Q = G\ (h_{out} - h_{in})$ wherein G is a mass flow rate of the refrigerant passing through the outdoor heat exchanger (23), h_{in} is specific enthalpy of the refrigerant at an inlet of the outdoor heat exchanger (23), and h_{out} is specific enthalpy of the refrigerant at an outlet of the outdoor heat exchanger (23).

[0085] Four outdoor heat exchangers (23) differing in the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) were subjected to performance measurement under the low-temperature heating conditions. Specifically, the four outdoor heat exchangers (23) had the effective cross-sectional area A of 152 mm^2, 188 mm^2, 214 mm^2, and 240 mm^2, respectively. As illustrated in FIG. 6, the measurement indicated that the outdoor heat exchanger (23) of which the subspaces (71a-71c) had the effective cross-sectional area A of 188 mm^2 provided the highest performance.

[0086] Four outdoor heat exchangers (23) differing in the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) were subjected to performance measurement under the rated heating conditions. Specifically, the four outdoor heat exchangers (23) had the effective cross-sectional area A of 117 mm^2, 152 mm^2, 188 mm^2, and 214 mm^2, respectively. As illustrated in FIG. 6, the measurement indicated that the outdoor heat exchanger (23) of which the subspaces (71a-71c) had the effective cross-sectional area A of 152 mm^2 provided the highest performance.

[0087] Six outdoor heat exchangers (23) differing in the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) were subjected to performance measurement under the intermediate heating conditions. Specifically, the six outdoor heat exchangers (23) had the effective cross-sectional area A of 54 mm^2, 79 mm^2, 117 mm^2, 152 mm^2, 188 mm^2, and 214 mm^2, respectively. As illustrated in FIG. 6, the measurement indicated that the outdoor heat exchanger (23) of which the subspaces (71a-71c) had the effective cross-sectional area A of 79 mm^2 provided the highest performance.

[0088] The effective cross-sectional area A of the subspaces (71a-71c) with which the outdoor heat exchanger (23) provides the highest performance varies depending on the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions. The reason for this variation is as follows.

[0089] When the mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) is constant, the flow velocity V of the refrigerant flowing in the subspaces (71a-71c) decreases as the effective cross-section A of the subspaces (71a-71c) increases. In the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70), the refrigerant being in the gas-liquid two-phase state flows upwardly. Accordingly, a decrease in the flow velocity V at which the refrigerant flows in the subspaces (71a-71c) causes a large proportion of the liquid refrigerant having a higher density to flow into lower ones of the flat tubes (31) and a large proportion of the gaseous refrigerant having a lower density to flow into upper ones of the flat tubes (31). That is, the mass flow rate at which the refrigerant flows from the subspaces (71a-71c) into the flat tubes (31) becomes nonuniform.

[0090] In the upper flat tubes (31) into which a small proportion of the liquid refrigerant has flowed, the refrigerant enters a single-phase gas state before reaching the first header-collecting pipe (60), and the temperature of the refrigerant approaches the temperature of outdoor air. This causes a decrease in the amount of heat which the refrigerant in an upper portion of each of the principal heat exchange sections (51a-51c) exchanges with the air, and thereby reduces the performance of the outdoor heat exchanger (23).

[0091] When the mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) is constant, the flow velocity V of the refrigerant flowing in the subspaces (71a-71c) increases as the effective cross-section A of the subspaces (71a-71c) decreases. When the flow velocity V of the refrigerant flowing in the subspaces (71a-71c) increases, inertial force acting on the liquid refrigerant having a higher density increases. In the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70), the refrigerant being in the gas-liquid two-phase state flows upwardly. Consequently, a large proportion of the liquid refrigerant having been raised with great force flows into upper ones of the flat tubes (31), and a large proportion of the gaseous refrigerant having a lower density flows into lower ones of the flat tubes (31). That is, the mass flow rate at which the refrigerant flows from the subspaces (71a-71c) into the flat tubes (31) becomes nonuniform.

[0092] In the lower flat tubes (31) into which a small proportion of the liquid refrigerant has flowed, the refrigerant enters a single-phase gas state before reaching the first header-collecting pipe (60), and the temperature of the refrigerant approaches the temperature of outdoor air. This causes a decrease in the amount of heat which the refrigerant in a lower portion of each of the principal heat exchange sections (51a-51c) exchanges with the air, and thereby reduces the performance of the outdoor heat exchanger (23).

[0093] Thus, in the outdoor heat exchanger (23), when the flow velocity V at which the refrigerant flows in the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) is excessively high as well as when the flow velocity V is excessively low, the amounts of the refrigerant distributed to the flat tubes (31) communicating with the subspaces (71a-71c) become nonuniform, and consequently, the performance of the outdoor heat exchanger (23) decreases. On the other hand, when the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) is constant, the flow velocity V of the refrigerant flowing in the subspaces (71a-71c) is inversely proportional to the effective cross-sectional area A of the subspaces (71a-71c). Therefore, as described above, the effective cross-sectional area A of the subspaces (71a-71c) with which the outdoor heat exchanger (23) provides the highest performance varies depending on the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions.

[0094] FIG. 7 illustrates the experimental results of FIG. 6 in a different manner. Specifically, FIG. 7 illustrates the experimental results rearranged and shown in terms of the relation between the mass flow rate M at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) and the effective cross-sectional area A of the subspaces (71a-71c).

[0095] A linear approximate equation for the points at which the outdoor heat exchanger (23) provides the highest performance under the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions (i.e., the point where M = 40 kg/h and A = 79 mm^2, the point where M = 80 kg/h and A = 152 mm^2, and the point where M = 90 kg/h and A =188 mm^2) is expressed as Equation 1 below:

$$\text{[Equation 1]} \qquad A = 1.96M$$

[0096] A linear approximate equation for the points at which the performance of the outdoor heat exchanger (23) becomes 95% of the highest performance and the effective cross-sectional area A becomes greater than the value calculated according to Equation 1, under the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions, (i.e., the point where M = 40 kg/h and A = 109 mm^2, the point where M = 80 kg/h and A = 187 mm^2, and the point where M = 90 kg/h and A = 207 mm^2) is expressed as Equation 2 below:

$$\text{[Equation 2]} \qquad A = 1.96M + 30.8$$

[0097] A linear approximate equation for the points at which the performance of the outdoor heat exchanger (23) becomes 95% of the highest performance and the effective cross-sectional area A becomes smaller than the value calculated according to Equation 1, under the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions, (i.e., the point where M = 40 kg/h and A = 53 mm^2, the point where M = 80 kg/h and A = 130 mm^2, and the point where M = 90 kg/h and A = 149 mm^2) is expressed as Equation 3 below:

$$\text{[Equation 3]} \qquad A = 1.91M - 22.7$$

[0098] Therefore, according to this embodiment, causing the effective cross-sectional area A of the subspaces (71a-71c) to become equal to or greater than ($1.91M_R$ - 22.7) and equal to or smaller than ($1.96M_R$ + 30.8) by adjusting the insertion length L of the flat tubes (31) in the second header-collecting pipe (70) enables the outdoor heat exchanger (23) functioning as the evaporator in an operational state where the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) is equal to the reference mass flow rate M_R to provide performance corresponding to 95% or more of the highest performance that can be provided in the same operational state.

- Advantages of Embodiment -

[0099] As described above, the outdoor heat exchanger (23) of this embodiment is configured such that the upper limit value (i.e., 90 kg/h) of the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) is determined as the reference mass flow rate M_R, and the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) is equal to or greater than ($1.91M_R$ - 22.7) and equal to or smaller than ($1.96M_R$ + 30.8). Therefore, according to this embodiment, under the low-temperature heating conditions where the operation capacity of the compressor (21) provided in the refrigerant circuit (20) is maximized, it is possible to cause the outdoor heat exchanger (23) to provide performance corresponding to 95% or more of its highest performance.

[0100] Under operating conditions where the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) is lower than the mass flow rate under the low-temperature heating conditions, the performance of the outdoor heat exchanger (23) of this embodiment can become less than 95% of the highest performance. Under such operating conditions, however, the operation capacity of the compressor (21) is smaller than the maximum. Therefore, under the operating conditions where the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) is lower than the mass flow rate under the low-temperature heating conditions, the heating performance of the air conditioner (10) can be ensured by increasing the operation capacity of the compressor (21).

[0101] Thus, as described in this embodiment, setting the effective cross-sectional area A of the subspaces (71a-71c) of the principal communicating space (71) on the basis of the reference mass flow rate M_R which is the upper limit value (i.e., 90 kg/h) of the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) enables the outdoor heat exchanger (23) to provide sufficient performance in a state where the operation capacity of the compressor (21) is maximized. As a result, the heating performance of the air conditioner (10) can be increased without increasing the size of the outdoor heat exchanger (23).

- Variation 1 of Embodiment -

[0102] In the outdoor heat exchanger (23) of this embodiment, a flow rate lower than the upper limit value of the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) may be determined as the reference mass flow rate M_R, and the insertion length L of each flat tube (31) in the second header-collecting pipe (70) may be set such that the effective cross-sectional area A of the subspaces (71a-71c) becomes equal to or greater than $(1.91M_R - 22.7)$ and equal to or smaller than $(1.96M_R + 30.8)$.

[0103] Here, the period during which the operation capacity of the compressor (21) of the refrigerant circuit (20) is maximized is not so long throughout the year. In other words, the period during which the compressor (21) is operated with an operation capacity smaller than the maximum operation capacity is longer than the period during which the compressor (21) is operated with the maximum operation capacity.

[0104] Accordingly, it is possible to determine, as the reference mass flow rate M_R, a mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) in an operational state which appears most frequently throughout the year. Setting, on the basis of this flow rate determined as the reference mass flow rate M_R, the insertion length L of each flat tube (31) in the second header-collecting pipe (70) such that the effective cross-sectional area A of the subspaces (71a-71c) becomes equal to or greater than $(1.91M_R - 22.7)$ and equal to or smaller than $(1.96M_R + 30.8)$ enables the outdoor heat exchanger (23) being under the operational state which appears most frequently throughout the year to provide performance corresponding to 95% or more of the highest performance which can be provided under the same operational state. Thus, according to this variation, it is possible to improve operational efficiency of the air conditioner (10) being in the operational state which appears most frequently throughout the year, and consequently, annual power consumption of the air conditioner (10) can be reduced.

-Variation 2 of Embodiment-

[0105] The effective cross-sectional area A of the subspaces (71a-71c) of the outdoor heat exchanger (23) of this embodiment may be set using only Equation 1, which is the linear approximate equation for the points at which the outdoor heat exchanger (23) provides the highest performance under the low-temperature heating conditions, the rated heating conditions, and the intermediate heating conditions.

[0106] Specifically, as illustrated in FIG. 8, when a value within the variation range of the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) of the second header-collecting pipe (70) is determined as the reference mass flow rate M_R, the insertion length L of each flat tube (31) in the second header-collecting pipe (70) may be set such that the effective cross-sectional area A of the subspaces (71a-71c) becomes equal to or greater than $(1.96M_R - b)$ and equal to or smaller than $(1.96M_R + a)$. For example, when a = 30.0 and b = 25.0, the outdoor heat exchanger (23) in a state where the mass flow rate at which the refrigerant flows into the subspaces (71a-71c) of the principal communicating space (71) is equal to the reference mass flow rate M_R provides performance corresponding to about 95% or more of the highest performance that can be provided in the same state.

INDUSTRIAL APPLICABILITY

[0107] As described above, the present invention is useful for heat exchangers including a plurality of flat tubes and header-collecting pipes connected to the plurality of flat tubes.

DESCRIPTION OF REFERENCE CHARACTERS

[0108]

20	Refrigerant circuit
23	Heat exchanger (Outdoor heat exchanger)
31	Flat tube

36 Fin
60 First header-collecting pipe
70 Second header-collecting pipe
71a First subspace (Flow space)
71b Second subspace (Flow space)
71c Third subspace (Flow space)

Claims

1. A refrigerant circuit (20) configured to perform a refrigerating cycle and comprising a heat exchanger (23), the heat exchanger (23) comprising:

   a plurality of flat tubes (31); a first header-collecting pipe (60) having inserted therein an end portion of each of the flat tubes (31); a second header-collecting pipe (70) having inserted therein the other end portion of each of the flat tubes (31); and a plurality of fins (36) joined to the flat tubes (31), wherein
   the second header-collecting pipe (70) forms flow spaces (71a-71c) which communicate with the plurality of flat tubes (31) and in which a refrigerant being in a gas-liquid two-phase state flows upwardly when the heat exchanger functions as an evaporator,
   an effective cross-sectional area of the flow spaces (71a-71c) is an area obtained by subtracting a projected area which corresponds to a portion of each flat tube (31) located in a corresponding one of the flow spaces (71a-71c) and which is projected onto a plane perpendicular to an axial direction of the second header-collecting pipe (70), from an area of a cross section of the corresponding one of the flow spaces (71a-71c) which is perpendicular to the axial direction of the second header-collecting pipe (70), **characterized in that** the effective cross-sectional area of the flow spaces (71a-71c) is set based on a mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator; and
   a value included in a variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator is determined as a reference mass flow rate M_R [kg/h], and

   the effective cross-sectional area A [mm^2] of the flow spaces (71a-71c) is equal to or greater than ($1.91M_R$ - 22.7) and equal to or smaller than ($1.96M_R$ + 30.8), or is equal to or greater than ($1.96M_R$ - 25.0) and equal to or smaller than ($1.96M_R$ + 30.0).

2. The refrigerant circuit of claim 1, wherein
   the reference mass flow rate M_R [kg/h] is an upper limit value of the variation range of the mass flow rate at which the refrigerant flows into the flow spaces (71a-71c) when the heat exchanger functions as the evaporator.

3. The refrigerant circuit of any one of claims 1-2, wherein
   the first header-collecting pipe (60) and the second header-collecting pipe (70) are in an upright position, and
   the heat exchanger is configured such that the refrigerant flows into a lower end portion of each of the flow spaces (71a-71c) when the heat exchanger functions as the evaporator.

Patentansprüche

1. Kältemittelkreislauf (20), konfiguriert zum Durchführen eines Kältezyklus und umfassend einen Wärmetauscher (23), wobei der Wärmetauscher (23) umfasst:

   eine Mehrzahl von flachen Rohren (31); ein erstes Grund-Sammelrohr (60), in welches ein Endbereich von jedem der flachen Rohre (31) eingeführt ist; ein zweites Grund-Sammelrohr (70), in welches der andere Endbereich von jedem der flachen Rohre (31) eingeführt ist; und eine Mehrzahl von Lamellen (36), die an die flachen Rohre (31) gefügt sind, wobei
   das zweite Grund-Sammelrohr (70) Strömungszwischenräume (71a - 71c) bildet, die mit der Mehrzahl von flachen Rohren (31) kommunizieren und in welchen ein Kältemittel, das in einem Gas-Flüssigkeits-Zweiphasen-Zustand ist, aufwärts strömt, wenn der Wärmetauscher als ein Verdampfer fungiert,
   eine effektive Querschnittsfläche der Strömungszwischenräume (71a - 71c) eine Fläche ist, die erhalten wird, indem eine projizierte Fläche, die einem Bereich von jedem flachen Rohr (31) entspricht, der sich in einem entsprechenden der Strömungszwischenräume (71a - 71c) befindet und der auf eine Ebene senkrecht zu einer Axialrichtung des zweiten Grund-Sammelrohrs (70) projiziert ist, von einer Fläche eines Querschnitts des entsprechenden der Strömungszwischenräume (71a - 71c) subtrahiert wird, die senkrecht zur Axialrichtung des zweiten Grund-Sammelrohrs (70) ist, **dadurch gekennzeichnet, dass** die effektive Querschnittsfläche der Strömungszwischenräume (71a - 71c) auf Basis einer Massenströmungsrate vorgegeben ist, bei welcher das Kältemittel in die Strömungszwischenräume (71a - 71c) strömt, wenn der Wärmetauscher als Verdampfer fungiert; und

ein Wert, der in einem Variationsbereich der Massenströmungsrate enthalten ist, bei welcher das Kältemittel in die Strömungszwischenräume (71a - 71c) strömt, wenn der Wärmetauscher als Verdampfer fungiert, als die Massenströmungsrate M_R [kg/h] bestimmt ist, und
die effektive Querschnittsfläche A [mm^2] der Strömungszwischenräume (71a - 71c), die gleich groß oder größer als (1,91 M_R - 22,7) ist und gleich groß oder kleiner als (1,96 M_R + 30,8) ist, gleich groß oder größer als (1,96 M_R - 25,0) ist und gleich groß oder kleiner als (1,96 M_R + 30,0) ist.

2. Kältemittelkreislauf nach Anspruch 1, wobei die Referenz-Massenströmungsrate M_R [kg/h] ein oberer Grenzwert des Variationsbereichs der Massenströmungsrate ist, bei welcher das Kältemittel in die Strömungs-Zwischenräume (71a - 71c) strömt, wenn der Wärmetauscher als Verdampfer fungiert.

3. Kältemittelkreislauf nach einem der Ansprüche 1 - 2, wobei
das erste Grund-Sammelrohr (60) und das zweite Grund-Sammelrohr (70) in einer aufrechten Position sind, und
der Wärmetauscher so konfiguriert ist, dass das Kältemittel in einen unteren Endbereich von jedem der Strömungs-Zwischenräume (71a - 71c) strömt, wenn der Wärmetauscher als der Verdampfer fungiert.

Revendications

1. Circuit de réfrigérant (20) configuré pour effectuer un cycle de réfrigération et comprenant un échangeur de chaleur (23), l'échangeur de chaleur (23) comprenant :

une pluralité de tubes plats (31) ; un premier collecteur-tuyau de collecte (60) dans lequel est insérée une partie d'extrémité de chacun des tubes plats (31) ; un second collecteur-tuyau de collecte (70) dans lequel est insérée l'autre partie d'extrémité de chacun des tubes plats (31) ; et une pluralité d'ailettes (36) reliées aux tubes plats (31), dans lequel
le second collecteur-tuyau de collecte (70) forme des espaces d'écoulement (71a-71c) qui communiquent avec la pluralité de tubes plats (31) et dans lesquels un réfrigérant se trouvant dans un état biphasique gazeux-liquide s'écoule vers le haut lorsque l'échangeur de chaleur fonctionne comme un évaporateur,
une aire de section transversale efficace des espaces d'écoulement (71a-71c) est une aire obtenue en soustrayant une aire projetée qui correspond à une partie de chaque tube plat (31) située dans l'un correspondant des espaces d'écoulement (71a-71c) et qui est projetée sur un plan perpendiculaire à une direction axiale du second collecteur-tuyau de collecte (70), d'une aire d'une section transversale de celui correspondant des espaces d'écoulement (71a-71c) qui est perpendiculaire à la direction axiale du second collecteur-tuyau de collecte (70), **caractérisé en ce que**
l'aire de section transversale efficace des espaces d'écoulement (71a-71c) est réglée sur la base d'un débit d'écoulement massique auquel le réfrigérant s'écoule dans les espaces d'écoulement (71a-71c) lorsque l'échangeur de chaleur fonctionne comme l'évaporateur ; et
une valeur incluse dans une plage de variation du débit d'écoulement massique auquel le réfrigérant s'écoule dans les espaces d'écoulement (71a-71c) lorsque l'échangeur de chaleur fonctionne comme l'évaporateur est déterminée en tant que débit d'écoulement massique de référence M_R[kg/h], et
l'aire de section transversale effective A [mm^2] des espaces d'écoulement (71a-71c) est égale à ou supérieure à (1,91M_R - 22,7) et égale à ou inférieure à (1,96M_R + 30,8), ou est égale à ou supérieure à (1,96M_R - 25,0) et égale à ou inférieure à (1,96M_R + 30,0).

2. Circuit de réfrigérant selon la revendication 1, dans lequel
le débit d'écoulement massique de référence M_R [kg/h] est une valeur limite supérieure de la plage de variation du débit d'écoulement massique auquel le réfrigérant s'écoule dans les espaces d'écoulement (71a-71c) lorsque l'échangeur de chaleur fonctionne comme l'évaporateur.

3. Circuit de réfrigérant selon l'une quelconque des revendications 1-2, dans lequel
le premier collecteur-tuyau de collecte (60) et le second collecteur-tuyau de collecte (70) sont dans une position droite, et
l'échangeur de chaleur est configuré de sorte que le réfrigérant s'écoule dans une partie d'extrémité inférieure de chacun des espaces d'écoulement (71a-71c) lorsque l'échangeur de chaleur fonctionne comme l'évaporateur.

FIG.1

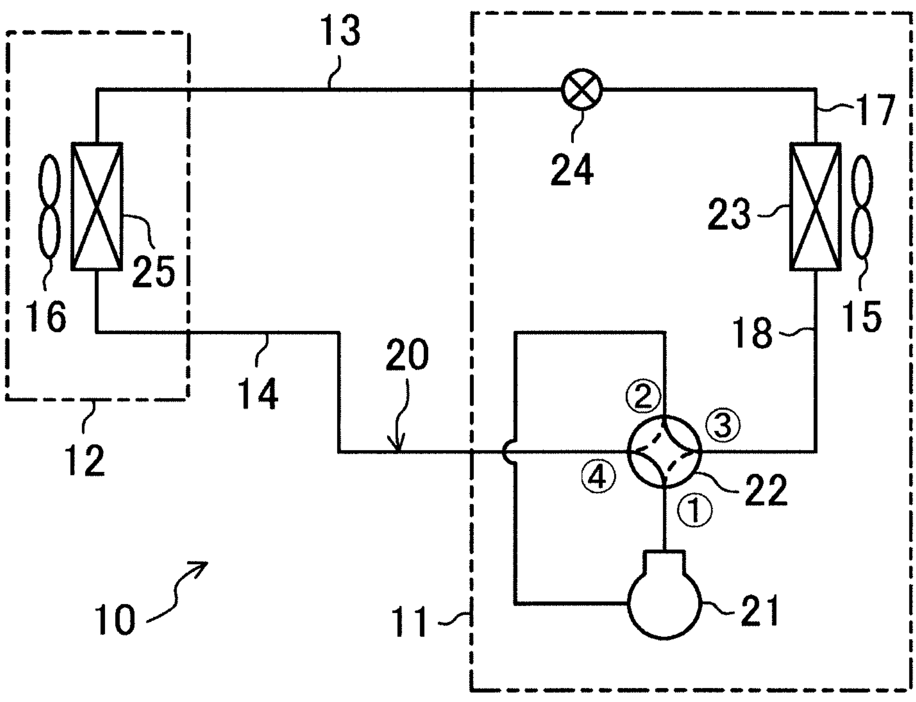


FIG.2

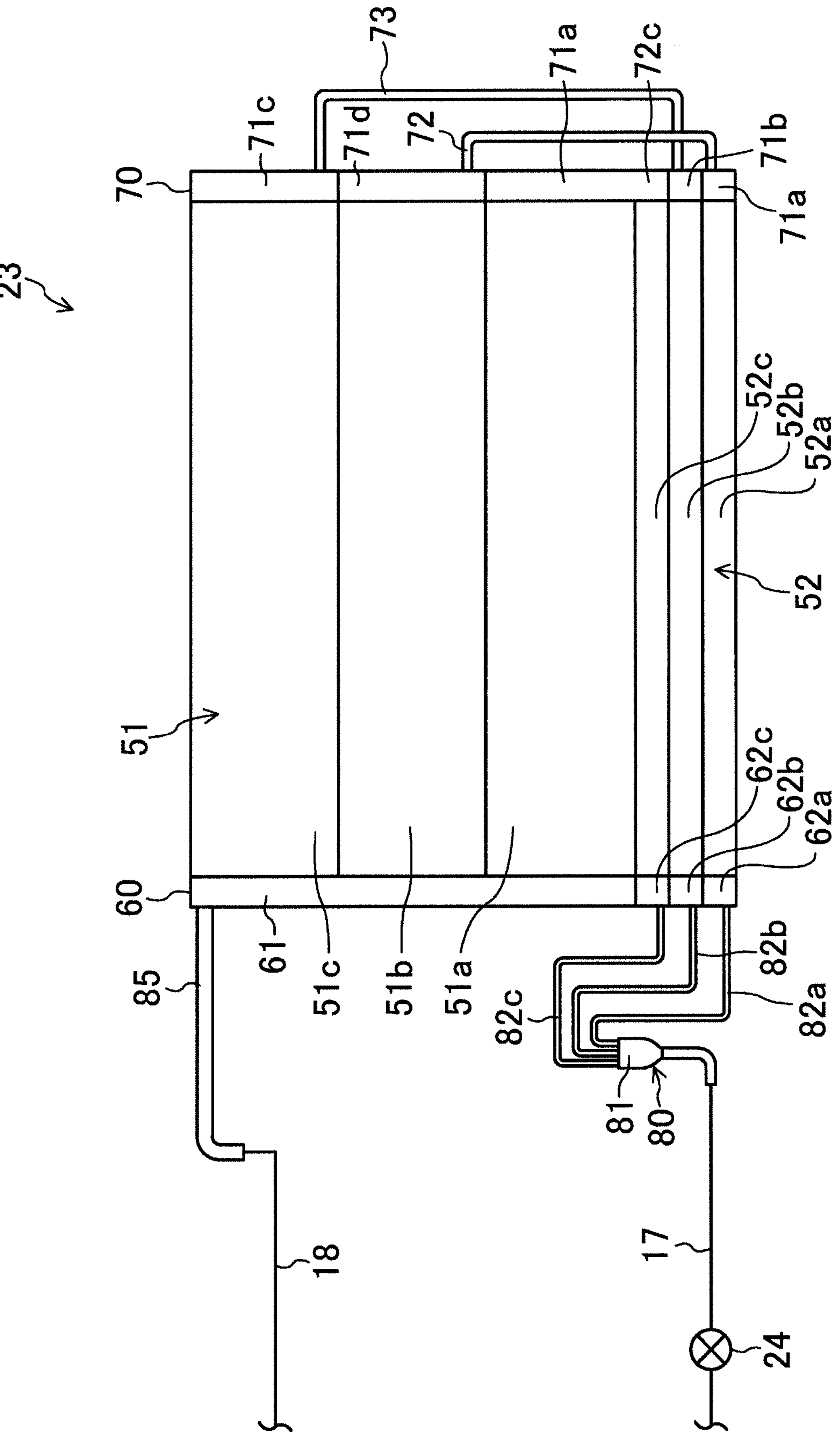

23
70
71c
71d
72
73
71a
72c
71b
71a
51
52
52c
52b
52a
60
61
51c
51b
51a
62c
62b
62a
85
82c
82b
82a
81
80
18
17
24

FIG.3

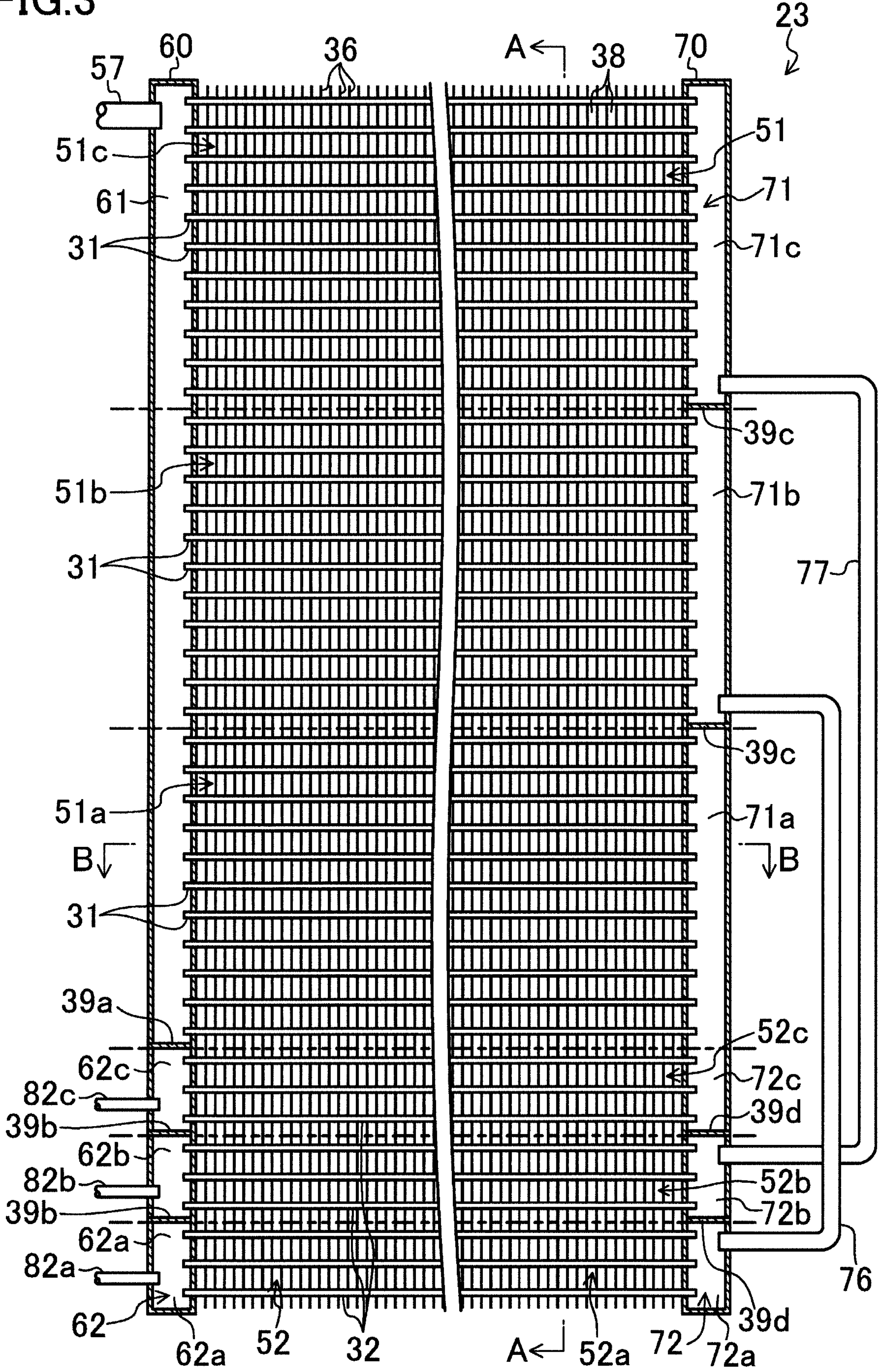

23
57
60
36
A
38
70
51c
51
61
71
31
71c
39c
51b
71b
31
77
39c
51a
71a
B
B
31
39a
52c
62c
72c
82c
39d
39b
62b
82b
52b
72b
39b
62a
82a
76
62
62a
52
32
A
52a
72
72a
39d

FIG.4

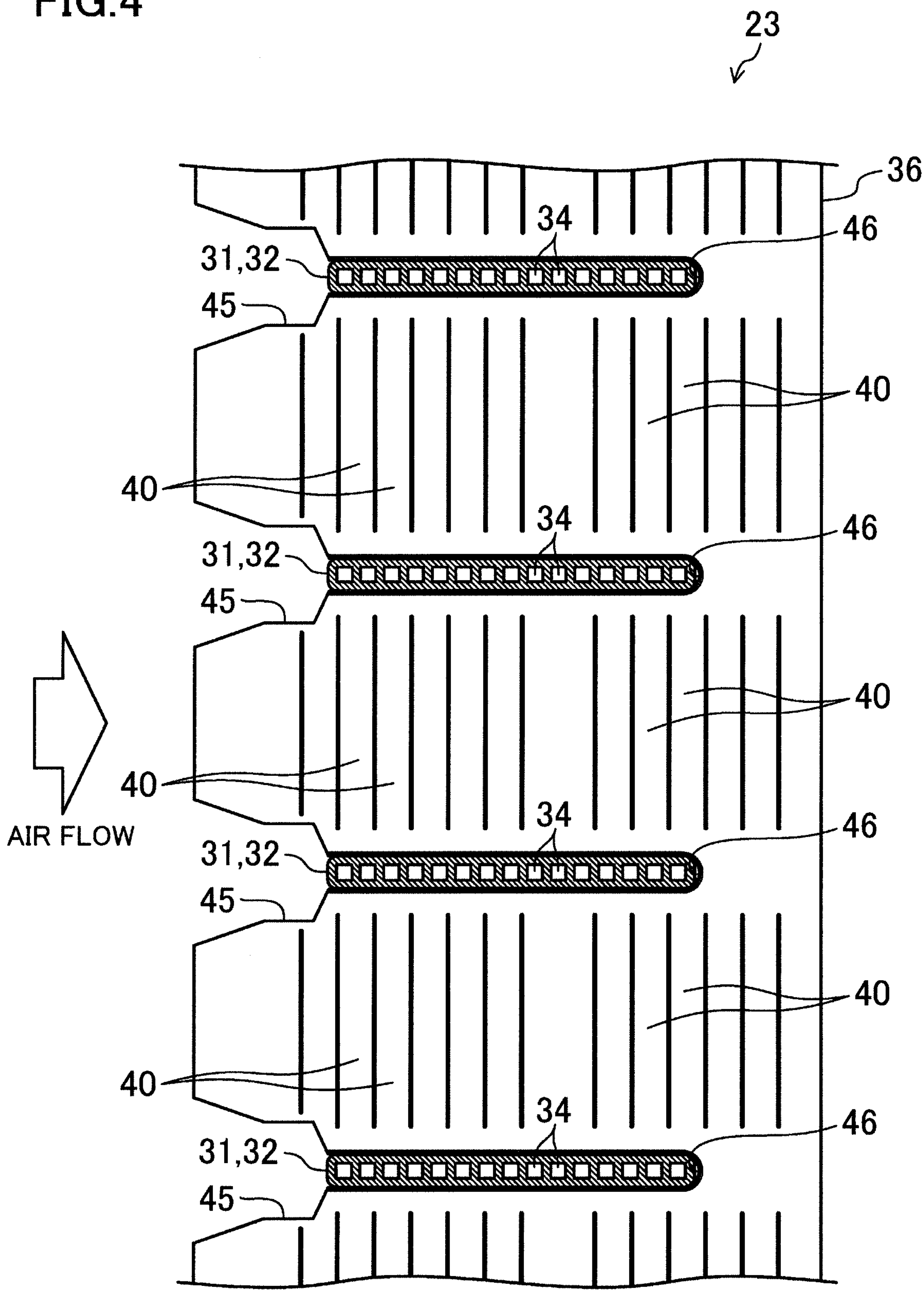


FIG.5A

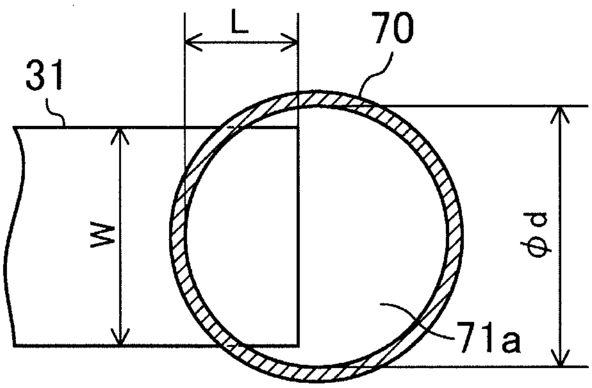


FIG.5B

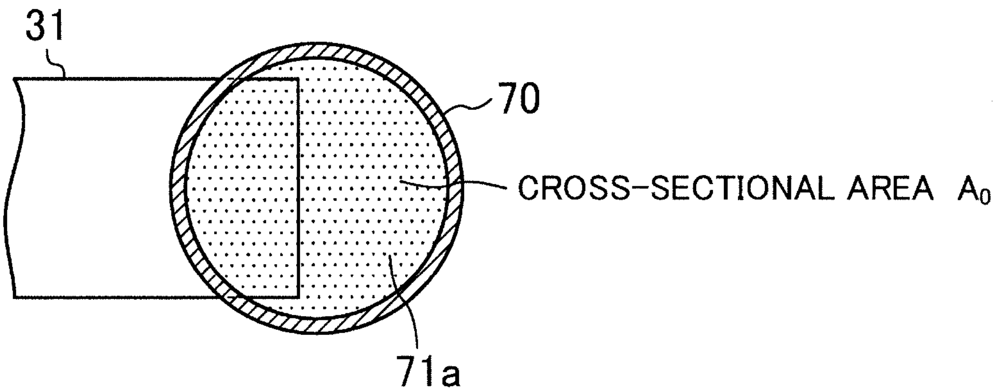


FIG.5C

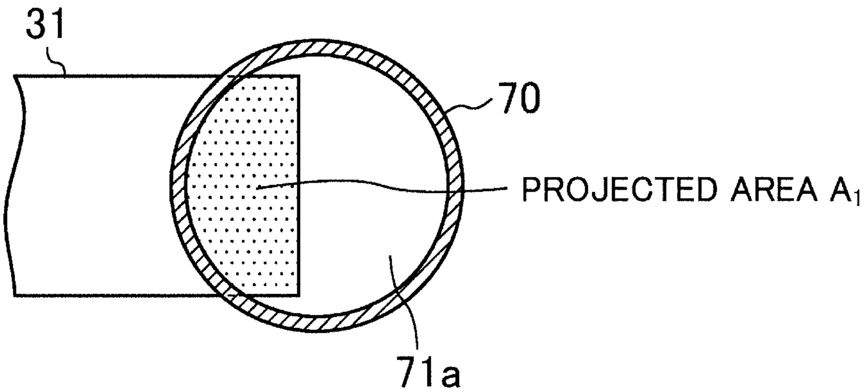


FIG.5D

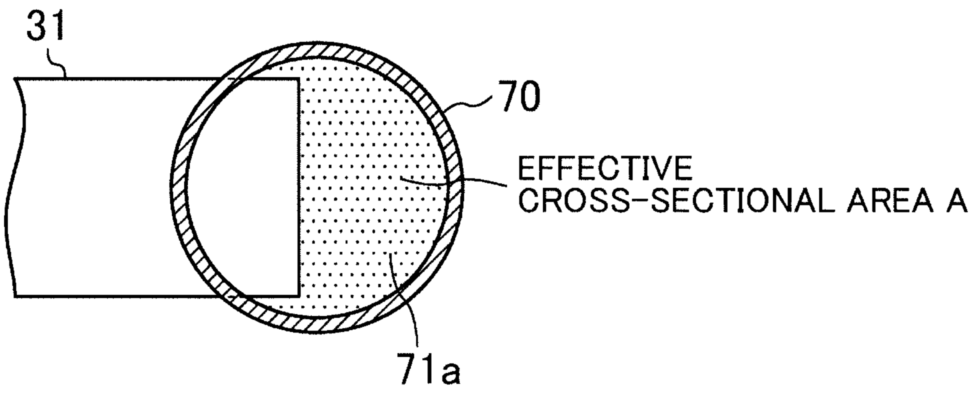


FIG.6

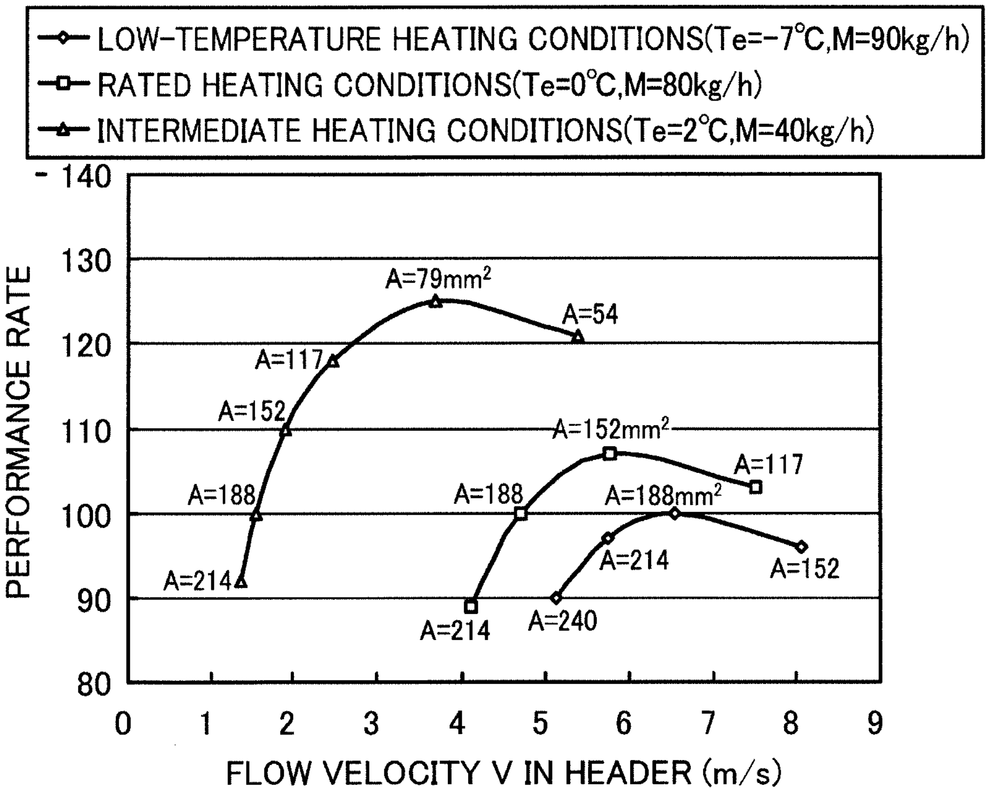

LOW-TEMPERATURE HEATING CONDITIONS(Te=-7°C,M=90kg/h)
RATED HEATING CONDITIONS(Te=0°C,M=80kg/h)
INTERMEDIATE HEATING CONDITIONS(Te=2°C,M=40kg/h)
PERFORMANCE RATE
140
130
120
110
100
90
80
A=79mm2
A=54
A=117
A=152
A=188
A=214
A=152mm2
A=117
A=188
A=188mm2
A=214
A=152
A=214
A=240
0 1 2 3 4 5 6 7 8 9
FLOW VELOCITY V IN HEADER (m/s)

FIG.7

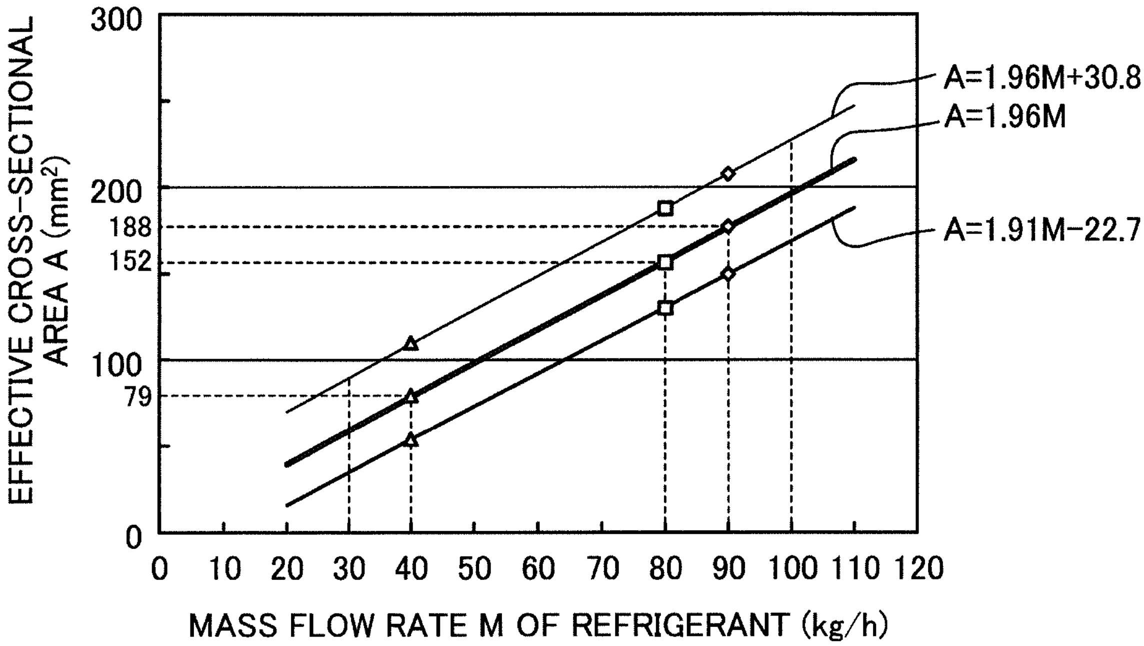
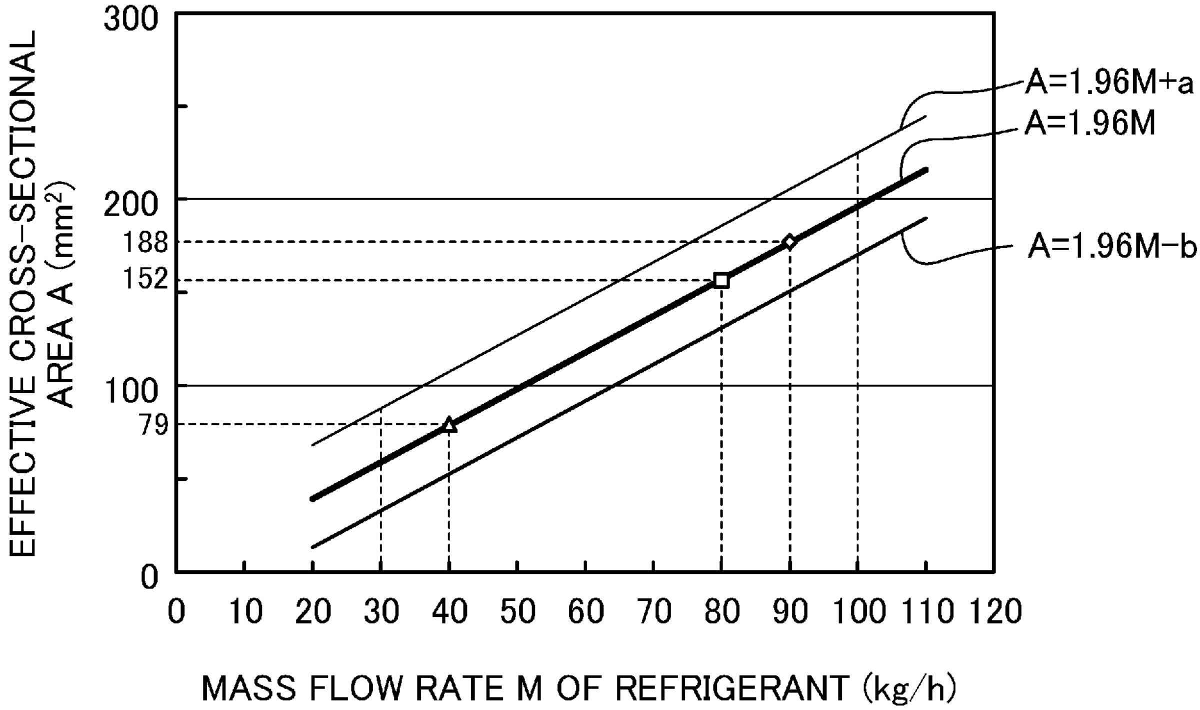

EFFECTIVE CROSS-SECTIONAL AREA A (mm2)
300
200
188
152
100
79
0
A=1.96M+30.8
A=1.96M
A=1.91M−22.7
0 10 20 30 40 50 60 70 80 90 100 110 120
MASS FLOW RATE M OF REFRIGERANT (kg/h)

FIG.8


EFFECTIVE CROSS-SECTIONAL AREA A (mm2)
300
200
188
152
100
79
0
0 10 20 30 40 50 60 70 80 90 100 110 120
MASS FLOW RATE M OF REFRIGERANT (kg/h)
A=1.96M+a
A=1.96M
A=1.96M−b

REFERENCES CITED IN THE DESCRIPTION

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- JP 2006105545 A **[0002] [0003]**
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