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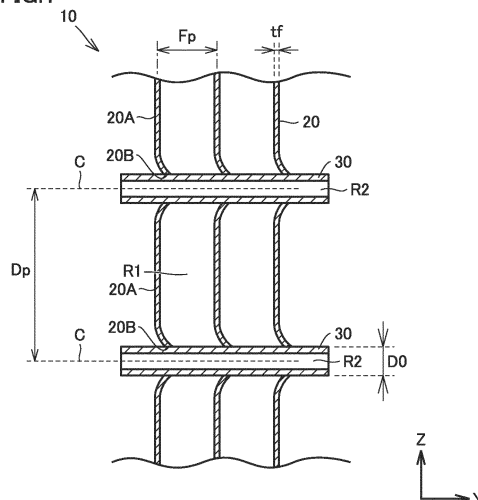
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(57) The heat exchanger (10) includes: a plurality of fins (20) disposed, spaced apart from each other; and a plurality of heat transfer tubes (30) inserted in the plurality of fins. The plurality of heat transfer tubes have round profiles. The plurality of heat transfer tubes have outer circumferential surfaces in contact with the plurality of

fins. The plurality of heat transfer tubes have outer diameters (D_0) of 5.4 mm or less. The plurality of fins and the heat transfer tubes are disposed so that ratios (t_f/D_0) of thicknesses (t_f) of the plurality of fins to the outer diameters (D_0) are 0.03 or greater.

FIG.1

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Description

TECHNICAL FIELD

5 [0001] The present disclosure relates to a heat exchanger.

BACKGROUND ART

10 [0002] As a method for enhancing the adhesion in a heat exchanger between a fin and a heat transfer tube that has a round profile, a mechanical tube expansion system is known which uses a pipe expander rod to expand the heat transfer tube (e.g., see Japanese Patent Laying-Open No. 2016-20757).

CITATION LIST

15 PATENT LITERATURE

[0003] PTL 1: Japanese Patent Laying-Open No. 2016-20757

[0004] Even with the mechanical tube expansion system, the heat exchange performance deteriorates if the contact thermal resistance between the fin and the heat transfer tube is high.

20 [0005] In particular, a heat exchanger capable of handling a reduced refrigerant of recent years, increasingly includes a heat transfer tube that has a reduced diameter. As shown in Fig. 8, the ratio of the contact thermal resistance to the entire thermal resistance of the heat exchanger is confirmed to increase with a reduction of the outer diameter of the heat transfer tube. Therefore, deterioration of the heat exchange performance associated with an increase of such a ratio is a concern with the heat exchanger capable of handling a reduced refrigerant.

25 [0006] A primary object of the present disclosure is to provide a heat exchanger capable of inhibiting the deterioration of heat exchanger performance associated with a contact thermal resistance.

SOLUTION TO PROBLEM

30 [0007] A heat exchanger according to the present disclosure includes: a plurality of fins disposed, spaced apart from each other; and a plurality of heat transfer tubes inserted in the plurality of fins. The plurality of heat transfer tubes have round profiles. The plurality of heat transfer tubes have outer circumferential surfaces in contact with the plurality of fins. The plurality of heat transfer tubes have outer diameters D_0 of 5.4 mm or less. The plurality of fins and the heat transfer tubes are disposed so that ratios t_f/D_0 of thicknesses t_f of the plurality of fins to the outer diameters D_0 are 0.03 or greater.

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ADVANTAGEOUS EFFECTS OF INVENTION

[0008] According to the present disclosure, a heat exchanger can be provided which is capable of inhibiting deterioration of the heat exchanger performance associated with a contact thermal resistance.

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BRIEF DESCRIPTION OF DRAWINGS

[0009]

45 Fig. 1 is a partial sectional view of a heat exchanger according to the present embodiment.
 Fig. 2 is a partial sectional view for illustrating arrangement of multiple heat transfer tubes in the heat exchanger of Fig. 1.
 Fig. 3 is a graph showing a ratio t_f/D_0 of a thickness t_f of a fin to an outer diameter D_0 of the heat transfer tube versus a ratio $A_0K/\Delta P$ of heat exchange performance A_0K to an extratube pressure loss ΔP_0 .
 50 Fig. 4 is a graph showing the thickness t_f of the fin where the heat transfer tube has the outer diameter D_0 of 5.3 mm versus the ratio of the heat exchange performance A_0K to the extratube pressure loss ΔP_0 .
 Fig. 5 is a graph showing the thickness t_f of the fin where the heat transfer tube has the outer diameter D_0 of 5.3 mm versus the ratio of the heat exchange performance A_0K to the extratube pressure loss ΔP_0 .
 Fig. 6 is a graph showing an amount by which the heat transfer tube is expanded versus a contact heat transfer coefficient between the fin and the heat transfer tube.
 55 Fig. 7 is a partial sectional view of a variation of the heat exchanger of Fig. 1.
 Fig. 8 is a graph showing the outer diameter D_0 of the heat transfer tube versus a ratio of a contact thermal resistance between the fin and the heat transfer tube to an entire thermal resistance of the heat exchanger.

DESCRIPTION OF EMBODIMENTS

[0010] Hereinafter, the present embodiment will be described, with reference to the accompanying drawings. Note that, for purpose of explanation, Figs. 1 and 2 introduce a first direction X, a second direction Y, and a third direction Z, which intersect with each other.

[0011] As shown in Figs. 1 and 2, a heat exchanger 10 according to the present embodiment is a plate-fin heat exchanger. The heat exchanger 10 includes multiple fins 20 and multiple heat transfer tubes 30.

[0012] The fins 20 are plate fins. The fins 20 each extend along the first direction X and the third direction Z. The fins 20 are disposed, spaced apart from each other in the second direction Y. The number of fins 20 may be two or more, for example, three or more fins 20. Three or more fins 20 are equidistantly disposed in the second direction Y, for example. Multiple fin collars are formed on each fin 20. Each fin collar is disposed so as to have one heat transfer tube 30 inserted therethrough. The center-to-center distance between adjacent two fins 20 in the second direction Y will be referred to as a fin pitch F_p .

[0013] As shown in Fig. 1, each fin 20 has a first surface 20A facing the first surface 20A of an adjacent fin 20 in the second direction Y, and a second surface 20B extending in a direction intersecting with the first surface 20A and in contact with an outer circumferential surface of the heat transfer tube 30. The second surface 20B is an inner circumferential surface of the fin collar of each fin 20. The length of the second surface 20B in the second direction Y is equal to a thickness t_f of the fin 20, for example.

[0014] The heat transfer tubes 30 are round tubes. Stated differently, each heat transfer tube 30 has a round profile. The heat transfer tube 30 is inserted into the fin 20. The outer circumferential surface of the heat transfer tube 30 is in contact with the fin 20. The heat transfer tubes 30 are expanded by a mechanical tube expansion system, for example. For example, one or more grooves are formed in the inner circumferential surface of each heat transfer tube 30. For example, the heat transfer tubes 30 are what is called grooved-heat transfer tubes. For example, the heat transfer tube 30 is not brazed to the fin 20. In this case, heat transfer tube 30 is formed of a single material, for example. Stated differently, the heat transfer tube 30 is not formed of a clad material. Note that the heat transfer tube 30 may be brazed to the fin 20. In this case, preferably, the heat transfer tube 30 is formed of a clad material. In the following, inside the heat transfer tube 30 will be referred to as intratube, and outside the heat transfer tube 30 will be referred to as extratube.

[0015] The heat transfer tubes 30 extend along the second direction Y. The number of heat transfer tubes 30 may be one or more, for example, four or more heat transfer tubes 30. The heat transfer tubes 30 are disposed spaced apart by a spacing L_p (see Fig. 2) from each other in the first direction X and spaced apart by a spacing D_p (see Fig. 2) from each other in the third direction Z. Note that the arrangement of the heat transfer tubes 30 along the first direction X in which an air flows will be referred to as a column, and the arrangement of the heat transfer tubes 30 along the third direction Z will be referred to as an array. The number of columns of the heat transfer tubes 30 in the first direction X may be one or more, for example, three or more columns. The number of tiers of heat transfer tubes 30 in the third direction Z may be one or more, for example, three or more tiers. The distance between the central axes of adjacent two heat transfer tubes 30 in the first direction X will be referred to as a column pitch L_p . The distance between the central axes of adjacent two heat transfer tubes 30 in the third direction Z will be referred to as a tier pitch D_p .

[0016] Examples of the materials comprising the fins 20 include, but are not particularly limited to, copper (Cu) or aluminum (Al). Examples of the materials comprising the heat transfer tube 30, but are not particularly limited to, Cu or Al. For example, the material comprising each fin 20 includes Al, and the material comprising each heat transfer tube 30 includes Cu.

[0017] A flow passage R1 is formed between adjacent two fins 20 in the second direction Y. A first heat-transfer medium such as an air flows through the flow passage R1 in the first direction X. A flow passage R2 is formed inside each heat transfer tube 30. A second heat-transfer medium such as a refrigerant flows through the flow passage R2 in the second direction Y. The first heat-transfer medium exchanges heat with the second heat-transfer medium via the fin 20 and the heat transfer tube 30.

[0018] The fins 20 and the heat transfer tubes 30 are disposed so that a ratio t_f/D_0 of the thickness t_f (unit: mm) of each fin 20 to the outer diameter D_0 (unit: mm) of each heat transfer tube 30 is greater than or equal to 0.03. The outer diameters D_0 of the heat transfer tubes 30 are for example, less than or equal to 5.4 mm. Stated from a different perspective, the fins 20 and the heat transfer tubes 30 are disposed so that a ratio $AoK/\Delta P$ of a heat exchanger performance AoK to an extratube pressure loss ΔP (unit: Pa) is not less than 100% when the outer diameters D_0 of the fins 20 and the heat transfer tubes 30 are at least less than or equal to 5.4 mm.

[0019] The outer diameters D_0 of the heat transfer tubes 30 are, for example, greater than or equal to 5.2 mm and less than or equal to 5.4 mm. In this case, preferably, the fins 20 and the heat transfer tubes 30 are disposed so that the ratio t_f/D_0 is greater than or equal to 0.03 and less than or equal to 0.034. Stated from a different perspective, the fins 20 and the heat transfer tubes 30 are disposed so that the ratio $AoK/\Delta P$ of the heat exchanger performance AoK to the extratube pressure loss ΔP is 102% or greater when the outer diameters D_0 of the fins 20 and the heat transfer tubes 30 are greater than or equal to 5.2 mm and less than or equal to 5.4 mm.

[0020] The outer diameters D_0 of the heat transfer tubes 30 are, for example, greater than or equal to 3.6 mm and less than or equal to 3.8 mm. In this case, preferably, the fins 20 and the heat transfer tubes 30 are disposed so that the ratio tf/D_0 is greater than or equal to 0.034 and less than or equal to 0.058. Stated from a different perspective, the fins 20 and the heat transfer tubes 30 are disposed so that the ratio $AoK/\Delta P$ of the heat exchanger performance AoK to the extratube pressure loss ΔP is 102% or greater when the outer diameters D_0 of the fins 20 and the heat transfer tubes 30 are greater than or equal to 3.6 mm and less than or equal to 3.8 mm. Methods of calculation of the heat exchanger performance AoK and the extratube pressure loss ΔP will be described below.

[0021] Note that, if the heat transfer tube 30 is expanded by the mechanical tube expansion system, the outer diameter D_0 is the outer diameter of the expanded heat transfer tube 30. The tube expansion rate is, but not particularly limited to, greater than or equal to 5% and less than or equal to 8%, for example. The thickness tf and the outer diameter D_0 are, while they can be measured by any method, measured by a vernier caliper, for example.

[0022] Fig. 3 is a graph showing the ratio tf/D_0 versus the ratio $AoK/\Delta P$, more specifically, a graph showing changes in ratio $AoK/\Delta P$ with varying fin thickness tf relative to the outer diameter D_0 of the heat transfer tube. In Fig. 3, the ratio tf/D_0 is indicated on the horizontal axis and the ratio $AoK/\Delta P$ is indicated on the vertical axis. As one example, Fig. 3 shows changes in ratio $AoK/\Delta P$ with varying thickness tf of the fin in contact with the heat transfer tube that has the outer diameter D_0 of 3.7 mm, 5.3 mm, and 7.4 mm.

[0023] Fig. 4 is a graph showing changes in ratio $AoK/\Delta P$ with varying thickness tf of the fin in contact with the heat transfer tube that has the outer diameter D_0 of 5.3 mm. Fig. 5 is a graph showing changes in ratio $AoK/\Delta P$ with varying thickness tf of the fin in contact with the heat transfer tube that has the outer diameter D_0 of 3.7 mm. In Figs. 4 and 5, the thickness tf is indicated on the horizontal axis, and the ratio $AoK/\Delta P$ is indicated on the vertical axis.

[0024] As shown in Figs. 3, 4, and 5, the ratio $AoK/\Delta P$ is confirmed as being at its peak when the thickness tf is varied relative to a given outer diameter D_0 .

[0025] As shown in Fig. 3, when the outer diameter D_0 is 5.3 mm, the ratio $AoK/\Delta P$ is 100% or greater if the ratio tf/D_0 is greater than or equal to 0.021 and less than or equal to 0.04. The ratio $AoK/\Delta P$ is 102% or greater if the ratio tf/D_0 is greater than or equal to 0.026 and less than or equal to 0.034. The ratio $AoK/\Delta P$ is 103% or greater if the ratio tf/D_0 is 0.03.

[0026] As shown in Fig. 4, when the outer diameter D_0 is 5.3 mm, the ratio $AoK/\Delta P$ is 100% or greater if the thickness tf of the fin 20 is greater than or equal to 0.11 mm and less than or equal to 0.21 mm. The ratio $AoK/\Delta P$ is 102% or greater if the thickness tf is thicker than 0.12 mm and thinner than 0.20 mm. The ratio $AoK/\Delta P$ is 103% or greater if the thickness tf is greater than or equal to 0.15 mm and less than or equal to 0.17 mm. The ratio $AoK/\Delta P$ is at the maximum if the thickness tf is greater than or equal to 0.15 mm and less than or equal to 0.16 mm.

[0027] As shown in Fig. 3, when the outer diameter D_0 is 3.7 mm, the ratio $AoK/\Delta P$ is 100% or greater if the ratio tf/D_0 is greater than or equal to 0.03. The ratio $AoK/\Delta P$ is 103% or greater if the ratio tf/D_0 is greater than or equal to 0.034. The ratio $AoK/\Delta P$ is 108% or greater if the ratio tf/D_0 is greater than or equal to 0.046 and less than or equal to 0.058.

[0028] As shown in Fig. 5, when the outer diameter D_0 is 3.7 mm, the ratio $AoK/\Delta P$ is 100% or greater if the thickness tf of the fin 20 is greater than or equal to 0.11 mm and less than or equal to 0.21 mm. The ratio $AoK/\Delta P$ is 103% or greater if the thickness tf is greater than or equal to 0.12 mm. The ratio $AoK/\Delta P$ is 107% or greater if the thickness tf is greater than or equal to 0.15 mm. The ratio $AoK/\Delta P$ is 108% or greater if the thickness tf is greater than or equal to 0.17 mm and less than or equal to 0.21 mm. The ratio $AoK/\Delta P$ is at the maximum if the thickness tf is greater than or equal to 0.18 mm and less than or equal to 0.20 mm.

[0029] A method of calculation of the heat exchanger performance AoK and the pressure loss ΔP is as follows: The heat exchanger performance AoK is defined by the following Equation (1), using an intratube heat transfer coefficient α_i , a contact heat transfer coefficient α_c , and an extratube heat transfer coefficient α_a . Note that Equation (1) disregards the thermal resistance of the heat transfer tube in the direction of thickness thereof and the thermal resistance due to fouling in the tube because they are very small, as compared to the contact thermal resistance between the heat transfer tube and the refrigerant flowing therethrough, and the contact thermal resistance between the heat transfer tube and an air flowing outside the heat transfer tube. The heat transfer coefficients α_i , α_c , and α_a are in unit of $W / (m^2 \cdot K)$.

[MATH 1]

$$AoK = \left(\frac{1}{A_{pi} \cdot \alpha_i} + \frac{1}{A_{co} \cdot \alpha_c} + \frac{1}{(A_p + \eta \cdot A_F) \cdot \alpha_a} \right)^{-1} \quad \cdot \quad \cdot \quad \cdot \quad (1)$$

[0030] An intratube heat transfer area A_{pi} , a contact area A_{co} of the fin and the heat transfer tube, a surface area A_p of the outer circumferential surface of the heat transfer tube, and a surface area A_F of the fin in Equation (1) are set as specifications of the heat exchanger 10. These areas are in unit of m^2 .

[0031] The intratube heat transfer coefficient α_i in Equation (1) is calculated using an equation by Koyama et. al,

specifically, Equations (2) and (3):
[MATH 2]

$$\alpha_i = 241.2 \cdot X_{tt}^{-0.2878} \cdot Pr_l^{0.33} \cdot \frac{k_l}{d_i} \cdot \cdot \cdot (2)$$

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \cdot \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \cdot \cdot \cdot (3)$$

[0032] A Prandtl number Pr_l of the refrigerant and a thermal conductivity k_l (unit: W/m · K) of the refrigerant in Equation (2), and a density ρ_l (unit: g/m³) of a saturated liquid, a viscous modulus μ_l of the saturated liquid, a density ρ_v (unit: g/m³) of a saturated vapor, and a viscous modulus μ_v of the saturated vapor in Equation (3) are physical property values of the refrigerant. An inner diameter d_i (unit: m) of the heat transfer tube in Equation (3) is set as a specification of the heat exchanger 10. A vapor quality x in Equation (3) is a representative vapor quality, specifically, 0.5. In other words, the intratube heat transfer coefficient α_i calculated using Equations (2) and (3) is an average heat transfer coefficient.

[0033] The contact heat transfer coefficient α_c in Equation (1) is calculated using Equation (4):
[MATH 3]

$$\alpha_c = (2 \cdot 104 \cdot \Delta d + 2.5) \cdot tf \cdot 10^7 \cdot \cdot \cdot (4)$$

[0034] where Δd (unit: m) is an amount of expansion of tube, that is, a difference between the outer diameter D_0 of the expanded heat transfer tube 30, and the diameter of the fin collar before the heat transfer tube 30 is expanded.

[0035] The extratube heat transfer coefficient α_a in Equation (1) is calculated using an equation by Seshita, specifically, Equation (5):

$$\alpha_a = Nu \cdot k_a / D_e \cdot \cdot \cdot (5)$$

$$Re = \frac{Ve \cdot De}{\nu_a} \cdot \cdot \cdot (6)$$

$$Nu_{a_lRe} = C_o \cdot \left(\frac{Re_{a_lRe} \cdot Pr_a \cdot De_c}{N_L \cdot L_p} \right)^{c_1} \cdot \cdot \cdot (7)$$

$$Ve_{ac} = \frac{F_p \cdot D_p}{A_c} \cdot \left(\frac{\rho_{ai}}{\rho_a} \right) \cdot Ve_{af} \cdot \cdot \cdot (8)$$

$$Nu_{a_hRe} = C_o \cdot \left(\frac{Re_{a_hRe} \cdot Pr_a \cdot De_{min}}{N_L \cdot L_p} \right)^{c_1} \cdot \cdot \cdot (9)$$

$$Ve_{amax} = \frac{F_p \cdot D_p}{A_{min}} \cdot \left(\frac{\rho_{ai}}{\rho_a} \right) \cdot Ve_{af} \cdot \cdot \cdot (10)$$

[0036] An air Nusselt number Nu in Equation (5) is calculated using Equation (7), if a calculated value of an air Reynolds number Re defined by Equation (6) is less than a threshold (e.g., 400). An air Reynolds number Re_{a_lRe} in Equation (7) is calculated by assigning an average velocity of fluid Ve_{ac} at a free through-flow cross-sectional area A_c calculated using Equation (8) to an average velocity of fluid Ve of Equation (6), and assigning a representative inter-fin length De_c relative to a free-flow volume Vo to a representative inter-fin length De of Equation (6).

[0037] The air Nusselt number Nu of Equation (5) is calculated using Equation (9), if the calculated value of the air Reynolds number Re defined by Equation (6) is equal to or greater than or equal to the threshold (e.g., 400). The air Reynolds number Re_{a_hRe} of Equation (9) is calculated by assigning an average velocity of fluid Ve_{max} at a minimum flow passage cross-sectional area A_{min} calculated using Equation (10) to the average velocity of fluid Ve of Equation (6), and assigning a representative inter-fin length D_{min} relative to a minimum free through-flow volume V_{min} to the representative inter-fin length De of Equation (6).

[0038] The thermal conductivity k_a (unit: $W/(m \cdot K)$) of air of Equation (5), the dynamic viscous modulus ν_a of air of Equation (6), and the Prandtl number Pra of air are physical property values of air that are determined depending on the temperature, pressure, etc. In Equations (8) and (10), ρ_a and ρ_{ai} are a density of air.

[0039] A number of columns N_L of the heat transfer tube and the column pitch L_P (unit: m) of the heat transfer tubes of Equations (7) and (9), and the fin pitch F_P (unit: m) and the tier pitch D_P (unit: m) of the heat transfer tubes of Equations (8) and (10) are set as specifications of the heat exchanger. In Equation (7), a constant C_0 is 2.1, and a constant C_1 is 0.38. In Equation (9), C_0 is 0.12, and C_1 is 0.64.

[0040] A fin efficiency η of Equation (1) is calculated using Equations (11) and (12) below. A thermal conductivity k_f (unit: $W/(m \cdot K)$) of the fin, an equivalent diameter (unit: m) of the fin, and a diameter D_c (unit: m) of the fin collar before the heat transfer tube 30 is expanded in Equation (11) below are set as specifications of the heat exchanger.

[MATH 5]

$$\eta = \left\{ 1 + \alpha_a \cdot \frac{(d_F - d_c)^2}{6 \cdot k_F \cdot t_f} \cdot \left(\frac{d_F}{d_c} \right)^{0.5} \right\}^{-1} \cdot \cdot \cdot (11)$$

$$d_F = \left\{ \frac{4}{\pi} \cdot D_p \cdot L_p \right\}^{0.5} \cdot \cdot \cdot (12)$$

[0041] The extratube pressure loss ΔP is calculated using Equation (13) below, if the calculated value of the air Reynolds number Re defined by Equation (6) is less than the threshold (e.g., 400). The representative inter-fin length De_c of Equation (13) is calculated relative to a free-flow volume V_c . A flow-loss factor f_{lRe} of Equation (13) is calculated using Equation (14).

[0042] The extratube pressure loss ΔP is calculated using Equation (15) below, if a calculated value of the air Reynolds number Re defined by Equation (6) is greater than or equal to the threshold (e.g., 400). A representative inter-fin length De_{min} of Equation (15) is calculated relative to a minimum flow passage cross-sectional area V_{min} . The flow-loss factor f_{lRe} of Equation (15) is calculated using Equation (16).

[MATH 6]

$$\Delta P_{o_lRe} = f_{lRe} \cdot \frac{2 \cdot L_p \cdot N_L \cdot \rho_a \cdot Ve_{ac}^2}{De_c} \cdot \cdot \cdot (13)$$

$$f_{lRe} = \left\{ 0.43 + 35.1 \left(\frac{Re_{a_lRe} \cdot De_c}{L_p \cdot N_L} \right)^{-1.07} \right\} \cdot \left(\frac{De_c}{L_p \cdot N_L} \right) \cdot \cdot \cdot (14)$$

$$\Delta P_{o_hRe} = f_{hRe} \cdot \frac{2 \cdot L_p \cdot N_L \cdot \rho_a \cdot Ve_{amax}^2}{De_{min}} \cdot \cdot \cdot (15)$$

$$f_{hRe} = 0.026 + 27.0 \cdot Re_{a_hRe}^{-1.27} \cdot \cdot \cdot (16)$$

[0043] Next, operational advantages of the heat exchanger 10 according to the present embodiment are described.

[0044] As described above, the ratio of the contact thermal resistance to the entire thermal resistance of the heat exchanger is confirmed to increase with a reduction of the outer diameter of the heat transfer tube to less than 7 mm, as shown in Fig. 8. Therefore, deterioration of the heat exchange performance associated with an increase of the ratio above is a concern with a heat exchanger. In order to reduce the contact thermal resistance, it is contemplated to increase the contact heat transfer coefficient α_c calculated from Equation (4). In view of Equation (4), at least one of an amount of tube expansion Δd and the fin thickness t_f needs to be increased in order to increase the contact heat transfer coefficient α_c .

[0045] Fig. 6 is a graph showing the amount of tube expansion Δd versus the contact heat transfer coefficient α_c calculated from Equation (4), where the fin thickness t_f is constant and the amount of tube expansion Δd is varied. Referring to Equation (4) and Fig. 6, the contact heat transfer coefficient α_c is independent of the outer diameter D_o of the heat transfer tube, and increases with an increase of the amount of tube expansion Δd . However, in practice, an increase of the amount of tube expansion increases the likelihood of causing failures in manufacturing such as breakage of fin collars. Thus, there is the upper limit for the amount of tube expansion.

[0046] In contrast, the contact heat transfer coefficient α_c increases with an increase of the fin thickness t_f . An increase of the wind speed of the air flowing between the fans, that is, increases of the average velocity of fluid Ve_{ac} of Equation (8) and the average velocity of fluid Ve_{max} of Equation (10) increase an extratube transfer coefficient α_a , where the fin pitch F_p is constant and the fin thickness t_f is increased. However, an increase of the wind speed of the air flowing between the fins increases the average velocity of fluid Ve_{ac} of Equation (13) and the average velocity of fluid Ve_{max} of Equation (15), thereby increasing the extratube pressure loss ΔP_o . Therefore, if a heat exchanger that has a high extratube pressure loss ΔP_o is mounted on a unit, the flow rate of the air flowing between the fans may be less than a sufficient rate expected, and the heat exchanger performance may therefore not be exercised as expected.

[0047] In contrast, in the heat exchanger 10, the outer diameters D_o of the heat transfer tubes 30 are less than or equal to 5.4 mm, and the fins 20 and the heat transfer tubes 30 are disposed so that the ratio t_f/D_o of the thickness t_f of each fin 20 to the outer diameter D_o of each heat transfer tube 30 is greater than or equal to 0.03. Therefore, the ratio $AoK/\Delta P$ of the heat exchanger 10 is 100% or greater, as shown in Figs. 3 through 5. In other words, in the heat exchanger 10, the fins 20 and the heat transfer tubes 30 are disposed so that an increase of the extratube pressure loss ΔP_o can be inhibited, while increasing the heat exchanger performance AoK by increasing the contact heat transfer coefficient α_c . As a result, the heat exchanger performance of the heat exchanger 10 is higher than the heat exchanger performance of the heat exchanger in which the fin thickness t_f is simply increased in order to inhibit an increase of the contact thermal resistance.

[0048] While Figs. 3 and 4 show only results of calculation when the outer diameter D_o is 5.3 mm, it should be noted that the inventors confirmed that the ratio $AoK/\Delta P$ is 100% or greater even when the outer diameter D_o is 5.4 mm, if the ratio t_f/D_o is 0.03 or greater.

[0049] In one embodiment of the heat exchanger 10, multiple fins 20 and multiple heat transfer tubes 30 are disposed so that the outer diameter D_o is greater than or equal to 5.2 mm and less than or equal to 5.4 mm and the ratio t_f/D_o is 0.034 or less.

[0050] In another one embodiment of the heat exchanger 10, multiple fins 20 and multiple heat transfer tubes 30 are disposed so that the outer diameter D_o is greater than or equal to 3.6 mm and less than or equal to 3.8 mm and the ratio t_f/D_o is greater than or equal to 0.034 and less than or equal to 0.058.

[0051] In the heat exchanger 10, since the ratio t_f/D_o is 0.03 or greater, the heat exchanger performance is enhanced even though the heat transfer tubes 30 are expanded by the mechanical tube expansion system. Therefore, there is no need for the heat exchanger 10 to have the heat transfer tubes 30 brazed to the fins 20 for the purpose of reducing the contact thermal resistance. When the heat transfer tubes 30 are not brazed to the fins 20, each heat transfer tube 30 may not be formed of a clad material. In other words, the heat transfer tube 30 may be formed of a single material. When the heat transfer tubes 30 are not brazed to the fins 20, the material cost of the heat transfer tubes 30 and the manufacturing cost associated with brazing can be reduced.

[0052] Fig. 7 is a cross-sectional view of a variation of the heat exchanger 10 of Fig. 1. As shown in Fig. 7, each fin 20 has the first surface 20A facing the first surface 20A of an adjacent fin 20 in the second direction Y, and the second surface 20B extending in a direction intersecting with the first surface 20A. A portion of the first surface 20A is an inner

circumferential surface of the fin collar of each fin 20 and in contact with the outer circumferential surface of the heat transfer tube 30. The ratio $AoK/\Delta P$ in the heat exchanger 10 of Fig. 7 is 100% or greater if the ratio tf/D_0 is greater than or equal to 0.03, as with the heat exchanger 10 of Fig. 1.

[0053] While the embodiment according to the present disclosure has been described as described above, the embodiment can be also modified in various ways. The scope of the present disclosure is not limited to the embodiment. The scope of the present disclosure is defined by the appended claims. All changes which come within the meaning and range of equivalency of the appended claims are to be embraced within their scope.

REFERENCE SIGNS LIST

[0054] 10 heat exchanger; 20 fin; 20A first surface; 20B second surface; and 30 heat transfer tube.

Claims

1. A heat exchanger, comprising:

a plurality of fins disposed, spaced apart from each other; and
a plurality of heat transfer tubes inserted in the plurality of fins, wherein
the plurality of heat transfer tubes have round profiles,
the plurality of heat transfer tubes have outer circumferential surfaces in contact with the plurality of fins,
the plurality of heat transfer tubes have outer diameters D_0 of 5.4 mm or less, and
the plurality of fins and the heat transfer tubes are disposed so that ratios tf/D_0 of thicknesses tf of the plurality of fins to the outer diameters D_0 are 0.03 or greater.

2. The heat exchanger according to claim 1, wherein
the plurality of fins and the heat transfer tubes are disposed so that the outer diameters D_0 are greater than or equal to 5.2 mm and less than or equal to 5.4 mm and the ratios tf/D_0 are 0.034 or less.

3. The heat exchanger according to claim 1, wherein
the plurality of fins and the heat transfer tubes are disposed so that the outer diameters D_0 are greater than or equal to 3.6 mm and less than or equal to 3.8 mm and the ratios tf/D_0 are greater than or equal to 0.034 and less than or equal to 0.058.

4. The heat exchanger according to any one of claims 1 to 3, wherein
the plurality of heat transfer tubes are each formed of a single material.

5. The heat exchanger according to any one of claims 1 to 4, wherein materials comprising the plurality of fins include aluminum, and
materials comprising the plurality of heat transfer tubes include copper.

FIG.1

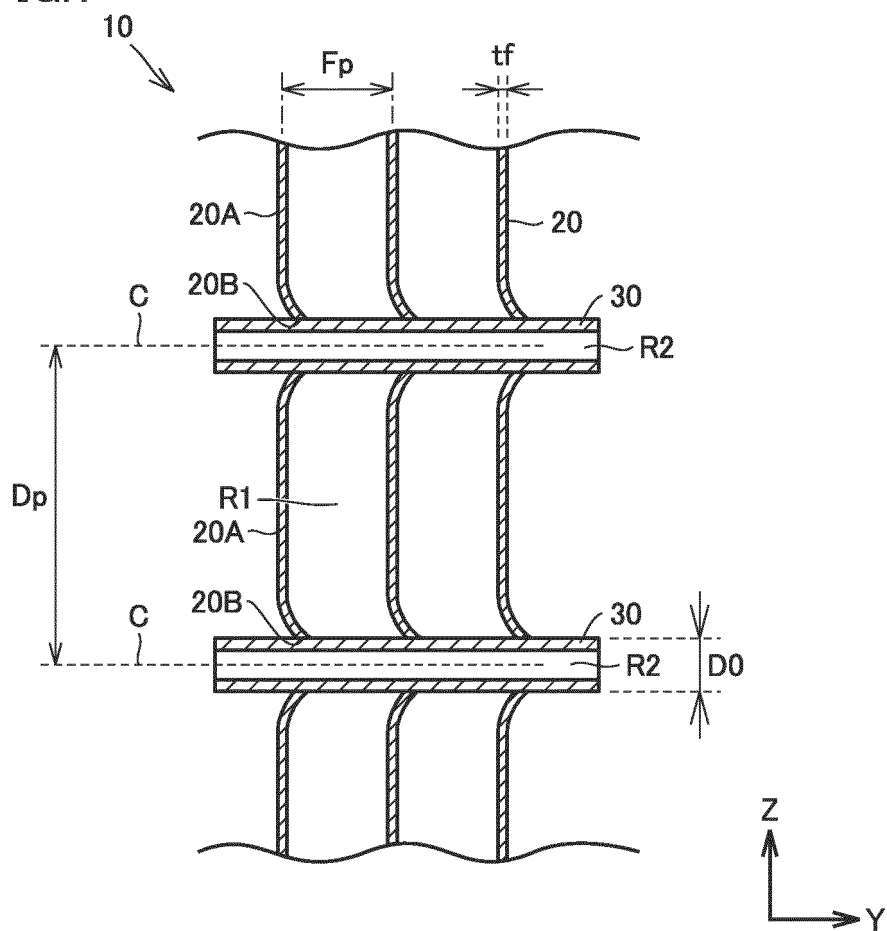


FIG.2

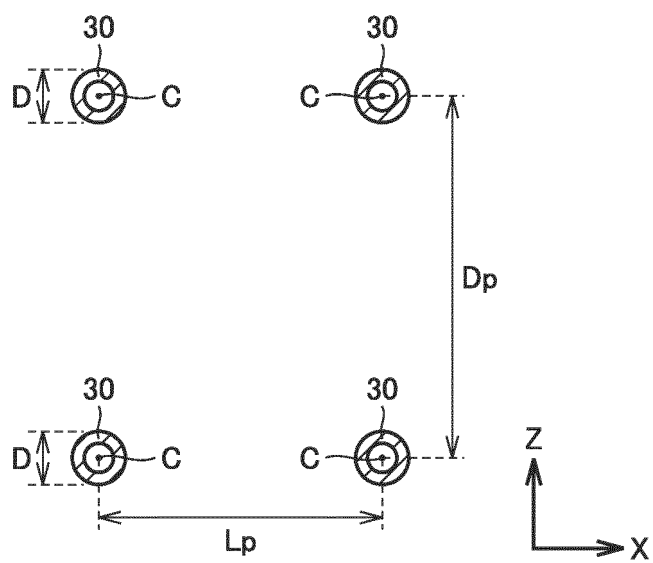


FIG.3

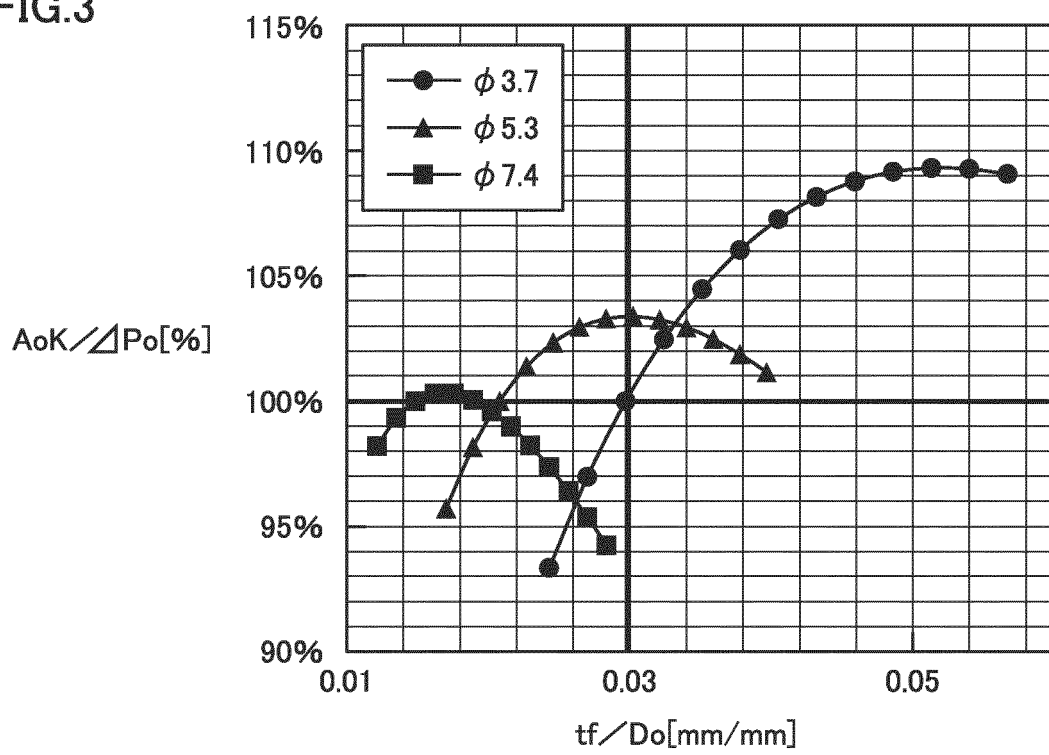


FIG.4

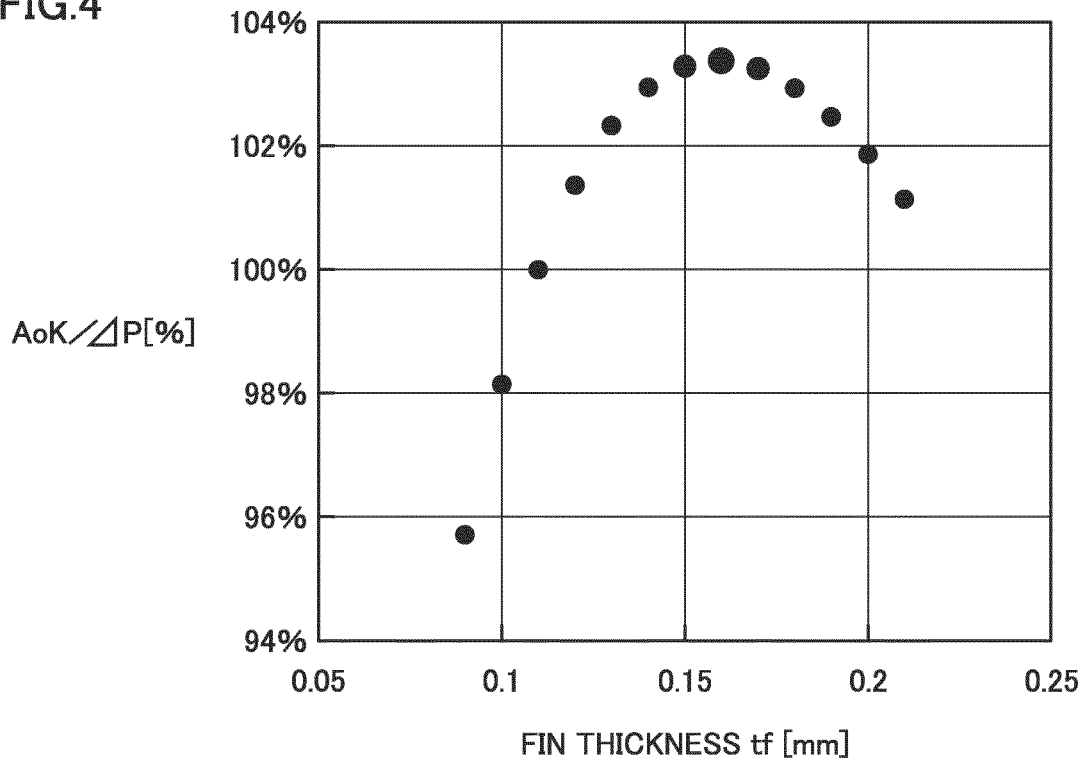


FIG.5

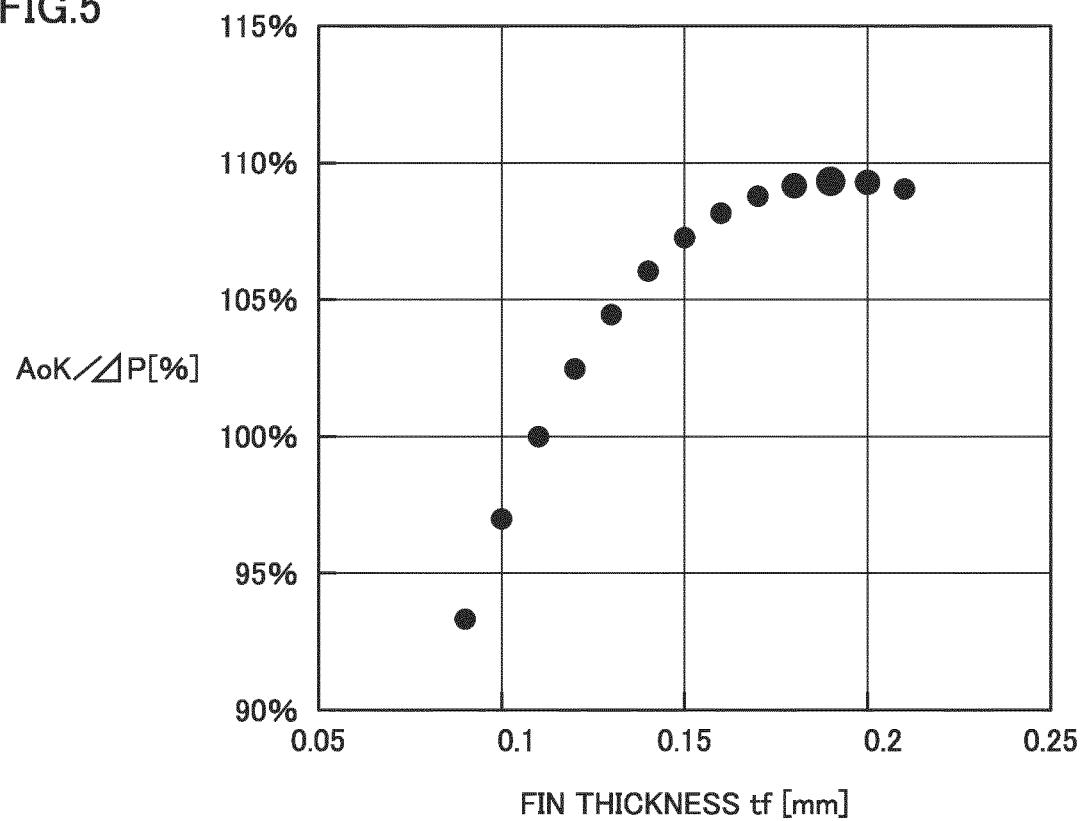


FIG.6

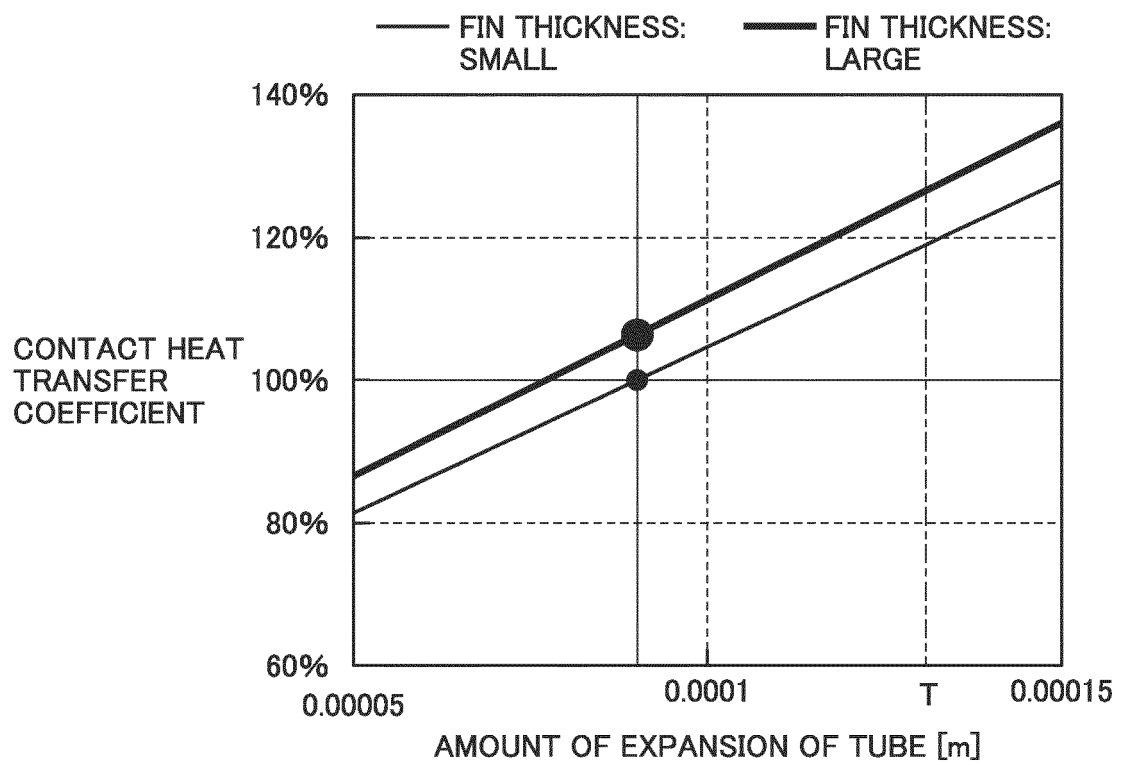


FIG.7

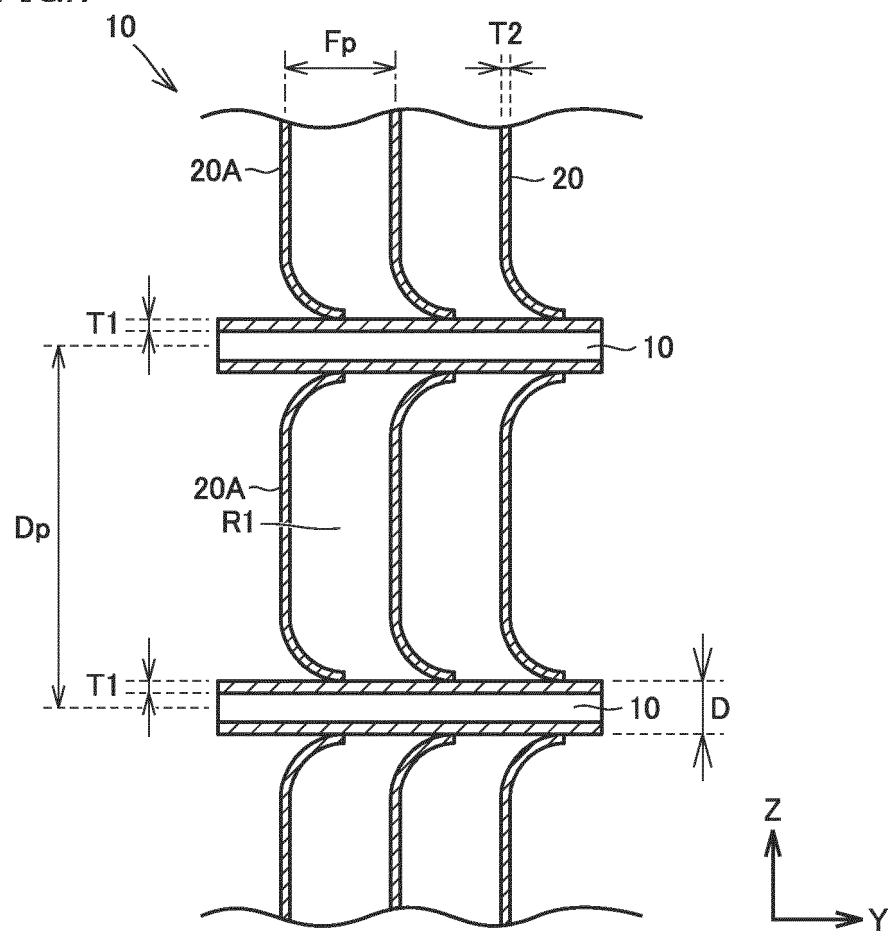
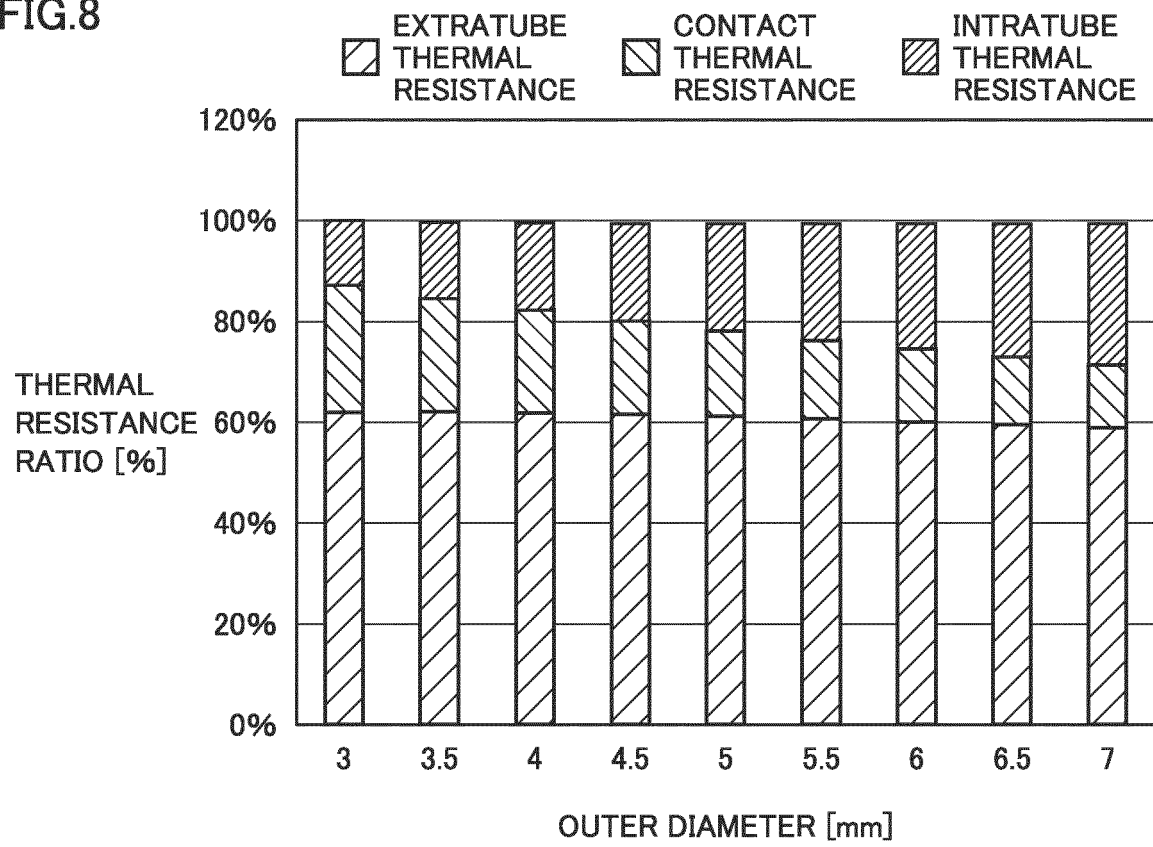


FIG.8



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2020/028293

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl. F28F1/32 (2006.01) i, F28F21/08 (2006.01) i
 FI: F28F1/32 F, F28F21/08 A, F28F21/08 E

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl. F28F1/32, F28F21/08

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Published examined utility model applications of Japan 1922-1996

Published unexamined utility model applications of Japan 1971-2020

Registered utility model specifications of Japan 1996-2020

Published registered utility model applications of Japan 1994-2020

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	WO 2014/147919 A1 (MITSUBISHI ELECTRIC CORP.) 25 September 2014, paragraphs [0001]-[0043], fig. 1-8	1-5
A	JP 2016-90122 A (PANASONIC INTELLECTUAL PROPERTY MANAGEMENT CO., LTD.) 23 May 2016, paragraphs [0001]-[0083], fig. 1-16	1-5
A	WO 2017/137956 A1 (THERMAX LIMITED) 17 August 2017, p. 1, line 2 to p. 9, line 29, fig. 1-5	1-5
A	WO 2007/108386 A1 (PANASONIC CORP.) 27 September 2007, paragraphs [0001]-[0120], fig. 1-38B	1-5



Further documents are listed in the continuation of Box C.



See patent family annex.

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Date of the actual completion of the international search
15.09.2020

Date of mailing of the international search report
29.09.2020

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2020/028293

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	EP 3617392 A1 (ELECTROLUX APPLIANCES AKTIEBOLAG) 04 March 2020, paragraphs [0001]–[0134], fig. 1–11	1–5

Form PCT/ISA/210 (continuation of second sheet) (January 2015)

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2020/028293

Patent Documents referred to in the Report	Publication Date	Patent Family	Publication Date
WO 2014/147919 A1	25.09.2014	US 2016/0025415 A1 paragraphs [0001]- [0106], fig. 1-8 GB 2527456 A CN 105143808 A	
JP 2016-90122 A	23.05.2016	US 2016/0123681 A1 paragraph [0001], fig. 1-16B CN 105571370 A	
WO 2017/137956 A1	17.08.2017	(Family: none)	
WO 2007/108386 A1	27.09.2007	US 2009/0199585 A1 paragraphs [0001]- [0205], fig. 1-38B EP 2006629 A2 CN 101405558 A	
EP 3617392 A1	04.03.2020	CN 110872775 A	

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2016020757 A [0002] [0003]