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(54) **FINNED TUBE HEAT EXCHANGER**

(57) A fin tube heat exchanger (100) includes fins (31) and heat transfer tubes (21). Each of the fins (31) has a flat portion (35), a first inclined portion (36), and a second inclined portion (38). A length of the fin (31) in a gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in a row direction is defined as S2, a diameter of the flat portion (35) is defined as D1, a flat plane passing through an upstream end and a downstream end of the fin (31) in the gas flow direction is defined as a reference plane H1, an angle between the reference plane H1 and the first inclined portion (36) is defined as $\theta 1$, an angle between the reference plane H1 and the second inclined portion (38) is defined as $\theta 2$, and a distance from the reference plane H1 to the flat portion (35) is defined as α. The fin tube heat exchanger (100) satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta 1 \pm 2\alpha)/(S2 - D1)\} \leq \theta 2 < 80° - \theta 1$.

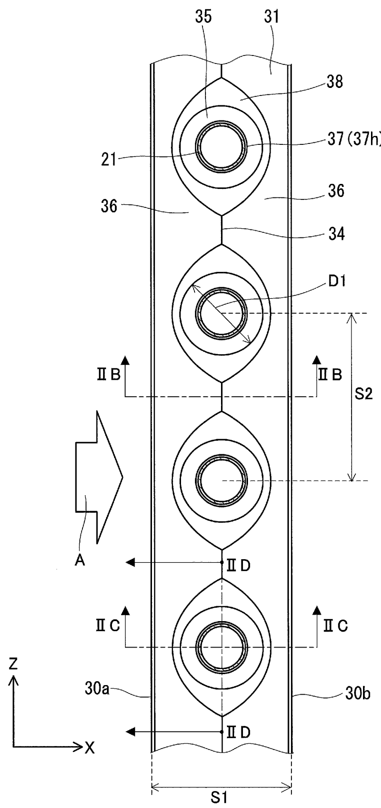

FIG.2A

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Description

TECHNICAL FIELD

[0001] The present invention relates to fin tube heat exchangers.

BACKGROUND ART

[0002] A fin tube heat exchanger includes a plurality of fins arranged at a predetermined spacing and heat transfer tubes penetrating the fins. Air flows between the fins and exchanges heat with a fluid in the heat transfer tubes.

[0003] FIG. 9A to FIG. 9D are a plan view of a fin used in a conventional fin tube heat exchanger, a cross-sectional view of the fins taken along the line IXB-IXB, a cross-sectional view thereof taken along the line IXC-IXC, and a cross-sectional view thereof taken along the line IXD-IXD, respectively. A fin 10 is formed to have a peak portion 4 and a valley portion 6 alternately in the air flow direction. This type of fin is commonly called a "corrugated fin". The corrugated fin is effective not only in increasing the heat transfer area but also in reducing the thickness of the thermal boundary layer by allowing the flow of air 3 to meander.

[0004] As shown in FIG. 10A to FIG. 10C, a technique of providing cut-and-raised portions in a corrugated fin to improve heat transfer performance is also known (Patent Literature 1). Cut-and-raised portions 41a, 41b, 41c and 41d are provided on inclined fin surfaces 42a, 42b, 42c and 42d of a fin 1. The heights H1, H2, H3 and H4 of the cut-and-raised portions 41a, 41b, 41c and 41d satisfy the relationship of $1/5 \cdot Fp \leq (H1, H2, H3, H4) \leq 1/3 \cdot Fp$, where Fp is the distance between adjacent fins 1.

[0005] Patent Literature 1 also describes another configuration of fins for minimizing airflow resistance during operation when frost is deposited on the fins. As shown in FIG. 11A to FIG. 11C, cut-and-raised portions 11a and 11b that satisfy the above relationship are provided on inclined fin surfaces 12a and 12b of the fin 1. The inclined fin surfaces 12a and 12b are inclined at a relatively small angle because the number of bends of the fin 1 is reduced.

CITATION LIST

Patent Literature

[0006] Patent Literature 1 JP 11(1999)-125495 A

SUMMARY OF INVENTION

Technical Problem

[0007] However, even if the height of the cut-and-raised portions is low enough, the cross-sectional area of the flow passage is reduced locally by 20% or more during operation when frost is deposited. Therefore, in the case where the cut-and-raised portions are provided, even if the number of bends is limited to one to reduce the inclination angle of the inclined surfaces, a significant increase in the airflow resistance is inevitable. In order to reduce the airflow resistance of the fins 1 shown in FIG. 11A to FIG. 11C to the same level of the fins 10 shown in FIG. 9A to FIG. 9D, the inclination angle of the fins 1 needs to be minimized as much as possible.

[0008] It is an object of the present invention to provide a fin tube heat exchanger exhibiting high basic performance during operation, regardless of whether frost is deposited or not.

Solution to Problem

[0009] The present disclosure provides a fin tube heat exchanger including: a plurality of fins arranged in parallel to form flow passages for a gas; and heat transfer tubes penetrating the fins and configured to allow a medium to flow through the heat transfer tubes to exchange heat with the gas. In this heat exchanger, each of the fins is a corrugated fin formed to have only one peak portion in a gas flow direction, and has a plurality of through holes through which the heat transfer tubes are fitted, a flat portion formed around each of the through holes, a first inclined portion inclined with respect to the gas flow direction to form the peak portion, and a second inclined portion connecting the flat portion and the first inclined portion. The plurality of through holes are formed in a row direction perpendicular to both the gas flow direction and a direction in which the fins are arranged. When a length of the fin in the gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in the row direction is defined as S2, a diameter of the flat portion is defined as D1, a flat plane passing through an upstream end and a downstream end of the fin in the gas flow direction is defined as a reference plane, an angle between the reference plane and the first inclined portion is defined as $\theta 1$, an angle between the reference plane and the second inclined portion is defined as $\theta 2$, and a distance from the reference plane to the flat portion is defined as α, the fin tube heat exchanger satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta 1 \pm 2\alpha)/(S2 - D1)\} \leq \theta 2 < 80° - \theta 1$.

Advantageous Effects of Invention

[0010] In the fin tube heat exchanger configured as above, the airflow resistance can be reduced sufficiently and the amount of heat exchange (heat exchange capacity) can be increased.

BRIEF DESCRIPTION OF DRAWINGS

[0011]

FIG. 1 is a perspective view of a fin tube heat exchanger according to an embodiment of the present invention.
FIG. 2A is a plan view of a fin used in the fin tube heat exchanger of FIG. 1.
FIG. 2B is a cross-sectional view of the fins shown in FIG. 2A, taken along the line IIB-IIB.
FIG. 2C is a cross-sectional view of the fins shown in FIG. 2A, taken along the line IIC-IIC.
FIG. 2D is a cross-sectional view of the fins shown in FIG. 2A, taken along the line IID-IID.
FIG. 3A is a graph showing a relationship between a second inclination angle $\theta 2$ and a surface area of a fin.
FIG. 3B is a graph showing a relationship between the second inclination angle $\theta 2$ and a surface area ratio (surface area of a V-shaped corrugated fin / surface area of an M-shaped corrugated fin).
FIG. 4A is a schematic view showing adjacent second inclined portions in contact with each other.
FIG. 4B is a schematic view showing a method for calculating a threshold angle $\theta 2L$.
FIG. 4C is a schematic view showing a method for calculating a maximum value αmax of a distance α.
FIG. 5A is a plan view showing a portion having a high heat transfer coefficient in the fin shown in FIG. 2A.
FIG. 5B is a plan view showing a portion having a high heat transfer coefficient in a conventional fin.
FIG. 6A is a cross-sectional view showing a section for air flow analysis.
FIG. 6B is a schematic view showing an air flow when the sum of a first inclination angle $\theta 1$ and a second inclination angle $\theta 2$ is 36°.
FIG. 6C is a schematic view showing an air flow when the sum of the first inclination angle $\theta 1$ and the second inclination angle $\theta 2$ is 66°.
FIG. 6D is a schematic view showing an air flow when the sum of the first inclination angle $\theta 1$ and the second inclination angle $\theta 2$ is 76°.
FIG. 6E is a schematic view showing an air flow when the sum of the first inclination angle $\theta 1$ and the second inclination angle $\theta 2$ is 86°.
FIG. 6F is a schematic view showing an air flow when the sum of the first inclination angle $\theta 1$ and the second inclination angle $\theta 2$ is 96°.
FIG. 7 is a graph showing a relationship between the second inclination angle $\theta 2$ and the performance (the amount of heat exchange and pressure loss) of the fin tube heat exchanger.
FIG. 8A is a plan view of a fin according to a second embodiment.
FIG. 8B is a cross-sectional view of the fins shown in FIG. 8A, taken along the line VIIIB-VIIIB.
FIG. 8C is a cross-sectional view of the fins shown in FIG. 8A, taken along the line VIIIC-VIIIC.
FIG. 8D is a cross-sectional view of the fins shown in FIG. 8A, taken along the line VIIID-VIIID.
FIG. 8E is a schematic view showing a method for calculating a threshold angle $\theta 2L$.
FIG. 9A is a plan view of a fin used in a conventional fin tube heat exchanger.
FIG. 9B is a cross-sectional view of the fins shown in FIG. 9A, taken along the line IXB-IXB.
FIG. 9C is a cross-sectional view of the fins shown in FIG. 9A, taken along the line IXC-IXC.
FIG. 9D is a cross-sectional view of the fins shown in FIG. 9A, taken along the line IXD-IXD.
FIG. 10A is a plan view of another fin used in a conventional fin tube heat exchanger.
FIG. 10B is a cross-sectional view of the fins shown in FIG. 10A, taken along the line XB-XB.
FIG. 10C is a cross-sectional view of the fins shown in FIG. 10A, taken along the line XC-XC.
FIG. 11A is a plan view of still another fin used in a conventional fin tube heat exchanger.
FIG. 11B is a cross-sectional view of the fins shown in FIG. 11A, taken along the line XIB-XIB.
FIG. 11C is a cross-sectional view of the fins shown in FIG. 11A, taken along the line XIC-XIC.

DESCRIPTION OF EMBODIMENTS

[0012] A first aspect of the present disclosure provides a fin tube heat exchanger including: a plurality of fins arranged in parallel to form flow passages for a gas; and heat transfer tubes penetrating the fins and configured to allow a medium to flow through the heat transfer tubes to exchange heat with the gas. In this heat exchanger, each of the fins is a corrugated fin formed to have only one peak portion in a gas flow direction, and has a plurality of through holes through which the heat transfer tubes are fitted, a flat portion formed around each of the through holes, a first inclined portion inclined with respect to the gas flow direction to form the peak portion, and a second inclined portion connecting the flat portion and the first inclined portion. The plurality of through holes are formed in a row direction perpendicular to both the gas flow direction and a direction in which the fins are arranged. When a length of the fin in the gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in the row direction is defined as S2, a diameter of the flat portion is defined as D1, a flat plane passing through an upstream end and a downstream end of the fin in the gas flow direction is defined as a reference plane, an angle between the reference plane and the first inclined portion is defined as $\theta 1$, an angle between the reference plane and the second inclined portion is defined as $\theta 2$, and a distance from the reference plane to the flat portion is defined as α, the fin tube heat exchanger satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta 1 \pm 2\alpha)/(S2 - D1)\} \leq \theta 2 < 80° - \theta 1$.

[0013] A second aspect of the present disclosure provides the fin tube heat exchanger as set forth in the first aspect, wherein the angle $\theta 2$ satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta 1 \pm 2\alpha)/(S2 - D1)\} \leq \theta 2 < 70° - \theta 1$.

[0014] A third aspect of the present disclosure provides the fin tube heat exchanger as set forth in the first or second aspect, wherein the fin is configured to prevent the gas from flowing from a front side to a back side of the fin in a region other than the plurality of through holes.

[0015] A fourth aspect of the present disclosure provides a fin tube heat exchanger including: a plurality of fins arranged in parallel to form flow passages for a gas; heat transfer tubes penetrating the fins and configured to allow a medium to flow through the heat transfer tubes to exchange heat with the gas. In this heat exchanger, each of the fins is a corrugated fin formed to have only one peak portion in a gas flow direction, and has a plurality of through holes through which the heat transfer tubes are fitted, a cylindrical fin collar being in close contact with the heat transfer tube around each of the through holes, a first inclined portion inclined with respect to the gas flow direction to form the peak portion, and a second inclined portion connecting the fin collar and the first inclined portion. The plurality of through holes are formed in a row direction perpendicular to both the gas flow direction and a direction in which the fins are arranged. When a length of the fin in the gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in the row direction is defined as S2, an outer diameter of the fin collar is defined as D2, a flat plane passing through an upstream end and a downstream end of the fin in the gas flow direction is defined as a reference plane, an angle between the reference plane and the first inclined portion is defined as $\theta 1$, and an angle between the reference plane and the second inclined portion is defined as $\theta 2$, the fin tube heat exchanger satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta 1)/(S2 - D2)\} \leq \theta 2 < 80° - \theta 1$.

[0016] A fifth aspect of the present disclosure provides the fin tube heat exchanger as set forth in the fourth aspect, wherein the angle $\theta 2$ satisfies a relationship: $\tan^{-1}(S1 \cdot \tan\theta 1)/(S2 - D2)\} \leq \theta 2 < 70° - \theta 1$.

[0017] A sixth aspect of the present disclosure provides the fin tube heat exchanger as set forth in the fourth or fifth aspect, wherein the fin is configured to prevent the gas from flowing from a front side to a back side of the fin in a region other than the plurality of through holes.

[0018] Hereinafter, the embodiments of the present invention are described with reference to the drawings. The present invention is not limited by the following embodiments.

(First Embodiment)

[0019] As shown in Fig. 1, a fin tube heat exchanger 100 of the present embodiment includes a plurality of fins 31 arranged in parallel to form flow passages for air A (gas) and heat transfer tubes 21 penetrating these fins 31. The fin tube heat exchanger 100 is configured to exchange heat between a medium B flowing in the heat transfer tubes 21 and the air A flowing along the surfaces of the fins 31. The medium B is, for example, a refrigerant such as carbon dioxide or hydrofluorocarbon. One continuous heat transfer tube 21 may be used, or a plurality of separate transfer tubes 21 may be used.

[0020] The fins 31 each has a leading edge 30a and a trailing edge 30b. The leading edge 30a and the trailing edge 30b are each linear. In the present embodiment, the fin 31 has a left and right symmetrical configuration with respect to the center of the heat transfer tube 21. Therefore, there is no need to consider the orientation of the fin 31 when the heat exchanger 100 is assembled.

[0021] In this description, a direction in which the fins 31 are arranged is defined as a height direction, a direction parallel to the leading edge 30a is defined as a row direction, and a direction perpendicular to the height direction and the row direction is defined as an air flow direction (a direction in which the air A flows). In other words, the row direction is a direction perpendicular to both the height direction and the air flow direction. The air flow direction is perpendicular to the longitudinal direction of the fin 31. The air flow direction, the height direction and the row direction correspond to X direction, Y direction and Z direction, respectively.

[0022] As shown in FIG. 2A to FIG. 2D, the fin 31 typically has a rectangular and flat plate shape. The longitudinal direction of the fin 31 coincides with the row direction. In the present embodiment, the fins 31 are arranged at a constant spacing (fin pitch FP). However, the spacing between two fins 31 adjacent to each other in the height direction does not necessarily have to be constant and may vary. The fin pitch FP is adjusted to, for example, a range of 1.0 to 1.5 mm. As shown in FIG. 2B, the fin pitch FP is represented by the distance between two adjacent fins 31.

[0023] A constant width portion including the leading edge 30a and a constant width portion including the trailing edge 30b are parallel to the air flow direction. These portions are used to fix the fin 31 to a mold to form the fin 31, and do not significantly affect the performance of the fin 31.

[0024] As the material of the fins 31, a punched-out aluminum flat plate with a thickness of 0.05 to 0.8 mm can be used suitably. The surface of the fin 31 may be subjected to hydrophilic treatment such as boehmite treatment or coating with a hydrophilic paint. Water repellent treatment may be applied instead of hydrophilic treatment.

[0025] The fin 31 is provided with a plurality of through holes 37h formed in a row at equal distances along the row direction. A straight line passing through the center of each of the through holes 37h is parallel to the row direction. The heat transfer tube 21 is fitted through each of the through holes 37h. A part of the fin 31 forms a cylindrical fin collar 37 around the through hole 37h, and this fin collar 37 is in close contact with the heat transfer tube 21. The diameter of the through hole 37h is, for example, 1 to 20 mm, and it may be 4 mm or less. The diameter of the through hole 37h is the same as the outer diameter of the heat transfer tube 21. The center-to-center distance (tube pitch) between two through holes 37h adjacent to each other in the row direction is, for example, 2 to 3 times larger than the diameter of the through hole 37h. The length of the fin 31 in the air flow direction is, for example, 15 to 25 mm.

[0026] As shown in FIG. 2A and FIG. 2B, the fin 31 is formed to have only one peak portion 34 in the air flow direction. The ridge line of the peak portion 34 is parallel to the row direction. That is, the fin 31 is a so-called corrugated fin. When a portion of the fin 31 protruding in the same direction as the protruding direction of the fin collar 37 is defined as a "peak portion 34", the fin 31 has only one peak portion 34 in the air flow direction in the present embodiment. The leading edge 30a and the trailing edge 30b correspond to the valley portions. In the air flow direction, the position of the peak portion 34 coincides with the center of the heat transfer tube 21.

[0027] In the present embodiment, the fin 31 is configured to prevent the air A from flowing from the front side (upper surface side) of the fine 31 to the back side (lower surface side) thereof in the region other than the through holes 37h. Thus, it is desirable that no opening other than the through holes 37h be provided in the fin 31. Without the opening, no clogging occurs when frost is deposited, and thus it is advantageous in terms of pressure loss. The phrase "no opening is provided" means that a slit, a louver, or the like is not provided, that is, a hole penetrating the fin is not provided.

[0028] The fin 31 further has flat portions 35, first inclined portions 36, and second inclined portions 38. The flat portion 35 is an annular portion formed adjacent to the fin collar 37 around the through hole 37h. The surface of the flat portion 35 is parallel to the air flow direction and perpendicular to the height direction. The first inclined portion 36 is a portion inclined with respect to the air flow direction to form the peak portion 34. The first inclined portion 36 has the largest area in the fin 31. The surface of the first inclined portion 38 is flat. The first inclined portion 36 is located on both sides of a reference line parallel to the row direction and passing through the center of the heat transfer tube 21. That is, the peak portion 34 is formed by the first inclined portion 36 located on the windward side and the first inclined portion 36 located on the leeward side. The second inclined portion 38 is a portion that connects the flat portion 35 smoothly to the first inclined portion 36 so as to eliminate the level difference between the flat portion 35 and the first inclined portion 36. The surface of the second inclined portion 38 is a gently curved surface. The flat portion 35 and the second inclined portion 39 form a concave portion around the fin collar 37 and the through hole 37h.

[0029] The edge portion between the first inclined portion 36 and the second inclined portion 38 may be rounded with an appropriate radius of curvature (for example, R = 0.5 mm to 2.0 mm). Likewise, the edge portion between the peak portion 34 and the second inclined portion 38 may be rounded with an appropriate radius of curvature (for example, R = 0.5 mm to 2.0 mm). The edge portions thus rounded improve the drainage performance of the fin 31.

[0030] As shown in FIG. 2A to FIG. 2D, the length of the fin 31 in the air flow direction is defined as S1. The center-to-center distance (tube pitch) between the heat transfer tubes 21 in the row direction is defined as S2. The diameter of the flat portion 35 is defined as D1. The flat plane passing through the upstream end and the downstream end of the fin 31 in the air flow direction is defined as a reference plane H1. The upstream end and the downstream end of the fin 31 correspond to the leading edge 30a and the trailing edge 30b respectively. The angle between the reference plane H1 and the first inclined portion 36 is defined as $\theta 1$. The angle between the reference plane H1 and the second inclined portion 38 is defined as $\theta 2$. The angle $\theta 1$ is an acute angle between the reference plane H1 and the first inclined portion 36. Likewise, the angle $\theta 2$ is an acute angle between the reference plane H1 and the second inclined portion 38. In this description, the angle $\theta 1$ and the angle $\theta 2$ are referred to as a "first inclination angle $\theta 1$" and a "second inclination angle $\theta 2$" respectively. The distance from the reference plane H1 to the flat portion 35 is defined as α. In the embodiment shown in FIG. 2A to FIG. 2D, the distance α is zero. That is, the flat portion 35, the leading edge 30a, and the trailing edge 30b are on the same level in the height direction. In this case, the reference plane H1 coincides with the plane including the surface of the flat portion 35.

[0031] When S1, S2, D1, $\theta1$, $\theta2$, and α are defined as described above, the fin tube heat exchanger 100 satisfies the following formula (1):

$$\tan^{-1}\{(S1 \cdot \tan\theta1 \pm 2\alpha)/(S2 - D1)\} \leq \theta2 < 80^\circ - \theta1 \ldots (1)$$

[0032] The flat portion 35 may be on a different level from the leading edge 30a and the trailing edge 30b in the height direction. Specifically, when the flat portion 35 is located closer to the ridge of the peak portion 34 than the reference plane H1, the left-hand side of the formula (1) is $\tan^{-1}\{(S1 \cdot \tan\theta1 - 2\alpha)/(S2 - D1)\}$. Since the angle between the first inclined portion 36 and the second inclined portion 38 increases when the flat portion 35 is located closer to the ridge of the peak portion 34 than the reference plane H1, the pressure loss decreases while the surface area of the fin 31 decreases. Thus, the fin 31 with low pressure loss is obtained.

[0033] On the other hand, when the flat portion 35 is located farther from the ridge of the peak portion 34 than the reference plane H1, the left-hand side of the formula (1) is $\tan^{-1}\{(S1 \cdot \tan\theta1 + 2\alpha)/(S2 - D1)\}$. Since the angle between the first inclined portion 36 and the second inclined portion 38 decreases when the flat portion 35 is located farther from the ridge of the peak portion 34 than the reference plane H1, the surface area of the fin 31 increases while the pressure loss increases. In addition, the increase in the angle $\theta2$ of the second inclined portion 38 is expected to have the effect of reducing a dead water region formed behind the heat transfer tube 21. Thus, the fin 31 with high heat exchange capacity is obtained.

[0034] The second inclined portion 38 is a curved surface as a whole, but the second inclination angle $\theta2$ can be determined in the cross section shown in FIG. 2C or FIG. 2D. The cross section of FIG. 2C is a cross section of the fins 31 taken along the plane perpendicular to the row direction and passing through the center of the heat transfer tube 21. The cross section of FIG. 2D is a cross section of the fins 31 taken along the plane perpendicular to the flow direction and passing through the center of the heat transfer tube.

[0035] The technical significance of the formula (1) is described below in detail.

(Regarding Lower Limit of Second Inclination Angle $\theta2$)

[0036] Assuming that the length of a fin in the air flow direction is fixed, a corrugated fin always has a larger surface area than a flat fin (unbent fin). Furthermore, when the first inclination angle $\theta1$ is fixed, a corrugated fin with only one bend (V-shaped corrugated fin) has a larger surface area than a corrugated fin with two or more bends (M-shaped corrugated fin). The reason for this can be understood by comparing the cross section of the fin 31 of the present embodiment with that of the conventional fin 10.

[0037] As can be understood by comparing FIG. 2B and FIG. 9B, the length of the contour of the cross section shown in FIG. 2B is equal to the length of the contour of the cross section shown in FIG. 9B. Since the cross section shown in FIG. 2C matches the cross section shown in FIG. 9C, the lengths of the contours of these cross sections are equal. In contrast, as can be understood by comparing FIG. 2D and FIG. 9D, the length of the contour of the cross section shown in FIG. 2D is greater than that of the contour of the cross section shown in FIG. 9D. This is because the cross section of the fin 31 of the present embodiment shown in FIG. 2D includes the second inclined portion 38 inclined at the second inclination angle $\theta2$. The cross section of the conventional fin 10 shown in FIG. 9D does not include the inclined portion 8 but includes only the flat portion 5 and the valley portion 6. The surface area of the fin 31 of the present embodiment is larger than that of the conventional fin 10 with two bends because the surface area of the fin 31 is increased by the presence of the second inclined portion 38.

[0038] In order to prove the above fact, the surface area of a V-shaped corrugated fin and the surface area of an M-shaped corrugated fin are calculated respectively, with the second inclination angle $\theta2$ varying. FIG. 3A and FIG. 3B show the results. The other conditions used for the calculations are as follows.

[0039]

*Fin length S1 = 18. 9 mm
*Center-to-center distance between heat transfer tubes S2 = 18.3 mm
*Diameter of flat portion D = 11 mm
*First inclination angle $\theta1$ = 16°
*Fin pitch FP = 1.3 mm

[0040] As shown in FIG. 3A, the surface area of the fin increases as the second inclination angle $\theta2$ increases, independently of the number of bends. However, the rate of the increase in the surface area of the V-shaped corrugated fin to the increase in the second inclination angle $\theta2$ is higher than that of the M-shaped corrugated fin. As shown in

FIG. 3B, when the second inclination angle $\theta 2$ is almost 0°, the surface area of the V-shaped corrugated fin is almost equal to the surface area of the M-shaped corrugated fin. That is, the ratio of these surface areas is about 100%. The larger the second inclination angle $\theta 2$, the greater the difference between the surface areas.

[0041] A detailed analysis shows that when the second inclination angle $\theta 2$ decreases from 80° to 40°, the slope of the curve representing the ratio of the surface areas gradually decreases. However, the slope of the curve drops sharply near the point A shown in FIG. 3B. As shown in FIG. 4A, the threshold angle $\theta 2L$ corresponding to this point A is an angle at which the second inclined portions 38 adjacent in the row direction contact each other in the V-shaped corrugated fin. The invasion between the adjacent second inclined portions 38 proceeds as the second inclination angle $\theta 2$ becomes smaller than the threshold angle $\theta 2L$. Thus, the decrease in the ratio of the surface areas is accelerated. Here, the threshold angle $\theta 2L$ is represented by the following formula (2) using the length of the fin 31 (S1), the center-to-center distance between the heat transfer tubes 21 (S2), the diameter of the flat portion 35 (D1), the first inclination angle ($\theta 1$), and the distance (α):

$$\theta 2L = \tan^{-1}\{(S1 \cdot \tan\theta 1 \pm 2\alpha)/(S2 - D1)\} \ldots (2)$$

[0042] The threshold angle $\theta 2L$ is the angle calculated by the following method. As shown in FIG. 4B, the height of the peak portion 34 is represented by $(S1/2) \cdot \tan\theta 1 \pm \alpha$. The tangent of the second inclination angle $\theta 2$ when the adjacent second inclined portions 38 just contact each other (= threshold angle $\theta 2L$) is represented by $\{(S1/2) \cdot \tan\theta 1 \pm \alpha\}/\{(S2 - D1)/2\}$. Therefore, the threshold angle $\theta 2L$ can be represented by the formula (2).

[0043] When the second inclination angle $\theta 2$ is smaller than the threshold angle $\theta 2L$, the adjacent second inclined portions 38 invade each other and the peak portion 34 disappears. Thus, the contact region between the adjacent second inclined portions 38 becomes almost parallel to the horizontal plane. When air passes over the horizontal plane in the contact region, the air flow is slowed down, which causes a decrease in the heat transfer coefficient. Therefore, when the second inclination angle $\theta 2$ is smaller than the threshold angle $\theta 2L$, not only the heat exchange capacity decreases due to a sudden decrease in the surface area, but also it further decreases due to a decrease in the heat transfer coefficient. As a result, the heat exchange capacity of the fin tube heat exchanger decreases significantly.

[0044] Hence, in order to enhance the heat exchange capacity of the fin tube heat exchanger, it is important for the second inclination angle $\theta 2$ to be equal to or larger than the threshold angle $\theta 2L$.

[0045] Another reason why the use of the fin 31 having only one peak portion 34 is expected to improve the heat exchange capacity is an increase in the average heat transfer coefficient. FIG. 5A shows the result obtained by numerical analysis of a V-shaped corrugated fin having only one peak portion. FIG. 5B shows the result obtained by numerical analysis of an M-shaped corrugated fin having two peak portions. High heat flux (large amount of heat exchange) regions are marked with thick lines. As shown in FIG. 5A, the heat flux is very high at the leading edge 30a and the peak portion 34. Likewise, as shown in FIG. 5B, the heat flux is very high at the leading edge 9 and the peak portion 4. However, a comparison of the total length of the thick lines shown in FIG. 5A and that shown in FIG. 5B shows that the former is longer than the latter. This means that the V-shaped corrugated fin can have a longer high heat flux region. Therefore, the fin 31 of the present embodiment is advantageous over the conventional fin 10 in terms of the heat transfer coefficient.

(Regarding Upper Limit of Second Inclination Angle $\theta 2$)

[0046] The disadvantage of the increase in the second inclination angle $\theta 2$ is "flow separation". As shown by a dashed line D in FIG. 6A, in the fin tube heat exchanger 100, a section where the flow of the air A is turned at the largest angle is present near the boundary between the first inclined portion 36 and the second inclined portion 38. The turning angle of the air flow in the section shown by the dashed line D can be represented by the sum ($\theta 1 + \theta 2$) of the first inclination angle $\theta 1$ and the second inclination angle $\theta 2$.

[0047] In order to examine the influence of the turning angle ($\theta 1 + \theta 2$) on the air flow, air flow analysis was performed using a model of a corrugated fin having the conditions used for the calculation of the surface area. Specifically, the size of the separation region in the turning section and the air flow direction in the separation region were examined with the turning angle ($\theta 1 + \theta 2$) varying. The face velocity was 1.3 m/sec. FIG. 6B to FIG. 6F show representative results.

[0048] As shown in FIG. 6B, when the turning angle ($\theta 1 + \theta 2$) is 36°, a separation region was formed near the outer corner wall of the turning section. However, the separation region was very thin, and the air flowed in the forward direction in the region along the main flow. As shown in FIG. 6C, when the turning angle ($\theta 1 + \theta 2$) was 66°, a separation region was formed near the outer corner wall of the turning section. The separation region was relatively thick, but the air flowed substantially in the forward direction in that separation region. A fraction of the air created a flow with a vector different from that of the main flow. When the turning angle ($\theta 1 + \theta 2$) was 76°, a fraction of the air created a flow with a vector different from that of the main flow in the same manner as in the case of the turning angle ($\theta 1 + \theta 2$) of 66°. When the turning angle ($\theta 1 + \theta 2$) was 86°, a flow with a vector different from that of the main flow clearly increased. When the turning angle ($\theta 1 + \theta 2$) was 96°, a wide and very thick separation region was formed near the outer corner wall of the turning section. Furthermore, most of the flow in the separation region was a turbulent flow including a flow with a vector opposite to that of the main flow. The turbulent flow in the separation region not only causes a significant increase in the airflow resistance but also results in a decrease in the effective heat transfer area. That is, when the turning angle ($\theta 1 + \theta 2$) is too large, the increase in the amount of heat exchange due to the increase in the surface area may be cancelled out. Therefore, it is desirable that the turning angle ($\theta 1 + \theta 2$) be within the range that can avoid a significant increase in the airflow resistance.

[0049] In the above analysis results, when the turning angle ($\theta 1 + \theta 2$) was 76°, a fraction of the air created a flow with a vector different from that of the main flow of the air. In contrast, when the turning angle ($\theta 1 + \theta 2$) was 86°, a flow with a vector different from that of the main flow clearly increased. This means that the creation of a turbulent flow in the separation region can be suppressed, and as a result, the increase in the airflow resistance can be suppressed, by limiting the turning angle ($\theta 1 + \theta 2$) to less than 80°, and preferably to less than 70°.

[0050] From the above results, a preferred range of the second inclination angle $\theta 2$ is represented by the above formula (1).

[0051] The first inclination angle $\theta 1$ is not particularly limited, but preferably it is less than 40°. When the first inclination angle $\theta 1$ is 40° or more, the bending angle of the peak portion 34 is 80° or more. In this case, a thick separation region is formed near the peak portion 34, and thus a turbulent flow including a flow with a vector opposite to that of the main flow may be created. Therefore, it is preferable that the first inclination angle $\theta 1$ be less than 40°. The lower limit of the first inclination angle $\theta 1$ is not particularly limited. In the corrugated fin, the first inclination angle $\theta 1$ is larger than 0°.

[0052] FIG. 7 is a graph showing the relationship between the second inclination angle $\theta 2$ and the performance (the amount of heat exchange and the pressure loss) of the fin tube heat exchanger. The rate of change in the amount of heat exchange changes significantly when the second inclination angle $\theta 2$ becomes smaller the threshold angle $\theta 2L$. That is, when the second inclination angle $\theta 2$ is equal to or larger than the threshold angle $\theta 2L$, a sufficient amount of heat exchange can be obtained. On the other hand, the rate of change in the airflow resistance changes significantly when the second inclination angle $\theta 2$ becomes larger than an angle $\theta 2H$ (= 80° - $\theta 1$ or 70° - $\theta 1$). That is, when the second inclination angle $\theta 2$ is smaller than the angle $\theta 2H$, the airflow resistance can be suppressed sufficiently.

[0053] The upper limit and the lower limit of the distance α in the formula (1) are discussed. As can be understood from FIG. 4B, the value of α in the distance ($(S1/2) \cdot \tan\theta 1 - \alpha$) between the flat portion 35 and the ridge of the peak portion 34 gradually increases as the flat portion 35 gradually approaches the ridge of the peak portion 34. When the flat portion 35 further approaches the ridge of the peak portion 34, it becomes necessary, at one point, to provide a step between the flat portion 35 and the first inclined portion 36. Such a step significantly impedes the air flow around the flat portion 35 and thus significantly increases the airflow resistance. As can be understood from FIG. 4C, the maximum α value θmax, at which no such step is required, is represented by $\tan\theta 1 \cdot (S1 - D1)/2$.

[0054] On the other hand, the value of α in the distance ($(S1/2) \cdot \tan\theta 1 + \alpha$) between the flat portion 35 and the ridge of the peak portion 34 gradually increases as the flat portion 35 gradually recedes from the ridge of the peak portion 34. In this case, as can be understood from the formula (2), the threshold angle $\theta 2L$ increases as the value of α increases. However, no step is seen because of the configuration of the fin. Therefore, the value of α is not limited as long as it is within the range ($\theta 2 < 80° - \theta 1$ or $\theta 2 < 70° - \theta 1$) that can avoid the occurrence of a clear turbulent flow in the separation region.

(Second Embodiment)

[0055] As shown in FIG. 8A to FIG. 8D, a fin 41 of the present embodiment has the same configuration as the fin 31 of the first embodiment, except that the flat portion 35 is not provided around the fin collar 37. The common components of the fin 41 of the present embodiment and the fin 31 of the first embodiment are designated by the same reference numerals, and the description thereof is omitted.

[0056] The fin 41 has the fin collars 37, the first inclined portions 36, and the second inclined portions 38. The fin collar 37 is a cylindrical portion being in close contact with the heat transfer tube 21 around the through hole 37h. The second inclined portion 38 is a portion connecting the fin collar 37 and the first inclined portion 36. When the outer diameter of the fin collar 37 is defined as D2, the fin 41 (specifically, the fin tube heat exchanger 100) satisfies the following formula (3):

$$\tan^{-1}\{(S1 \cdot \tan\theta 1)/(S2 - D2)\} \leq \theta 2 < 80^{\circ} - \theta 1 \ldots (3)$$

[0057] In the present embodiment, the lower end of the fin collar 37 is on the same level as the reference plane H1, and its level does not vary unlike the flat portion 35 of the first embodiment. As shown in FIG. 8E, the height of the peak portion 34 is represented by $(S1 \cdot \tan\theta1)/2$. Since the fin 41 does not have the flat portion 35, when the adjacent second inclined portions 38 contact each other in the row direction, the length of the second inclined portion 38 in the row direction is represented by $(S2 - D2)/2$. As inferred from the results of the air flow analysis shown in FIG. 6A to FIG. 6F, the presence or absence of the flat portion 35 is considered to have no significant influence on the increase or decrease in the airflow resistance. For these reasons, all the descriptions of the formula (1) apply to the formula (3). The fin tube heat exchanger 100 including the fins 41 has low airflow resistance and high heat exchange capacity when it satisfies the formula (3). It is desirable that the second inclined angle $\theta2$ be less than $(70° - \theta1)$, as in the case of the first embodiment.

INDUSTRIAL APPLICABILITY

[0058] The fin tube heat exchanger of the present invention is useful for heat pumps used in air conditioners, water heaters, heating apparatuses, etc. In particular, it is useful for evaporators for evaporating a refrigerant.

Claims

1. A fin tube heat exchanger comprising:

a plurality of fins arranged in parallel to form flow passages for a gas; and
heat transfer tubes penetrating the fins and configured to allow a medium to flow through the heat transfer tubes to exchange heat with the gas,
wherein each of the fins is a corrugated fin formed to have only one peak portion in a gas flow direction, and has a plurality of through holes through which the heat transfer tubes are fitted, a flat portion formed around each of the through holes, a first inclined portion inclined with respect to the gas flow direction to form the peak portion, and a second inclined portion connecting the flat portion and the first inclined portion,
the plurality of through holes are formed in a row direction perpendicular to both the gas flow direction and a direction in which the fins are arranged, and
when a length of the fin in the gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in the row direction is defined as S2, a diameter of the flat portion is defined as D1, a flat plane passing through an upstream end and a downstream end of the fin in the gas flow direction is defined as a reference plane, an angle between the reference plane and the first inclined portion is defined as $\theta1$, an angle between the reference plane and the second inclined portion is defined as $\theta2$, and a distance from the reference plane to the flat portion is defined as α, the fin tube heat exchanger satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta1 \pm 2\alpha)/(S2 - D1)\} \leq \theta2 < 80° - \theta1$.

2. The fin tube heat exchanger according to claim 1, wherein the angle $\theta2$ satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta1 \pm 2\alpha)/(S2 - D1)\} \leq \theta2 < 70° - \theta1$.

3. The fin tube heat exchanger according to claim 1, wherein the fin is configured to prevent the gas from flowing from a front side to a back side of the fin in a region other than the plurality of through holes.

4. A fin tube heat exchanger comprising:

a plurality of fins arranged in parallel to form flow passages for a gas;
heat transfer tubes penetrating the fins and configured to allow a medium to flow through the heat transfer tubes to exchange heat with the gas,
wherein each of the fins is a corrugated fin formed to have only one peak portion in a gas flow direction, and has a plurality of through holes through which the heat transfer tubes are fitted, a cylindrical fin collar being in close contact with the heat transfer tube around each of the through holes, a first inclined portion inclined with respect to the gas flow direction to form the peak portion, and a second inclined portion connecting the fin collar and the first inclined portion,
the plurality of through holes are formed in a row direction perpendicular to both the gas flow direction and a direction in which the fins are arranged, and
when a length of the fin in the gas flow direction is defined as S1, a center-to-center distance between the heat transfer tubes in the row direction is defined as S2, an outer diameter of the fin collar is defined as D2, a flat plane passing through an upstream end and a downstream end of the fin in the gas flow direction is defined as a reference plane, an angle between the reference plane and the first inclined portion is defined as $\theta1$, and an angle between the reference plane and the second inclined portion is defined as $\theta2$, the fin tube heat exchanger

satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta1)/(S2 - D2)\} \leq \theta2 < 80° - \theta1$.

5. The fin tube heat exchanger according to claim 4, wherein the angle $\theta2$ satisfies a relationship: $\tan^{-1}\{(S1 \cdot \tan\theta1)/(S2 - D2)\} \leq \theta2 < 70° - \theta1$.

6. The fin tube heat exchanger according to claim 4, wherein the fin is configured to prevent the gas from flowing from a front side to a back side of the fin in a region other than the plurality of through holes.

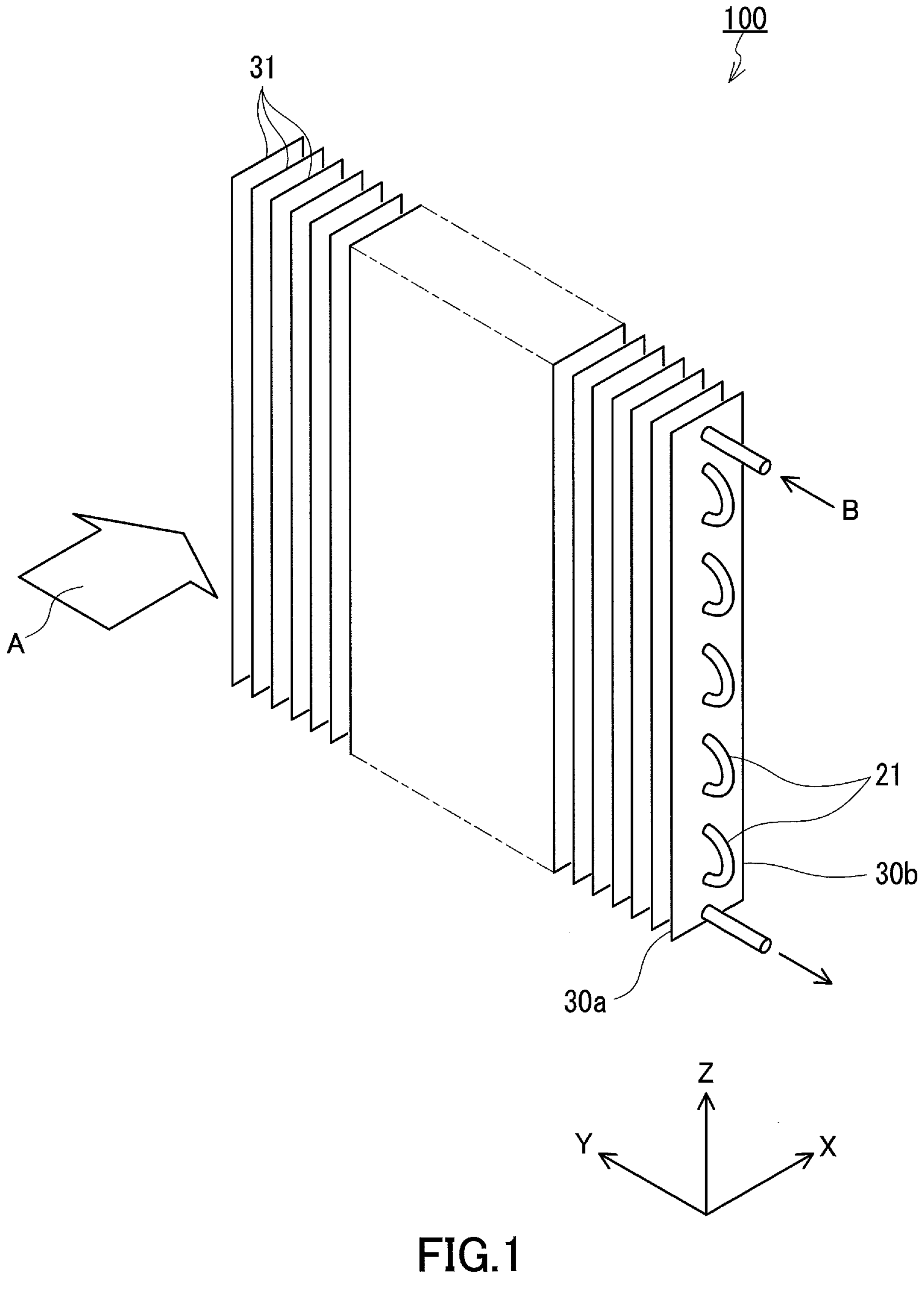


FIG.1

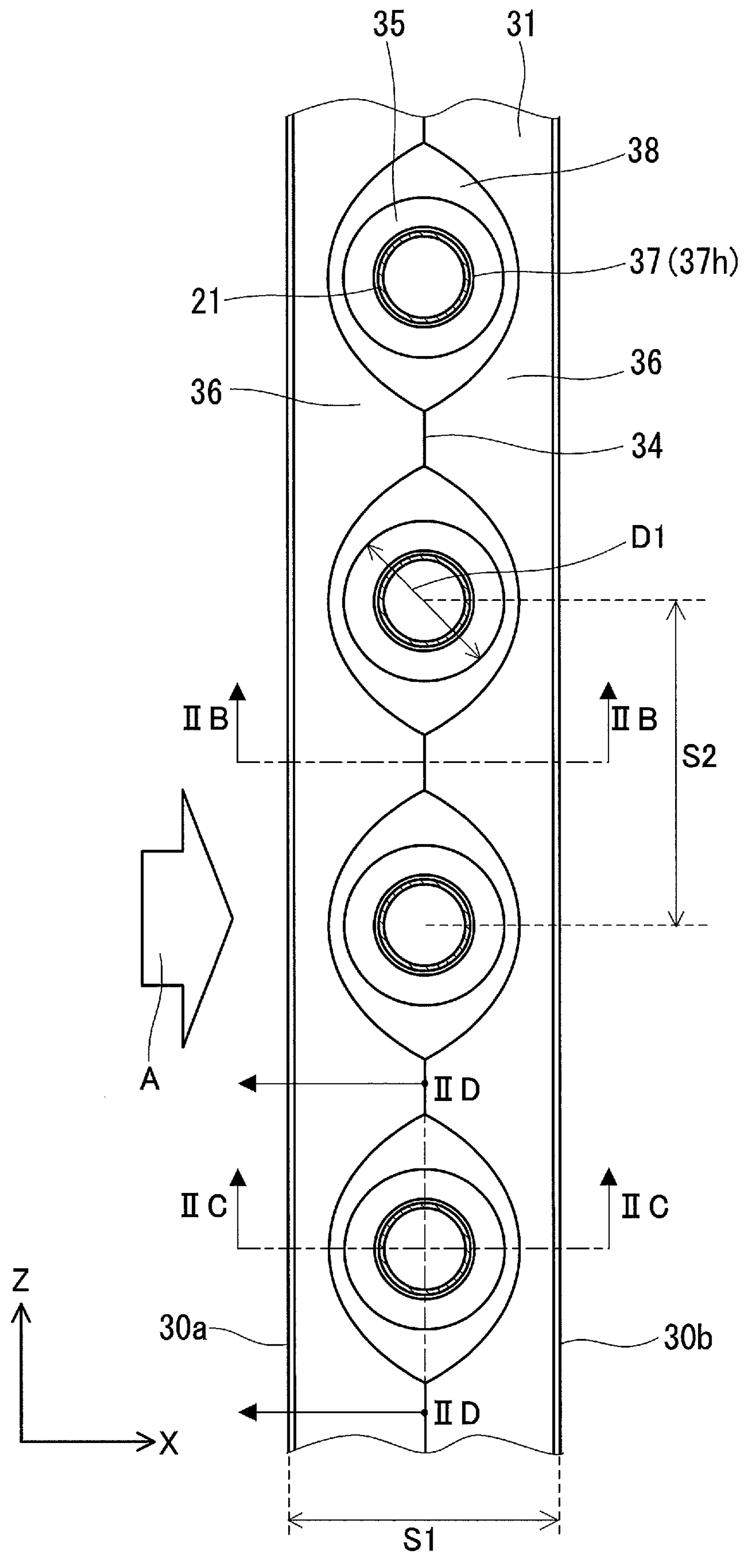


FIG.2A

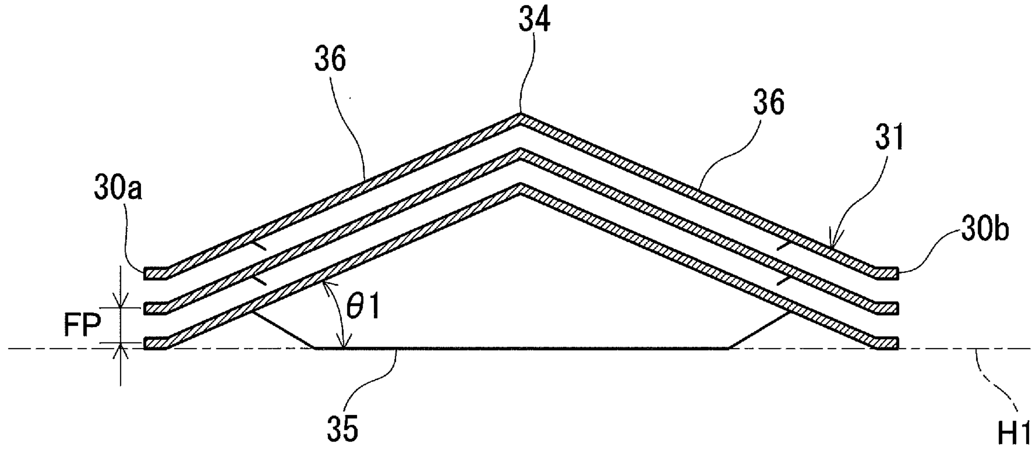


FIG.2B

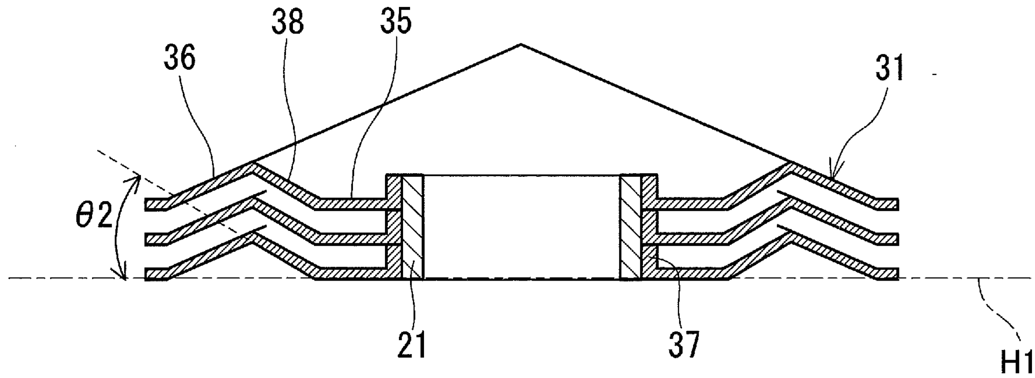


FIG.2C

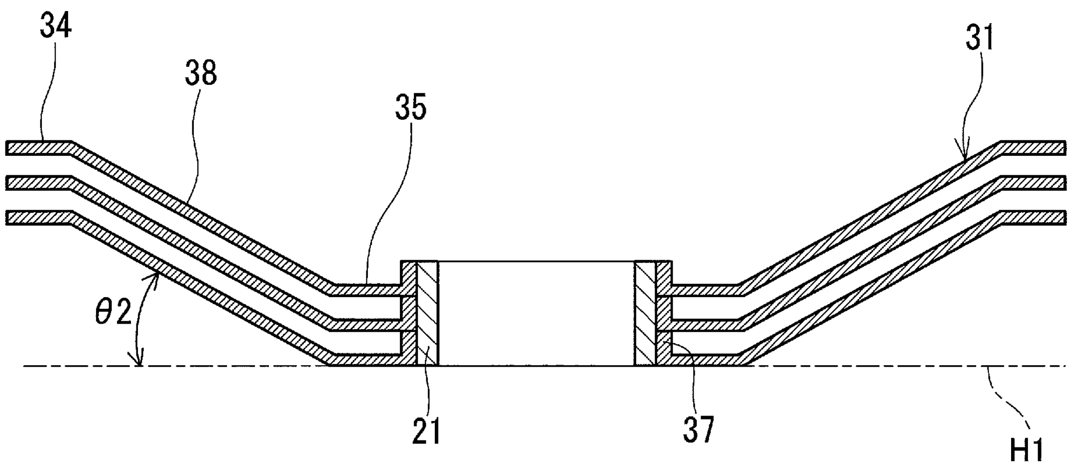


FIG.2D

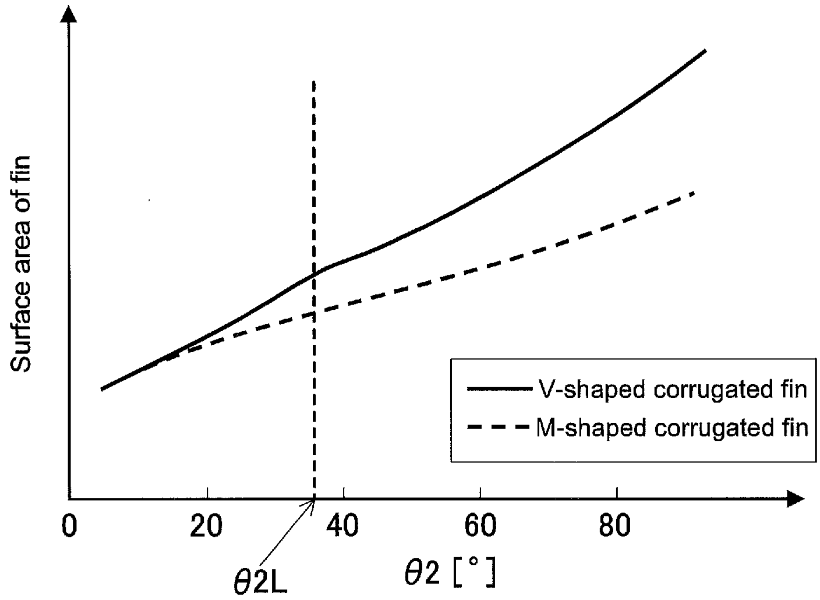

Surface area of fin
V-shaped corrugated fin
M-shaped corrugated fin
0
20
40
60
80
θ2L
θ2 [°]

FIG.3A

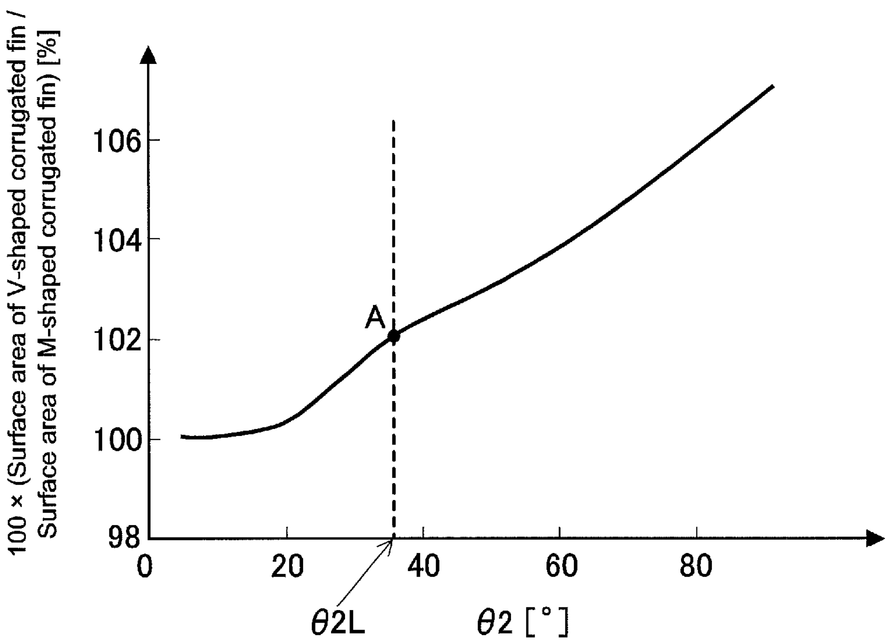

100 × (Surface area of V-shaped corrugated fin / Surface area of M-shaped corrugated fin) [%]
106
104
102
100
98
A
0
20
40
60
80
θ2L
θ2 [°]

FIG.3B

FIG.4A

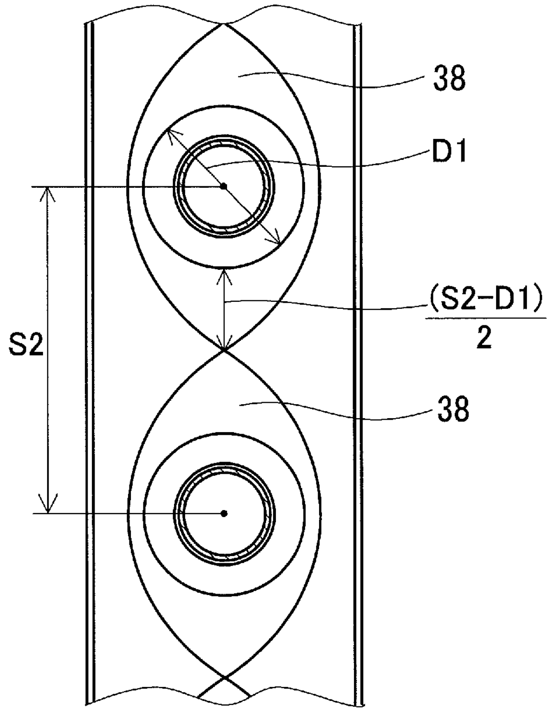
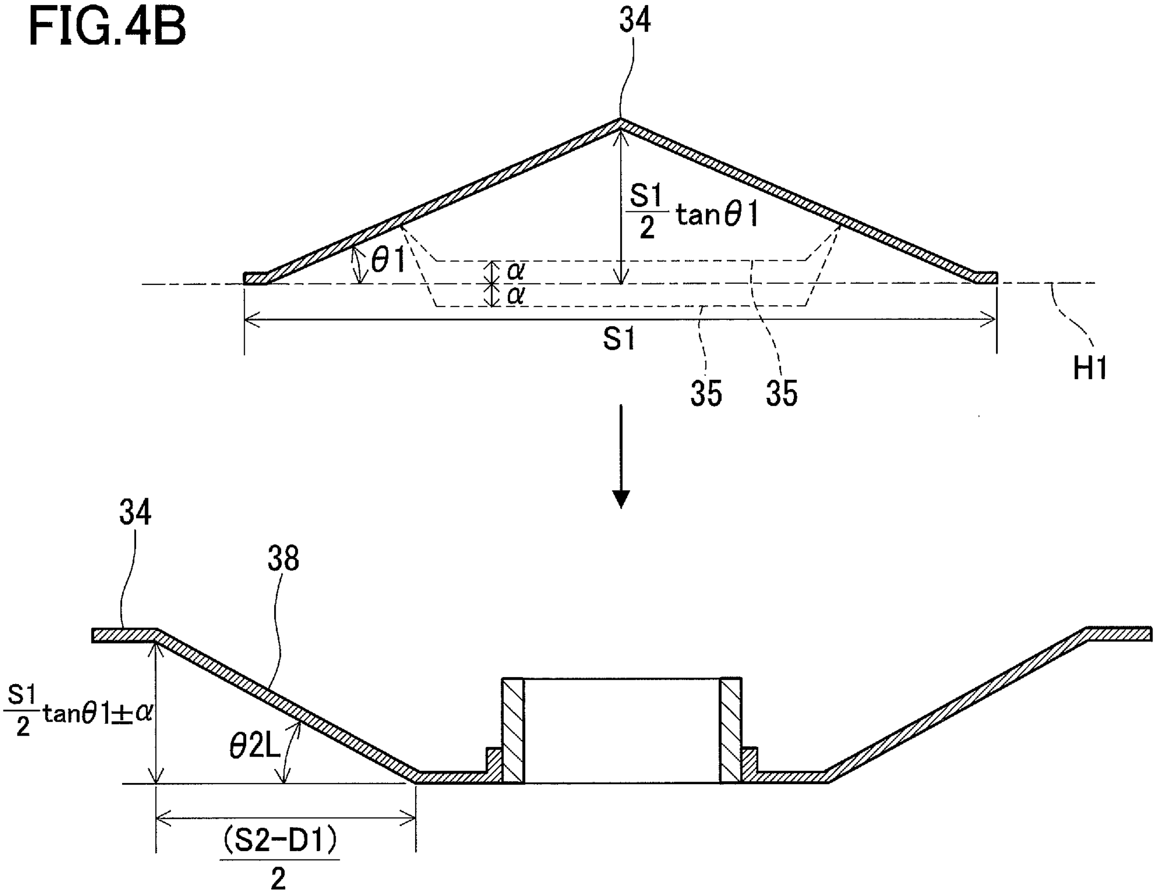

38
D1
(S2-D1)
2
S2
38

FIG.4B


34
S1
2
tanθ1
θ1
α
α
S1
H1
35
35
34
38
S1
2
tanθ1±α
θ2L
(S2-D1)
2

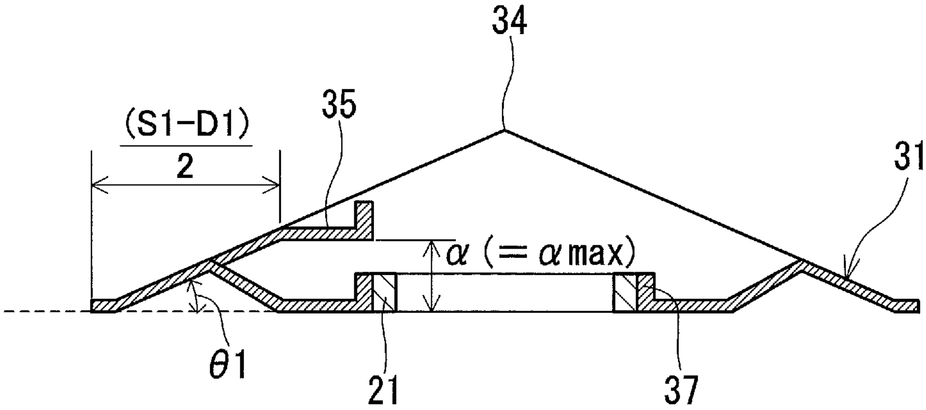


FIG.4C

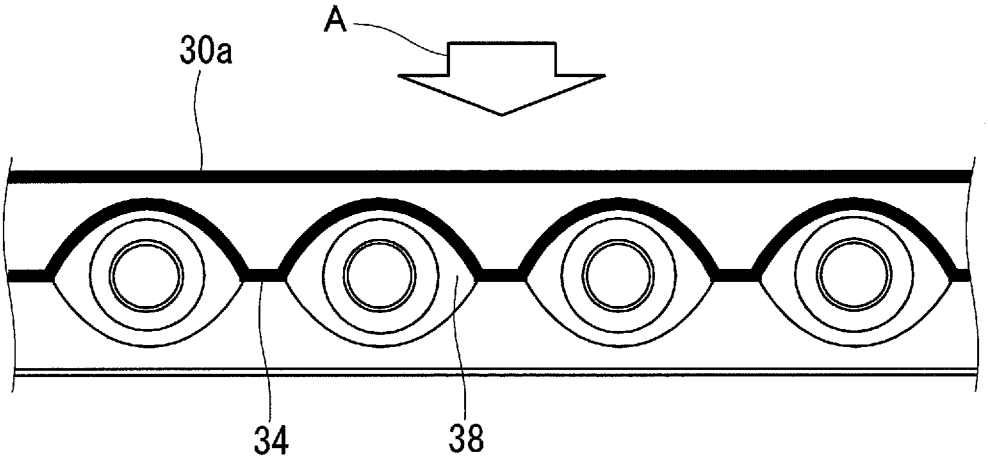


FIG.5A

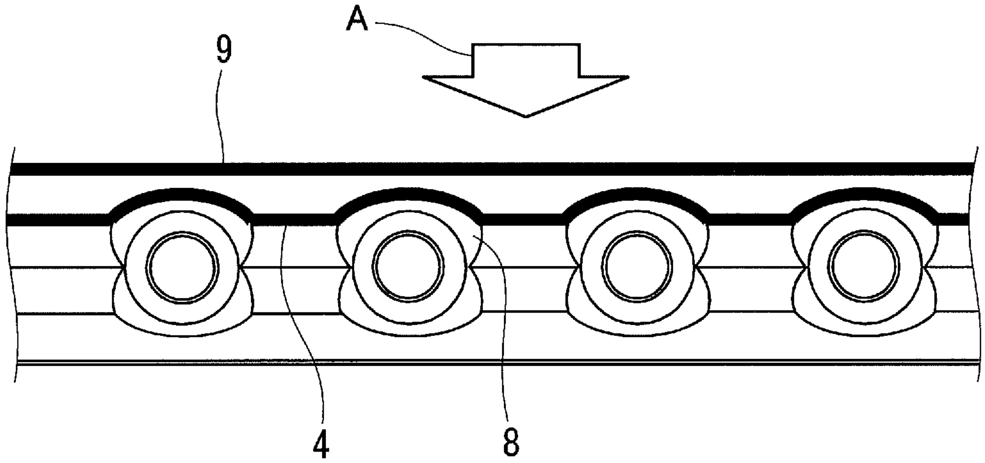


FIG.5B

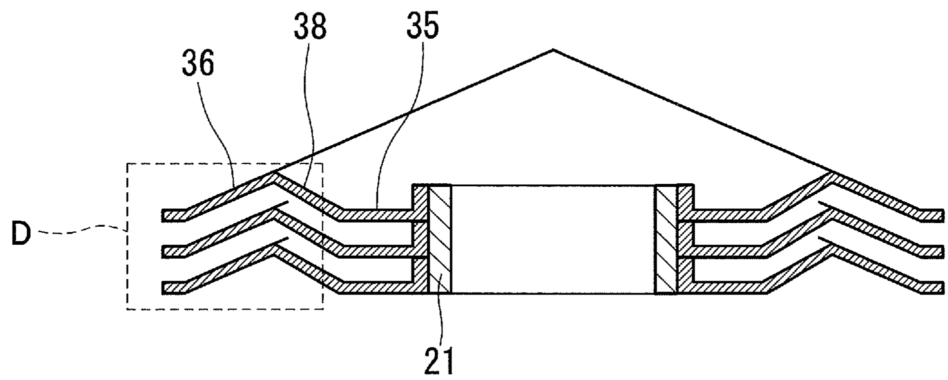


FIG.6A

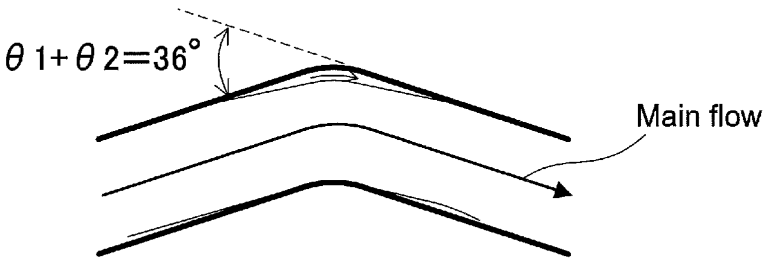


FIG.6B

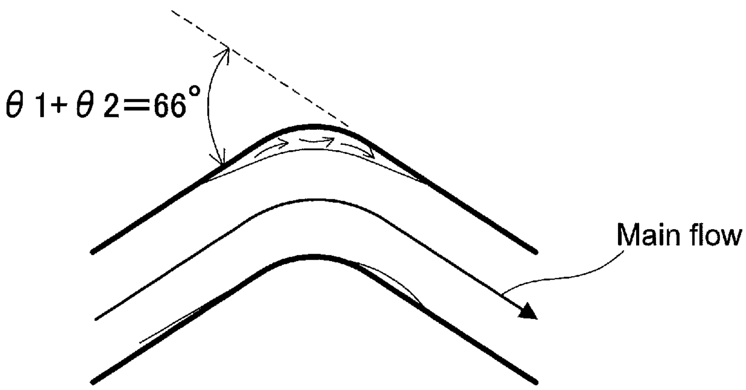


FIG.6C

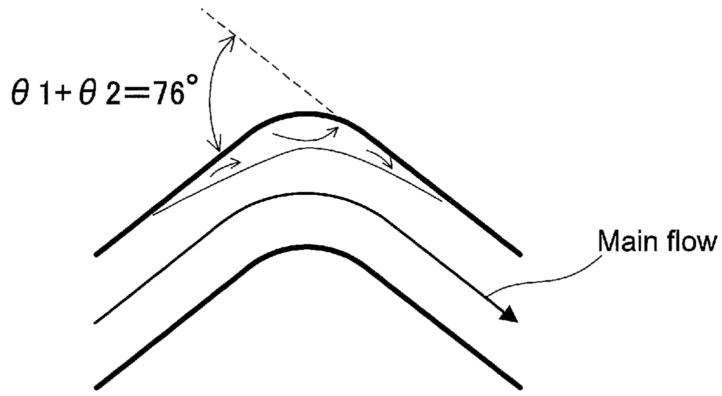


FIG.6D

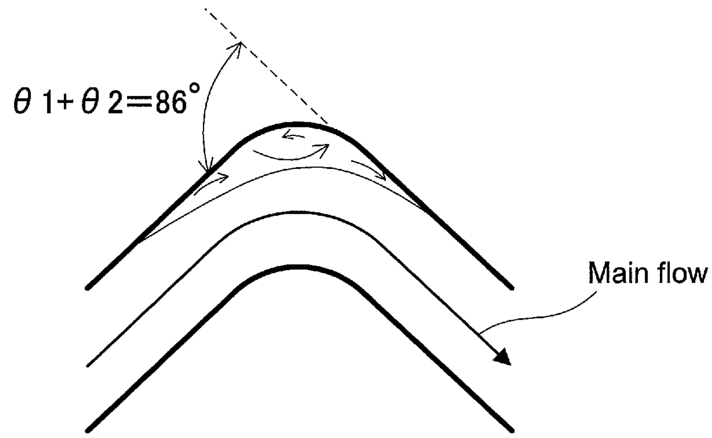


FIG.6E

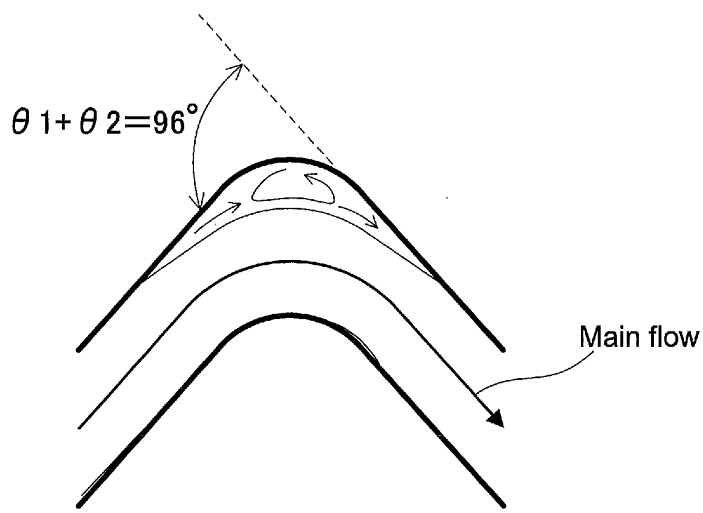


FIG.6F

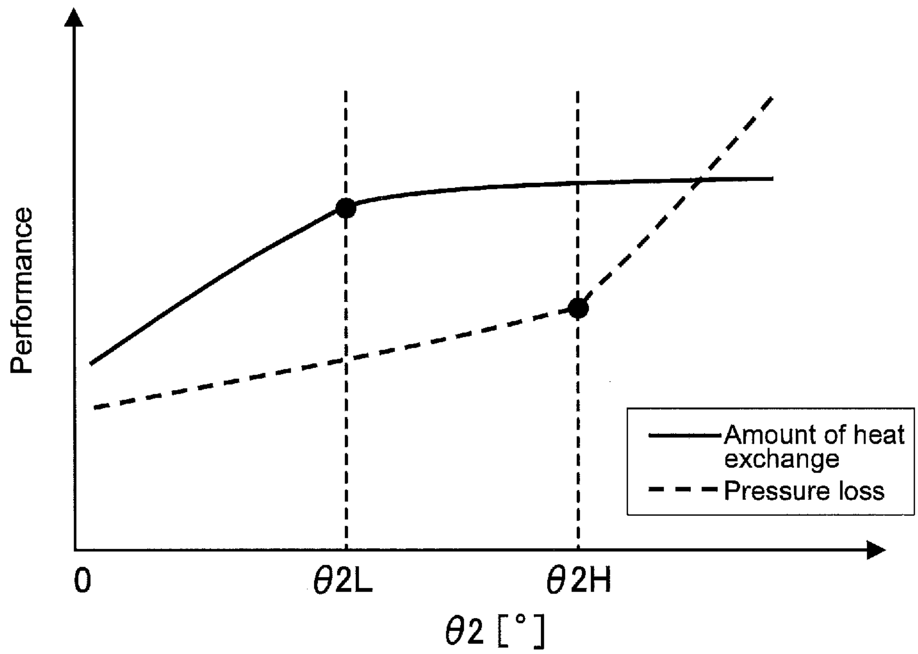

Performance
Amount of heat exchange
Pressure loss
0
θ2L
θ2H
θ2 [°]

FIG.7

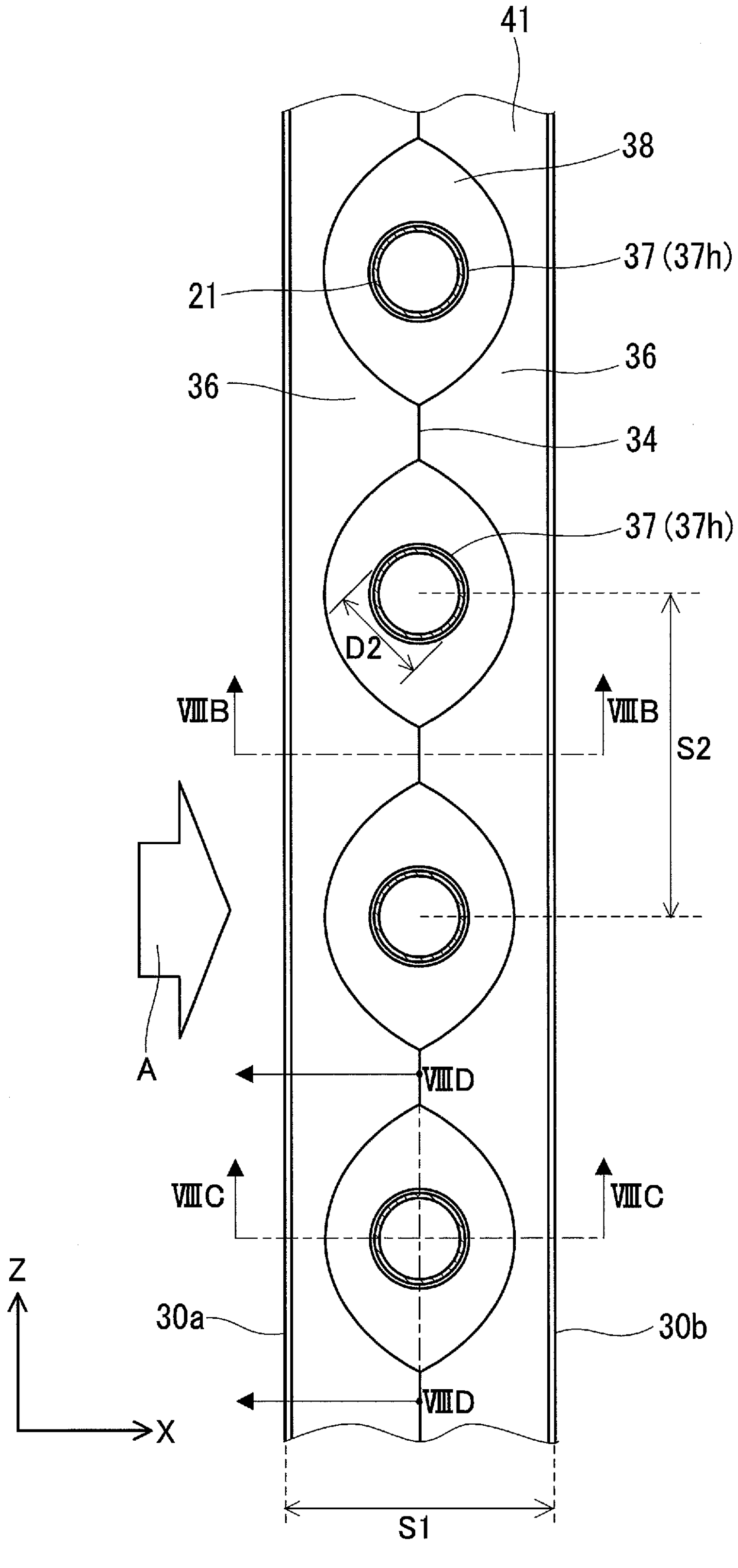


FIG.8A

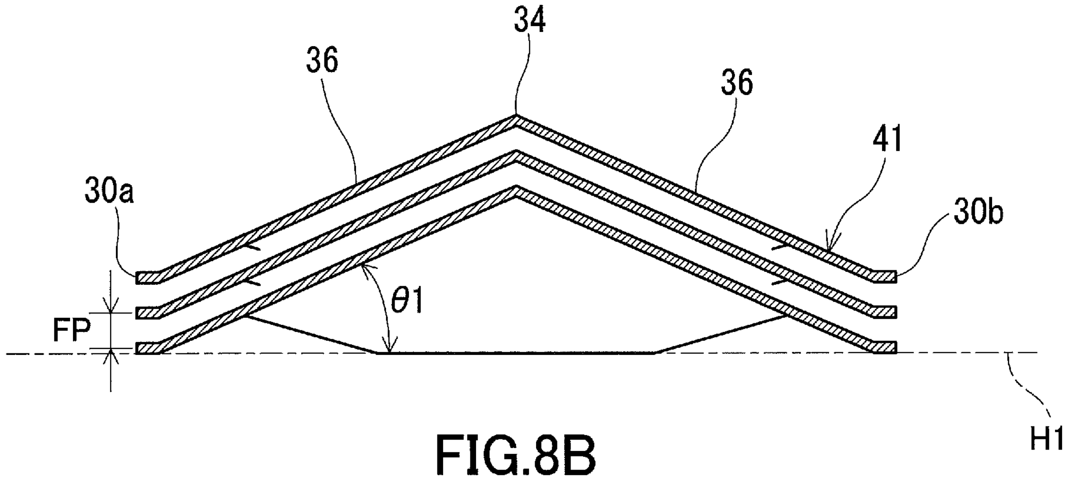


FIG.8B

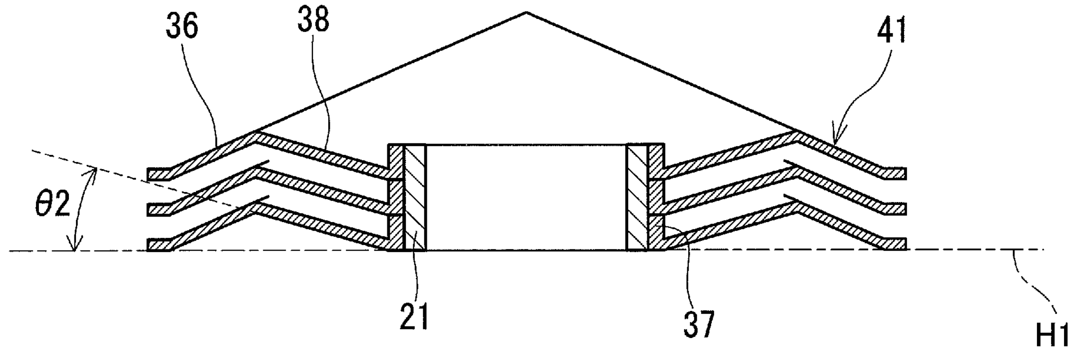


FIG.8C

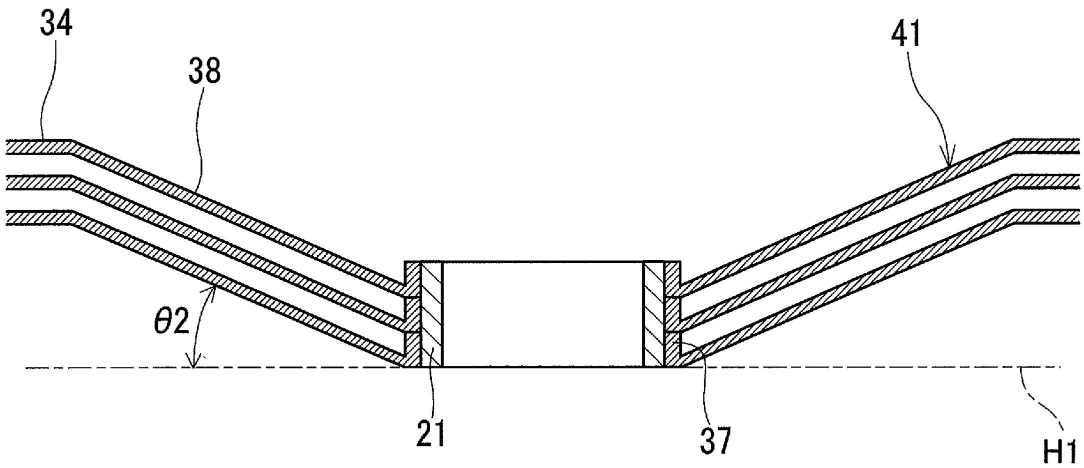


FIG.8D

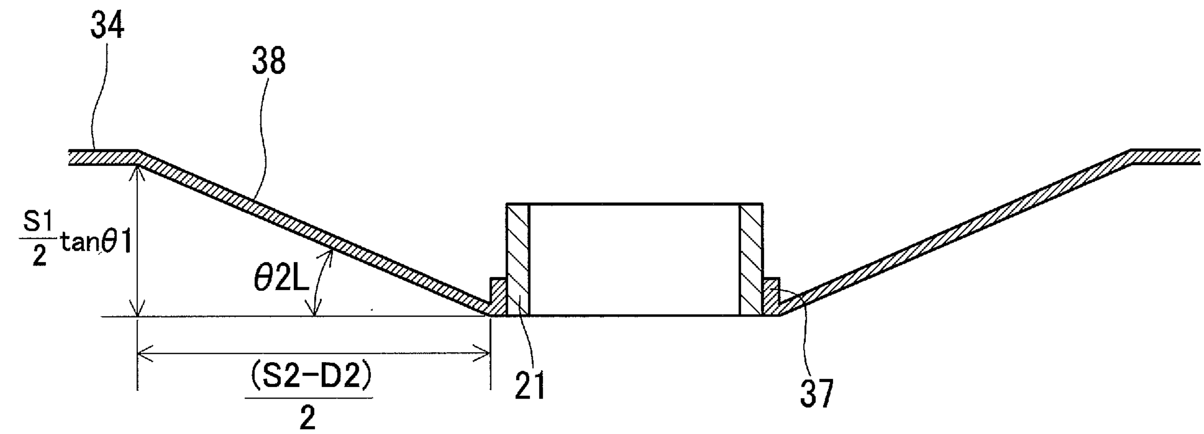


FIG.8E

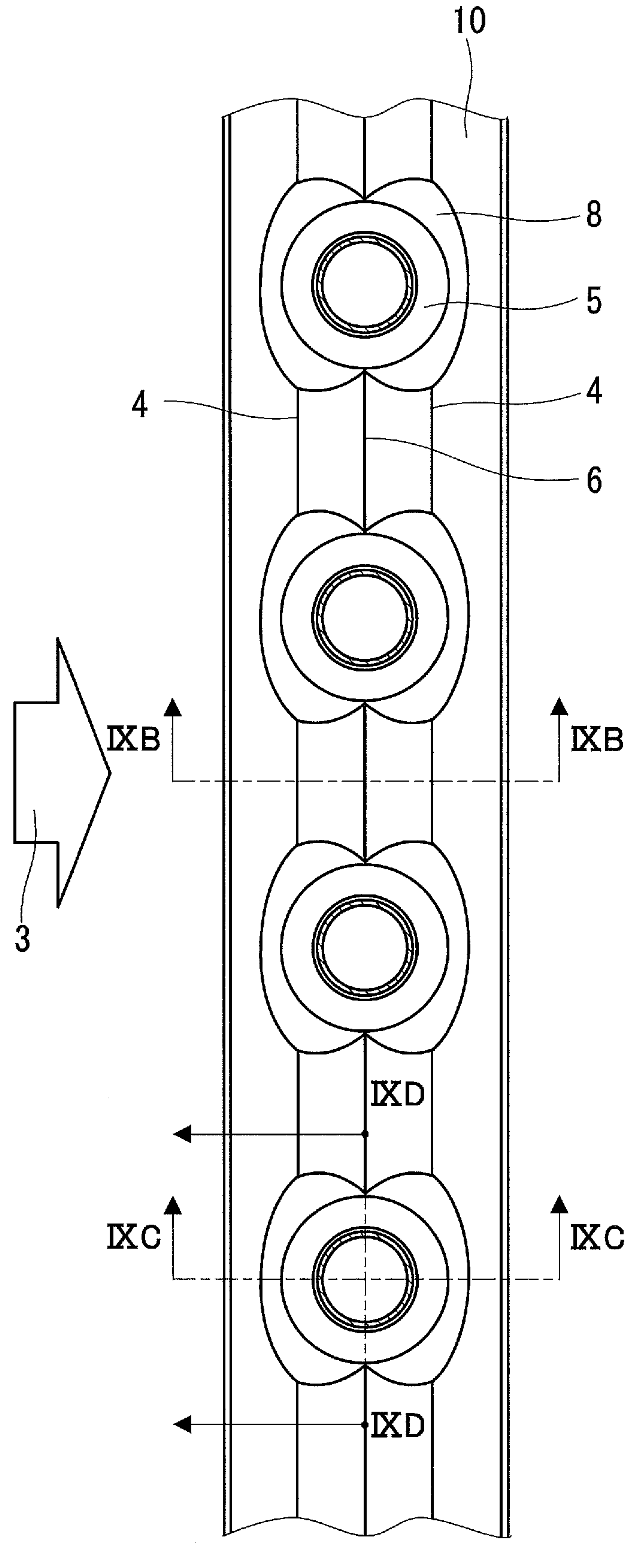


FIG.9A

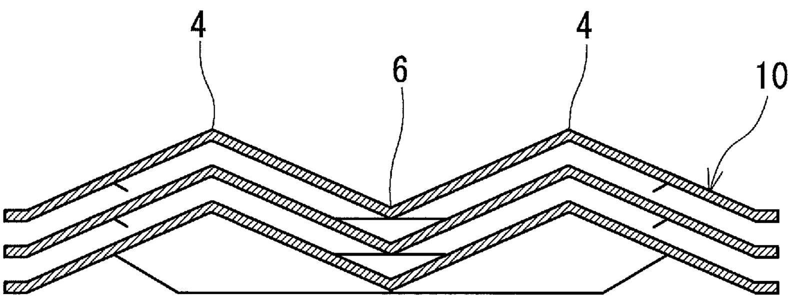


FIG.9B

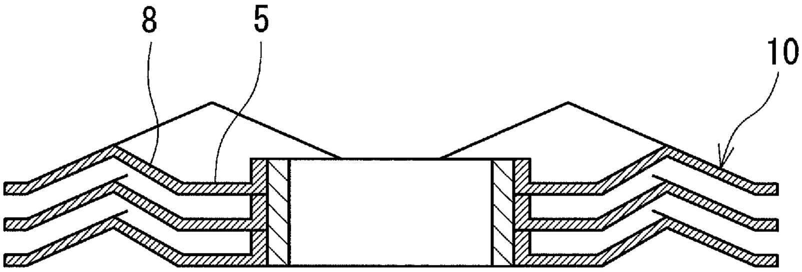


FIG.9C

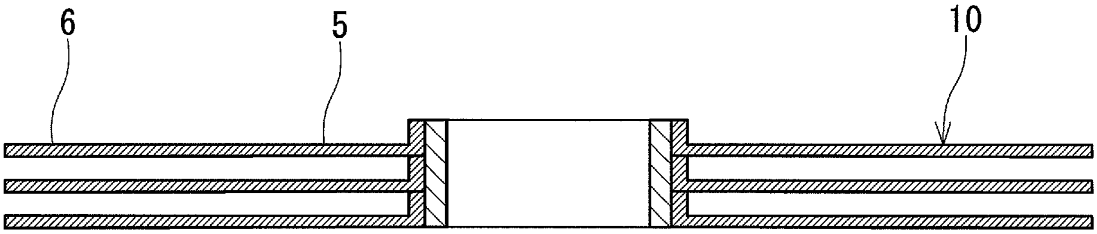


FIG.9D

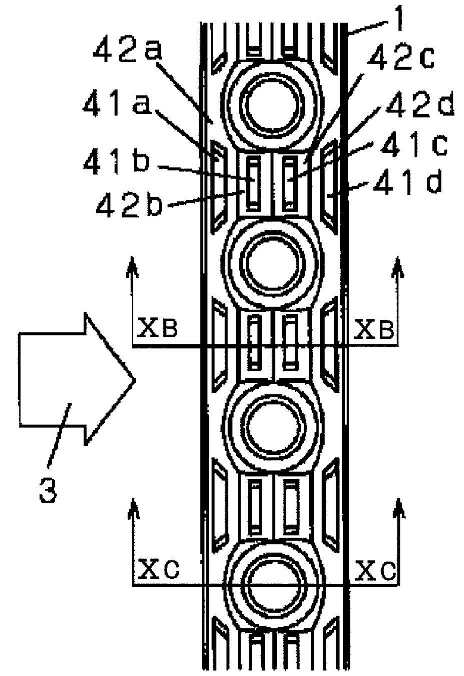


FIG.10A

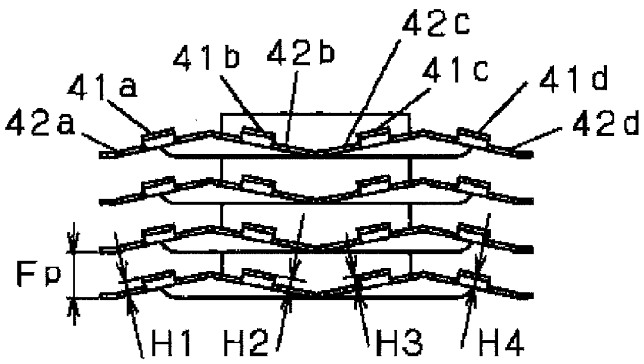


FIG.10B

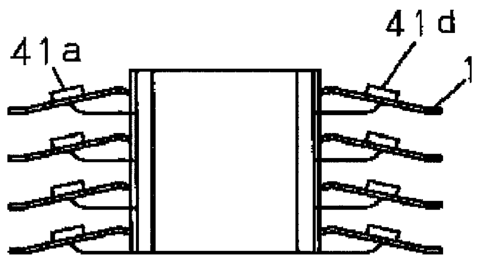


FIG.10C

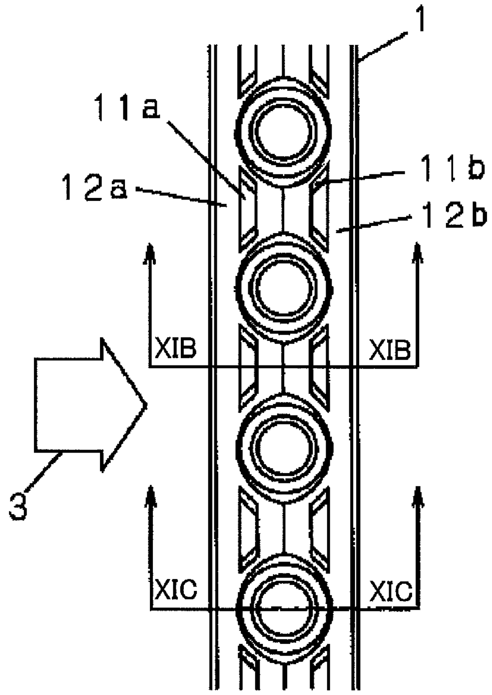


FIG.11A

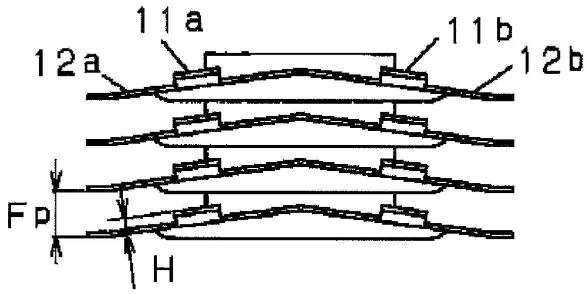


FIG.11B

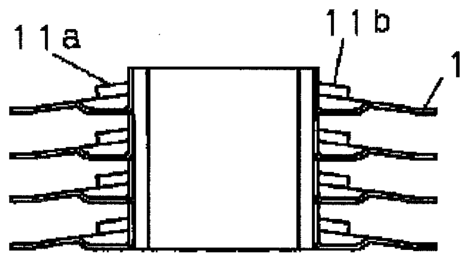


FIG.11C

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2012/006469

A. CLASSIFICATION OF SUBJECT MATTER
F28F1/32(2006.01)i

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)
F28F1/32

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2012
Kokai Jitsuyo Shinan Koho	1971-2012	Toroku Jitsuyo Shinan Koho	1994-2012

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.
A	JP 11-125495 A (Matsushita Electric Industrial Co., Ltd.), 11 May 1999 (11.05.1999), fig. 1 to 3 (Family: none)	1-6
A	JP 10-213386 A (Hitachi, Ltd.), 11 August 1998 (11.08.1998), entire text; all drawings & US 6050328 A & CN 1189605 A	1-6
A	JP 10-141880 A (Matsushita Electric Industrial Co., Ltd.), 29 May 1998 (29.05.1998), entire text; all drawings (Family: none)	1-6

☒ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
05 November, 2012 (05.11.12)	13 November, 2012 (13.11.12)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 2009)

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2012/006469

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 163181/1980(Laid-open No. 087979/1982) (Sanyo Electric Co., Ltd.), 31 May 1982 (31.05.1982), entire text; all drawings (Family: none)	1-6
A	JP 2005-90939 A (LG Electronics Inc.), 07 April 2005 (07.04.2005), entire text; all drawings & US 2005/0056407 A1 & EP 1515107 A1 & KR 10-2005-0027407 A & CN 1598434 A	1-6
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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

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