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(54) **FIN-TUBE HEAT EXCHANGER, FIN FOR HEAT EXCHANGER, AND HEAT PUMP DEVICE**

(57) A fin-tube heat exchanger 1 includes a plurality of fins 3 arrayed spaced apart from and parallel to each other so as to form gaps for allowing a first fluid to flow therethrough, and a plurality of heat transfer tubes 2 penetrating the plurality of fins 3 and for allowing a second fluid to flow therethrough. The plurality of heat transfer tubes 2 includes first heat transfer tubes 2A and second heat transfer tubes 2B arranged in a predetermined row direction that intersects the flow direction of the first fluid. The fins 3 have protrusions 5 each disposed between a first heat transfer tube 2A and a second heat transfer tube 2B, for guiding the first fluid toward the first heat transfer tube 2A side and the second heat transfer tube 2B side. The equivalent diameter of the protrusion 5, as viewed in the axis direction of the heat transfer tubes 2, is equal to or greater than the outer diameter of the heat transfer tubes 2.

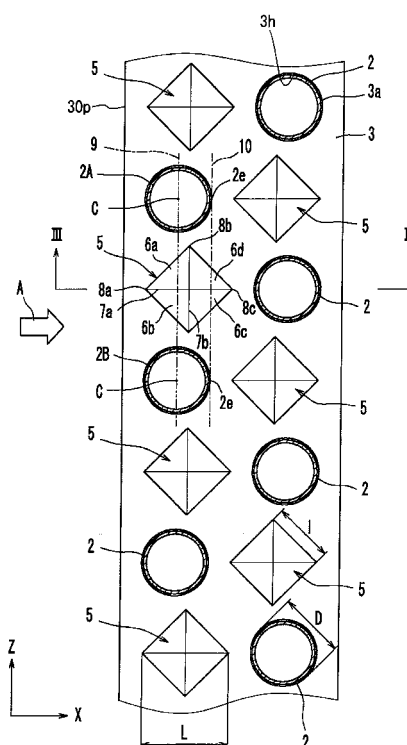


FIG.2

Description

TECHNICAL FIELD

[0001] The present invention relates to fin-tube heat exchangers, fins for heat exchangers, and heat pump apparatuses.

BACKGROUND ART

[0002] Conventionally, fin-tube heat exchangers have been used for various apparatuses such as air conditioners, freezer-refrigerators, dehumidifiers, and hot water heaters. A fin-tube heat exchanger is composed of a plurality of fins that are arranged parallel to each other and spaced apart with a predetermined gap, and heat transfer tubes that extend through these fins.

[0003] Known fin-tube heat exchangers include ones with various fin shape designs so as to, for example, enhance heat transfer and reduce pressure loss. For example, in a fin-tube heat exchanger, the leeward side of the heat transfer tube usually becomes a dead fluid zone, in which the heat transfer coefficient is locally low. In view of this, there are known fin-tube heat exchangers having fins provided with protuberances on the surfaces of the fins so as to reduce the dead fluid zone.

[0004] For example, JP 7-239196 A discloses a fin-tube heat exchanger that uses a fin on the surface of which a large number of very small dimples are provided. Specifically, it is disclosed that, as illustrated in Fig. 34, the air is guided toward the leeward side of heat transfer tubes 101 by a large number of dimples 102 on a fin 103, so that the heat transfer coefficient is improved. Nevertheless, the size of the dimples 102 is so small that the effect of guiding the air to the leeward side of the heat transfer tubes 101 is not as great as expected, and not much improvement in the heat transfer coefficient can be expected.

[0005] JP 63-294494 A discloses a fin-tube heat exchanger in which protuberances in a triangular pyramidal shape are provided on the surface of the fins. In this heat exchanger, triangular pyramidal-shaped protuberances 111 are provided on both sides of each heat transfer tube 112, as illustrated in Fig. 35. JP 63-294494 A mentions that the air is guided to the leeward side of each heat transfer tube 112 by these protuberances 111, and a flow A becomes a narrow flow C at the rear of each heat transfer tube 112, increasing the flow velocity. Accordingly, a violent turbulent flow is produced at the rear of the heat transfer tube 112, and as a result, the dead fluid zone becomes significantly small. (See page 4, top right column, line 17 to bottom left column, line 1 of JP 63-294494 A.)

[0006] The heat exchanger disclosed in JP 63-294494 A (see Fig. 35) has the problem that the pressure loss tends to be great because the flow is greatly narrowed between two triangular pyramidal-shaped protuberances 111. Moreover, the heat exchanger disclosed in JP 63-294494 A is the one used for a turbulent flow region in which the air flow velocity is high and is intended to obtain a turbulent flow promotion effect. For this reason, it cannot necessarily exhibit satisfactory performance in a laminar flow region in which the air flow velocity is low.

[0007] JP 6-300474 A discloses a fin-tube heat exchanger in which quadrangular pyramidal-shaped protrusions are provided on the surface of each fin. In this heat exchanger, as illustrated in Fig. 36, cut-and-raised pieces 122 are formed on the leeward side of each heat transfer tube 121, and the quadrangular pyramidal-shaped protrusions 123 are disposed between adjacent heat transfer tubes 121 and more leeward than the heat transfer tubes 121. JP 6-300474 A describes that the air is guided toward the cut-and-raised pieces 122 by the protrusions 123, so the heat transfer coefficient on the leeward side of the heat transfer tube 121 improves (see page 4, right column, lines 30 to 36 of JP 6-300474 A).

[0008] In the heat exchanger disclosed in JP 6-300474 A (see Fig. 36), the protrusions 123 are disposed more leeward than the center line 124 of the heat transfer tubes 121. For this reason, the air flow direction is changed in a more downstream side than the center of the heat transfer tubes 121. However, it is difficult to change the air flow abruptly since the protrusions 123 are relatively small. Thus, it is difficult to guide the air toward the rear of the heat transfer tubes 121.

[0009] JP 2002-90085 A discloses a fin-tube heat exchanger that uses a fin 105 in which raised parts 106 are formed, as illustrated in Fig. 37. This type of fin 105 is commonly referred to as a slit fin. Although the slit fin 105 is slightly disadvantageous in terms of pressure loss, it significantly improves the heat transfer performance of the heat exchanger by the effect of local increase in the heat transfer coefficient at the leading edge of the raised part 106, i.e., what is called the leading edge effect. However, since it tends to cause clogging because of frost formation at the raised part 106 and thus to result in a considerable decrease in heat exchange efficiency, the applications of the slit fin 105 are limited. Specifically, it is difficult to apply the slit fin to such an evaporator that evaporates the refrigerant in an atmosphere at relatively low temperatures.

[0010] It has been known that, in the fin-tube heat exchanger, if the velocity of the fluid is increased to raise the heat transfer coefficient for the purpose of increasing the amount of heat transfer between the refrigerant and the fluid (for example, the air), the pressure loss when the fluid passes through the heat exchanger also increases, so the mechanical power required for the fan for causing the fluid to flow becomes too high. In other words, there is a trade-off between

pressure loss and heat transfer coefficient, which is an indicator of heat transfer performance. As an attempt for the fin-tube heat exchanger to achieve both good heat transfer performance and low pressure loss, various types of heat exchangers that employ a corrugated fin, in which a plate-shaped fin is bent in a wave-like shape, have been proposed. **[0011]** For example, Figs. 38A and 38B show a corrugated fin disclosed in JP 1-90995 A. This corrugated fin 109 is formed so that ridges 109a and valleys 109b appear alternately along the fluid flow direction indicated by the arrow. Such a corrugated fin 109 has the advantages of showing good balance between heat transfer performance and pressure loss, being free of the clogging problem due to frost formation, and having no limitation on the applications.

DISCLOSURE OF THE INVENTION

[0012] In recent years, demands for reducing energy consumption of heat pump apparatuses used for hot water heaters and air conditioners have been growing due to various issues such as urban heat island issues, natural resource issues, and global environment issues. For further reduction in energy consumption of a heat pump apparatus, it is essential to improve the heat exchanger, as well as the compression mechanism and the expansion mechanism. Specifically, a fin-tube heat exchanger that achieves better heat transfer performance and lower pressure loss than the one that uses a corrugated fin has been desired.

[0013] The present invention has been accomplished in view of the foregoing circumstances, and it is an object of the invention to provide a fin-tube heat exchanger that has excellent heat transfer performance and at the same time shows low pressure loss. It is another object of the invention to provide a heat pump apparatus that has the fin-tube heat exchanger. It is yet another object of the invention to provide a fin that suitably can be employed for the fin-tube heat exchanger.

[0014] Accordingly, the present invention provides a fin-tube heat exchanger for exchanging heat between a first fluid and a second fluid, including:

a plurality of fins arranged spaced apart from and parallel to each other so as to form a space or spaces for allowing the first fluid to flow therethrough; and

a plurality of heat transfer tubes penetrating the plurality of fins, for allowing the second fluid to flow therethrough, wherein:

the plurality of heat transfer tubes include a first heat transfer tube and a second heat transfer tube arranged in a predetermined row direction that intersects with a flow direction of the first fluid;

the first heat transfer tube and the second heat transfer tube are adjacent to each other with respect to the row direction;

each of the fins has a protrusion formed between the first heat transfer tube and the second heat transfer tube, the protrusion guiding the first fluid toward a first heat transfer tube side and a second heat transfer tube side; and the equivalent diameter of the protrusion, as viewed in an axis direction of the heat transfer tubes, is equal to or greater than the outer diameter of the heat transfer tubes.

[0015] The present invention also provides a fin that is used for the just-mentioned fin-tube heat exchanger.

[0016] The present invention also provides a heat pump apparatus including:

a compressor for compressing a refrigerant;

a radiator for cooling the refrigerant compressed by the compressor;

an expansion mechanism for expanding the refrigerant cooled by the radiator; and

an evaporator for evaporating the refrigerant expanded by the expansion mechanism, wherein

at least one of the evaporator and the radiator includes the above-described fin-tube heat exchanger.

[0017] The foregoing fin-tube heat exchanger according to the present invention aims at enlarging the heat transfer area in the fin and at the same time hindering development of a thermal boundary layer and a velocity boundary layer, by forming a protrusion with a large surface area between the first heat transfer tube and the second heat transfer tube. By enlarging the heat transfer area and hindering development of the boundary layers, the heat transfer performance of the fin-tube heat exchanger is improved. In addition, the equivalent diameter of the protrusion, as viewed in the axis direction of the heat transfer tubes, is equal to or greater than the outer diameter of the heat transfer tubes. In other words, when the protrusion is orthogonally projected onto a plane parallel to the plurality of fins, the area of the image of the protrusion appearing on the plane becomes greater than the cross-sectional area of the heat transfer tube. Such a protrusion makes it possible to gain a sufficient surface area of the fin. Moreover, the protrusion with a relatively large size has the effect of increasing the flow velocity on the first fluid in the flat region between the protrusion and the first heat transfer tube, or in the flat region between the protrusion and the second heat transfer tube. An increased flow

velocity is desirable because the heat transfer coefficient accordingly becomes high. In particular, it allows the side face portion in the outer circumferential surface of heat transfer tube (including the outer circumferential surface of the fin collar surrounding the heat transfer tube) that faces the protrusion to contribute to the heat transfer. Furthermore, the protrusion guides the first fluid to the rear of the heat transfer tubes. This prevents the development of a large dead fluid zone in the rear of the heat transfer tubes, resulting in an improvement of the heat transfer performance of the fin-tube heat exchanger.

BRIEF DESCRIPTION OF THE DRAWINGS

[0018]

Fig. 1 is a perspective view of a fin-tube heat exchanger.

Fig. 2 is a plan view of a fin according to Embodiment 1.

Fig. 3 is a cross-sectional view taken along line III-III in Fig. 2.

Fig. 4 is a view illustrating a modified example and corresponding to Fig. 3.

Fig. 5 is a plan view of the fin according to Embodiment 1, which illustrates an air flow.

Fig. 6 is a perspective view of the fin according to Embodiment 1.

Fig. 7 is a plan view of a simulation model.

Fig. 8 is a performance characteristic diagram of a fin-tube heat exchanger according to Embodiment 1.

Fig. 9 is a cross-sectional view of the fin according to Embodiment 1, which schematically illustrates a thermal boundary layer.

Fig. 10 is a plan view of a fin according to Embodiment 2.

Fig. 11 is a perspective view of the fin according to Embodiment 2.

Fig. 12 is a plan view of a fin according to Embodiment 3.

Fig. 13 is a plan view illustrating a modified example of the fin according to Embodiment 1.

Fig. 15 is a plan view of a fin according to Embodiment 4.

Fig. 15 is a cross-sectional view taken along line XV-XV in Fig. 14.

Fig. 16A is a schematic view illustrating one example of the shape of a protrusion.

Fig. 16B is a schematic view illustrating another example of the shape of a protrusion.

Fig. 17A is a graph showing a clothoid curve.

Fig. 17B is a cross-sectional view of a protrusion a surface of which forms a clothoid curve.

Fig. 18 is a view illustrating the work of a fin-tube heat exchanger that employs the fin according to Embodiment 4.

Fig. 19 is a view illustrating the work of a fin-tube heat exchanger that employs a fin of a comparative example.

Fig. 20 is a plan view of a fin according to Embodiment 5.

Fig. 21 is a plan view of a fin according to Embodiment 6.

Fig. 22 is a plan view of a fin according a modified example of Embodiment 6.

Fig. 23A is a plan view of a fin according to Embodiment 7.

Fig. 23B is a cross-sectional view taken along line D1-D1 in Fig. 23A.

Fig. 24A is a plan view of a fin according a modified example of Embodiment 7.

Fig. 24B is a cross-sectional view taken along line D2-D2 in Fig. 24A.

Fig. 25 is a configuration diagram of a heat pump apparatus.

Fig. 26 is a schematic view illustrating an application example of the heat pump apparatus of Fig. 25.

Fig. 27A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Example 1.

Fig. 27B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 27A.

Fig. 28A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Example 2.

Fig. 28B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 28A.

Fig. 29A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Example 3.

Fig. 29B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 29A.

Fig. 30A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Comparative Example 1.

Fig. 30B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 30A.

Fig. 31A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Example 4.

Fig. 31B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 31A.

Fig. 32A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger

of Example 5.

Fig. 32B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 32A.

Fig. 33A is a contour diagram illustrating the simulation results (Nusselt number distribution) for a heat exchanger of Comparative Example 2.

Fig. 33B is a contour diagram illustrating the simulation results (flow velocity distribution) following Fig. 33A.

Fig. 34 is a plan view of a fin of a conventional fin-tube heat exchanger.

Fig. 35 is a plan view of a fin of a conventional fin-tube heat exchanger.

Fig. 36 is a plan view of a fin of a conventional fin-tube heat exchanger.

Fig. 37 is a perspective view of slit fins.

Fig. 38A is a plan view of a corrugated fin.

Fig. 38B is a cross-sectional view of a corrugated fin.

BEST MODE FOR CARRYING OUT THE INVENTION

(Embodiment 1)

[0019] Hereinbelow, one embodiment of the present invention is described with reference to the appended drawings.

[0020] Fig. 1 is an overall perspective view illustrating a fin-tube heat exchanger according to the present embodiment. The fin-tube heat exchanger 1 has a plurality of fins 3 arranged parallel to each other with a predetermined gap for forming spaces for allowing the first fluid to flow therethrough, and a plurality of heat transfer tubes 2 penetrating these fins 3. The heat exchanger 1 is for exchanging heat between the first fluid flowing along the principal surfaces of the fins 3 and the second fluid flowing inside the heat transfer tubes 2. In the present embodiment, air A flows along the principal surfaces of the fins 3, and refrigerant B flows inside the heat transfer tubes 2. The plurality of heat transfer tubes 2 penetrating the fins 3 are connected in a single line so that the refrigerant B flows therein in turn. It should be noted that the type and state of the fluid that flows inside the heat transfer tubes 2 and of the fluid that flows along the principal surfaces of the fins 3 are not particularly limited. Each of the fluids may be either a gas or a liquid. In addition, the plurality of heat transfer tubes 2 may not necessarily be connected in a single tube.

[0021] The heat exchanger 1 is installed in such a position that the flow direction of the air A (X direction) is approximately perpendicular to the stacking direction of the fins 3 (Y direction) and the row direction of the heat transfer tubes 2 (Z direction). That said, the airflow direction may be slightly inclined from the X direction as long as a sufficient heat exchange amount can be ensured. It should be noted that, in the present specification, the stacking direction (Y direction) that is perpendicular to the principal surfaces of the fins 3 is defined as a height direction.

[0022] Each of the fins 3 has a rectangular, flat plate-like shape, and they are arranged along the Y direction shown in Fig. 1. In the present embodiment, the fins 3 are arranged at a predetermined gap (fin pitch). The fin pitch is, for example, from 1.0 mm to 1.5 mm. The fin pitch may not necessarily be uniform, and it may be varied. As illustrated in Fig. 3, fin pitch FP is defined as the distance between the centers of two adjacent fins 3, 3 with respect to the thickness direction. Each of the fins 3 may be composed of, for example, a metal plate subjected to a punch-out process, having a thickness of 0.08 mm to 0.2 mm. The metal plate is, for example, a flat plate made of aluminum. A plurality of through holes 3h (see Fig. 2) are formed in a staggered manner along the longitudinal direction of the fin 3, and the heat transfer tubes 2 are fitted into the respective through holes 3h. It is preferable that the surface of the fin 3 be subjected to a hydrophobic treatment or a hydrophilic treatment, such as a boehmite treatment or coating with a hydrophilic paint.

[0023] Fig. 2 is a plan view of a fin used in the heat exchanger shown in Fig. 1. It should be noted that the heat transfer tubes 2 are shown in their cross sections that are parallel to the principal surface of the fin 3 (more specifically, the principal surface in the flat region of the fin 3 on which no protrusion 5 is formed). As illustrated in Fig. 2, the heat transfer tubes 2 are disposed in two rows, a front row and a rear row, along the row direction that is parallel to the longitudinal direction of the fins 3. In other words, each of the linear lines connecting the centers of the through holes 3h in each of the rows are parallel to the leading edge 30p of the fin 3. The heat transfer tubes 2 in the first row and the heat transfer tubes 2 in the second row are staggered in the Z direction by 1/2 of the tube pitch. In other words, the heat transfer tubes 2 are arranged in a staggered manner. The shortest distance between the centers of two heat transfer tubes 2, 2 that belong to the same row (= tube pitch) may be set at from 2 times to 3 times of the outer diameter D of the heat transfer tubes 2. The outer diameter D of the heat transfer tubes 2 is, for example, from 1 mm to 20 mm, which agrees with the opening diameter of the through holes 3h formed in the fin 3. The heat transfer tubes 2 are in intimate contact with fin collars 3a that form the through holes 3h, and they are fitted to the fin collars 3a. The heat transfer tubes 2 as described above are composed of a good conductive metal, such as copper or a copper alloy, and each of them is a smooth tube, the inner surface of which is flat and smooth, or a grooved tube in which grooves are formed in the inner surface thereof.

[0024] Protrusions 5 in a regular square pyramidal shape are formed on the surface of the fin 3. The protrusions 5 protrude from one of the surfaces of the fin 3. The protrusions 5 are disposed between the respective heat transfer tubes 2 in each of the rows. In the present embodiment, the protrusions 5 are disposed at the midpoints between the adjacent

heat transfer tubes 2 in a row direction. The area of the protrusions 5 viewed in the Y direction, that is, the area of the protrusions 5 in the plan view of Fig. 2, is set at equal to or greater than the area of the heat transfer tubes 2. In other words, the equivalent diameter d (the equivalent diameter d is defined by the equation $\pi d^2/4 = S$ (area)) of the protrusions 5 is equal to or greater than the outer diameter D of the heat transfer tubes 2. More specifically, in the present embodiment, the length 1 of the bottom side of the protrusion 5 is equal to the outer diameter D of the heat transfer tubes 2, and the equivalent diameter d of the protrusions 5 is greater than the outer diameter D of the heat transfer tubes 2. It should be noted that reference character L indicates the airflow-wise length (the length along the X direction) of the protrusions 5.

[0025] The width of each of the protrusions 5 along the Z direction increases from an upstream edge 8a thereof to an intermediate portion 8b thereof, and it decreases from the intermediate portion 8b to a downstream edge 8c thereof, along the flow direction of the air A. Each of the protrusions 5 had a first slanted surface 6a facing toward the top left of Fig. 2, a second slanted surface 6b facing toward the bottom left, a third slanted surface 6c facing toward the bottom right, and a fourth slanted surface 6d facing toward the top right. The first slanted surface 6a and the second slanted surface 6b are divided by a ridge line 7a extending in the X direction. The first slanted surface 6a is slanted toward a heat transfer tube 2A (hereinafter referred to as a first heat transfer tube), one of the adjacent heat transfer tubes, and the second slanted surface 6b is slanted toward a heat transfer tube 2B (hereinafter referred to as a second heat transfer tube), the other one of the adjacent heat transfer tubes. The first slanted surface 6a and the fourth slanted surface 6d are divided by a ridge line 7b extending in the Z direction. The second slanted surface 6b and the third slanted surface 6c are also divided by the ridge line 7b.

[0026] In the present heat exchanger 1, the protrusions 5 are disposed at a relatively upstream side. Specifically, the upstream edges 8a of the protrusions 5 are located upstream of the centers C of the heat transfer tubes 2. The intermediate portions 8b of the protrusions 5 are located upstream of the downstream edges 2e of the heat transfer tubes 2. In other words, the upstream edges 8a of the protrusions 5 are located upstream of the line 9 connecting the centers C of the heat transfer tubes 2, and the intermediate portions 8b of the protrusions 5 are located upstream of the line 10 connecting the downstream edges 2e of the heat transfer tubes 2. The downstream edges 8c of the protrusions 5 are located downstream of the downstream edges 2e of the heat transfer tubes 2.

[0027] In the present embodiment, the height H of the protrusions 5 is greater than the fin pitch FP, as illustrated in Fig. 3. Thus, in the fins 3 adjacent to each other along the Y direction, portions of the protrusions 5 of one of the fins 3 are placed in the back of the protrusions 5 of the other one of the fins 3. However, it is possible that the height H of the protrusions 5 may be equal to the fin pitch FP, or that it may be less than the fin pitch FP. The height H of the protrusions 5 may be, for example, from 0.2 times to 2 times the fin pitch FP, or from 0.5 times to 2 times the fin pitch FP.

[0028] However, as shown in Fig. 4, if the height H of the protrusions 5 is less than the fin pitch FP, straight gaps G extending in the X direction are formed between the adjacent fins 3, when viewed from the upstream side to the downstream side. For this reason, it is preferable that the height H of the protrusions 5 be less than the fin pitch FP, from the viewpoint of reducing pressure loss. Taking the balance between improvement in heat transfer characteristics and reduction in pressure loss into consideration, it is preferable that the height H of the protrusions 5 be equal to or greater than 0.2 times the fin pitch FP (more preferably equal or greater than 0.25 times) but less than the fin pitch FP.

[0029] Next, the flow of the air in the present heat exchanger 1 will be discussed.

[0030] As illustrated in Fig. 5 (see also Fig. 6), airflow A1 coming from the front of the fins 3 collides against the protrusions 5. Subsequently, partial airflow A2, after the collision, is guided toward the first heat transfer tube 2A side by the first slanted surface 6a, while the other airflow A2' is guided toward the second heat transfer tube 2B side by the second slanted surface 6b. Then, the airflow A2 guided by the first slanted surface 6a flows around to the rear of the first heat transfer tubes 2A. Also, the airflow A2' guided by the second slanted surface 6b flows around to the rear of the second heat transfer tubes 2B. As a result, the degradation of the heat transfer coefficient is hindered and the area of the dead fluid zone is reduced at the rear of the first heat transfer tubes 2A and the second heat transfer tubes 2B of the fin 3.

[0031] Next, airflow A3 that has flowed around to the rear of the first heat transfer tube 2A collides against a protrusion 5 in the second row, and it is guided toward a heat transfer tube 2C side in the second row by the second slanted surface 6b of the protrusion 5. Likewise, airflow A3' that has flowed around to the rear of the second heat transfer tube 2B collides against a protrusion 5 in the second row, and it is guided toward a heat transfer tube 2C side in the second row by the first slanted surface 6a of the protrusion 5. Then, airflows A4 and A4' guided by the slanted surfaces 6a and 6b flow around to the rear of the heat transfer tube 2C. As a result, in the rear of the heat transfer tube 2C of the fin 3 as well, the heat transfer coefficient is hindered from degrading, and the area of the dead fluid zone is reduced.

[0032] Table 1 shows simulation results in which the fin-tube heat exchanger according to the present embodiment (see Fig. 7 for the specific configuration) is compared with a fin-tube heat exchanger having a conventional corrugated fin (a fin bent in a wave-like form. For example, see Figs. 1 and 2 in JP 1-90995 A). In this simulation, the thickness of the fin was 0.1 mm, the fin pitch was 1.5 mm, the outer diameter of the heat transfer tubes was 7.0 mm, and the front velocity V_{air} was 1 m/s. In Table 1, "circular conic fin" refers to a fin-tube heat exchanger according to the later-described Embodiment 2.

[TABLE 1]

						(Ratio to the corrugated fin)			
	Fin No.	Vair (m/s)	H (mm)	L (mm)	Occupied area (%)	α (%)	ΔP (%)	L/H	$\alpha/\Delta P$
Corrugated fin (Conventional)	No. 1	1				100	100		
Tetrafin (Embodiment1)	No. 2	1	2.235	6.85	27	103.4	124.9	3.06	0.83
	No. 3	1	0.765	13.7	55	101.1	73.1	17.91	1.38
	No. 4	1	1.49	13.7	55	101.9	79.6	9.19	1.28
	No. 5	1	2.235	13.7	55	102.3	90.9	6.13	1.13
	No. 6	1	1.49	18.2	73	105.3	84.8	12.21	1.24
	No. 7	1	2.235	18.2	73	101.9	84.4	8.14	1.21
	No. 8	1	2.63	18.2	73	102.3	98.3	6.92	1.04
Circular conic fin (Embodiment2)	No. 9	1			43	100.8	87.9		1.15

[0033] In evaluating the performance of a heat exchanger, it is more preferable that heat transfer coefficient α be greater, while it is more preferable that pressure loss ΔP be less. That is, it is more preferable that $\alpha/\Delta P$ be greater. As seen from Table 1, the fin-tube heat exchanger of the present embodiment shows that the longer the airflow-wise length L of the protrusions 5, the greater the $\alpha/\Delta P$, while the higher the height H of the protrusions 5, the less the $\alpha/\Delta P$. In other words, the greater the parameter L/H is, the greater the $\alpha/\Delta P$. Fig. 8 shows a graph in which the parameter L/H is plotted along the horizontal axis while the parameter $\alpha/\Delta P$ (the ratio with respect to the conventional fin) is plotted along the vertical axis. As seen from this graph, when the parameter L/H is greater than 5.5, the value $\alpha/\Delta P$ becomes higher than that for the conventional corrugated fin type heat exchanger. Therefore, it is preferable that the parameter L/H be greater than 5.5.

[0034] As discussed above, in the fin-tube heat exchanger 1 according to the present embodiment, each of the fins 3 has the protrusions 5 in a quadrangular pyramidal shape between the heat transfer tubes 2A and 2B, and the protrusions 5 are formed so as to divide the air toward the one heat transfer tube 2A side and the other heat transfer tube 2B side. Specifically, each of the protrusions 5 has the first slanted surface 6a for guiding the air toward the one heat transfer tube 2A side and the second slanted surface 6b for guiding the air toward the other heat transfer tube 2B side. In addition, the upstream edge 8a of the protrusion 5 is located upstream of the centers C of the heat transfer tubes 2A and 2B. Thus, the air starts to be guided in a region upstream of the centers C of the heat transfer tubes 2A and 2B, so the flow directions change at a relatively early stage. As a result, the air tends to flow around to the rear of the heat transfer tubes 2A and 2B easily. Thus, according to the present embodiment, the dead fluid zone can be reduced.

[0035] Moreover, the intermediate portion 8b, which is the widest portion of the protrusion 5, is located upstream of the downstream edges 2e of the heat transfer tubes 2A and 2B. This also serves to allow the air to flow around to the rear of the heat transfer tubes 2A and 2B more easily, thus the dead fluid zone reduces.

[0036] Furthermore, in the present embodiment, after the air is divided by the protrusion 5 toward the one heat transfer tube 2A side and toward the other heat transfer tube 2B side, the flow of the air is accelerated in the spaces between the protrusion 5 and each of the heat transfer tubes 2A and 2B. Therefore, the heat transfer coefficient of the fin 3 improves corresponding to the acceleration of the flow of the air.

[0037] In addition, the accelerated air collides against a protrusion 5 provided at a more downstream side. As a result, the thermal boundary layer becomes thinner at the protrusion 5 in the downstream side. Accordingly, the heat transfer coefficient at the protrusions 5 of the downstream side improves, leading to an improvement in the heat transfer coefficient of the fin 3 as a whole.

[0038] In addition, the width of each of the protrusions 5 increases from the upstream edge 8a to the intermediate portion 8b, while it decreases from the intermediate portion 8b to the downstream edge 8c. Thus, the protrusions 5 are configured so that they do not narrow the flow passage of the air after the air is guided toward the heat transfer tube 2A and 2B sides by the portion from the upstream edge 8a to the intermediate portion 8b (the first slanted surface 6a and the second slanted surface 6b). Accordingly, by the protrusions 5 of the present embodiment, the pressure loss can be prevented from becoming too large.

[0039] In the present heat exchanger 1, the protrusions 5 are disposed in a relatively upstream side. Therefore, as illustrated in Fig. 9, the thermal boundary layer BL that develops from the leading edge of the fin 3 collides against the protrusion 5 before it develops considerably (before the thermal boundary layer BL becomes thick). As a result, the

thermal boundary layer at the protrusion 5 becomes thin, and the heat transfer coefficient of the protrusion 5 becomes high. In other words, the present heat exchanger 1 can improve the heat transfer coefficient of the protrusions 5 since the protrusions 5 are disposed in the upstream side. In this respect too, the present heat exchanger 1 achieves an improvement in the heat transfer performance.

[0040] In the present heat exchanger 1, the equivalent diameter d of the protrusions 5 is equal to or greater than the outer diameter D of the heat transfer tubes 2, which means that the protrusions 5 is formed to be relatively large. Therefore, the flow direction can be changed in a relatively large scale. Accordingly, it is possible to guide the air to the rear of the heat transfer tubes 2 desirably even when the flow velocity of the air is relatively small (for example, when the front velocity is less than 2 m/s) and even when it is particularly small (for example, when the front velocity is less than 1 m/s). The present heat exchanger 1 can exhibit good heat transfer characteristics even for the airflow in a laminar flow condition. Moreover, since the protrusions 5 are formed to be relatively large in this way, the air can be accelerated greatly locally between the protrusions 5 and the heat transfer tubes 2, so that the heat transfer coefficient can be improved.

[0041] From the viewpoint of making the protrusions 5 large, it is preferable that the occupied area of the protrusions 5 in the entire fin 3 (excluding the portion of cross-sectional area of the heat transfer tubes) be made large to a certain degree. For this reason, the occupied area of the protrusions 5 may be, for example, equal to or greater than the occupied area as in the above-described simulation model (30%) but less than the maximum possible value at which the protrusions 5 can be placed in between the heat transfer tubes 3 (for example, 75%). More desirably, when the occupied area is from 43% to 73%, as is shown in Table 1, it is appropriate because the $\alpha/\Delta P$ value will be 1 or greater.

[0042] Furthermore, in the present embodiment, the protrusions 5 are formed in a quadrangular pyramidal shape, and therefore, the flow direction of the air can be changed relatively abruptly at the first slanted surface 6a and the second slanted surface 6b. As a result, it becomes possible to guide the air to the rear of the heat transfer tubes 2 more efficiently.

[0043] In addition, the surface area of the fin 3 is greater in the present embodiment by the area of the protrusions 5 than in the case in which the entire surface of the fin is flat. Thus, the amount of heat exchange can be increased because of the enlargement of the heat transfer area. The amount of the increased heat transfer area may be, but is not limited to, from 3% to 5%, for example.

[0044] When the present heat exchanger 1 is used as a condenser for cooling the air (for example, an evaporator in a refrigeration cycle apparatus), dew condensation may take place on the surfaces of the fins 3. Also, when the present heat exchanger 1 is installed in an outdoor unit in a cold area, frost formation may take place on the surfaces of the fins 3. However, in the present heat exchanger 1, the portion of the fin 3 other than the protrusions 5 is a flat surface. For this reason, water drops formed due to dew condensation or after defrosting tend not to stay on the surface of the fin 3 and tend to fall down more quickly in comparison with what is called a slit fin. Therefore, the present heat exchanger 1 can exhibit an excellent advantageous effect also as a condenser.

[0045] In the present embodiment, the protrusions 5 are configured to protrude from one surface of the fin 3. However, it is also possible that some of the protrusions 5 are configured to protrude from one surface of the fin 3 while the other protrusions 5 are configured to protrude from the other surface of the fin 3. For example, a plurality of protrusions 5 arranged in a row direction may be configured to protrude alternately from the obverse side and the reverse side of the fin 3.

[0046] The airflow-wise length of each protrusion 5 is not particularly limited. For example, when the airflow-wise length of the fins 3 is 36 mm, the length of the protrusions 5 may be set to be equal to or greater than 4.5 mm but less than 36 mm.

[0047] The shape of each of the protrusions 5 is not limited to a quadrangular pyramidal shape. The shape of the protrusion 5 may be other pyramidal shapes, such as a triangular pyramidal shape, as long as it is possible to obtain a significant effect shown in the present specification, such as the effect of guiding the air A toward the first heat transfer tube 2A and the second heat transfer tube 2B or the effect of hindering the dead fluid zone from developing.

[0048] In addition, a fin 43 as shown in Fig. 13 may be proposed as one modified example of the fin according to Embodiment 1. Protrusions 45 of the fin 43 shown in Fig. 13 are such that the outer shape 45s of each of the protrusions 45 shows a rhombus when the fin 43 is viewed in plan. It is preferable that each of the protrusions 45 be in a quadrangular pyramidal shape composed of four slanted surfaces. The orientation of the protrusions 45 is determined so that the longer one of the two diagonal lines of the rhombic outer shape 45s is parallel to the X direction and the shorter diagonal line is parallel to the Z direction. The rest of the configurations of the protrusions 45 are common to those of the protrusions 5 of Embodiment 1.

[0049] Furthermore, the protrusions having other shapes described in the following embodiments may be employed suitably likewise.

(Embodiment 2)

[0050] As illustrated in Figs. 10 and 11, a fin 13 according to the present embodiment has protrusions 15 formed in a circular conic shape. In the present embodiment, the protrusion 15 has no clear ridge line. However, assuming a virtual line 7a extending from an upstream edge 8a to an apex 11 in the X direction and a virtual line 7b extending through the

apex 11 in the Z direction, it is understood that a first slanted surface 6a, which guides the air toward the first heat transfer tube 2A side, and a second slanted surface 6b, which guides the air toward the second heat transfer tube 2B side, are formed between the virtual line 7a and the virtual line 7b.

[0051] In the present embodiment as well, the width of each of the protrusions 15 increases from the upstream edge 8a to the intermediate portion 8b, while it decreases from the intermediate portion 8b to the downstream edge 8c. The upstream edges 8a of the protrusions 15 are located upstream of the centers C of the heat transfer tubes 2A and 2B. The intermediate portions 8b of the protrusions 15 are located upstream of the downstream edges 2e of the heat transfer tubes 2A and 2B. The diameter d of the protrusions 15 is equal to or greater than the diameter D of the heat transfer tubes 2.

[0052] The height of the protrusions 15 may be either greater or less than the fin pitch. The height of the protrusions 15 may be equal to the fin pitch.

[0053] The rest of the configurations are the same as those in Embodiment 1 and the description thereof will be omitted.

[0054] In the present embodiment as well, the area of the dead fluid zone at the rear of the heat transfer tubes 2 becomes smaller, as in Embodiment 1. As a result, the heat transfer characteristics can be improved. Moreover, according to the present embodiment, the air can be guided relatively gently toward the first heat transfer tube 2A side and toward the second heat transfer tube 2B side because the first slanted surface 6a and the second slanted surface 6b are curved surfaces.

(Embodiment 3)

[0055] As illustrated in Fig. 12, a fin 23 according to Embodiment 3 has protrusions 25 that are formed in an elliptic conic shape. Herein, the ellipticity (the ratio of the major axis to the minor axis) is set at about 2. However, the ellipticity of the protrusions 25 is not particularly limited. The ellipticity may be greater than 1 but equal to or less than 2, or it may be equal to or greater than 0.5 but less than 1. The protrusions 25 may have an elliptic conic shape that is slender in the X direction or an elliptic conic shape that is slender in the Z direction.

[0056] In the present embodiment as well, the protrusion 25 has no clear ridge line. However, as in the case of Embodiment 2, assuming a virtual line 7a extending from an upstream edge 8a to an apex 11 in the X direction and a virtual line 7b extending through the apex 11 in the Z direction, it is understood that a first slanted surface 6a, which guides the air toward the first heat transfer tube 2A side, and a second slanted surface 6b, which guides the air toward the second heat transfer tube 2B side, are formed between the virtual line 7a and the virtual line 7b.

[0057] In the present embodiment as well, the width of each of the protrusions 25 increases from the upstream edge 8a to the intermediate portion 8b, while it decreases from the intermediate portion 8b to the downstream edge 8c. The upstream edges 8a of the protrusions 25 are located upstream of the centers C of the heat transfer tubes 2A and 2B. The intermediate portions 8b of the protrusions 25 are located upstream of the downstream edges 2e of the heat transfer tubes 2A and 2B. The equivalent diameter d of the protrusions 25 is equal to or greater than the diameter D of the heat transfer tubes 2. The height of the protrusions 25 may be either greater or less than the fin pitch, or may be equal to the fin pitch.

[0058] The rest of the configurations are the same as those in Embodiment 1 and the description thereof will be omitted.

[0059] In the present embodiment as well, the area of the dead fluid zone at the rear of the heat transfer tubes 2 becomes smaller and the heat transfer characteristics improve, as in Embodiment 1. Also, as in Embodiment 2, the air can be guided relatively gently toward the first heat transfer tube 2A side and the second heat transfer tube 2B side because the first slanted surface 6a and the second slanted surface 6b are curved surfaces. Furthermore, according to the present embodiment, the degree of guiding the air to the first heat transfer tubes 2A and the second heat transfer tubes 2B can be set as appropriate by changing the ellipticity of the protrusions 25 appropriately. Thus, by appropriately setting the ellipticity of the protrusions 25 according to the conditions of use of the heat exchanger 1, the heat transfer characteristics can be more finely adjusted or optimized.

[0060] Next, embodiments of the fins in which the protrusions are formed in a circular hump shape or in an elliptical hump shape will be discussed.

(Embodiment 4)

[0061] Fig. 14 is a plan view illustrating a fin according to Embodiment 4. Protrusions 35 having an elliptical hump shape are formed on the surface of the fin 30. All the protrusions 35 protrude in the same direction from one side of the fin 30. Also, each of the protrusions 35 is located between the first heat transfer tube 2A and the second heat transfer tube 2B, which are two adjacent heat transfer tubes in the same row. In other words, the heat transfer tubes 2 and the protrusions 35 appear alternately along the row direction. When the protrusion 35 is projected orthogonally onto a virtual plane parallel to the fin 30, the image appearing on the virtual plane is in an elliptical shape. In addition, the area of the image of the protrusion 35 appearing on the virtual plane is greater than the cross-sectional area of the heat transfer tube 2 when cross-sectioned in a radial direction perpendicular to the lengthwise direction. In other words, in Fig. 14,

which is a plan view showing the fin 30 viewed in plan from the height direction (Y direction), which is perpendicular to the principal surface, the outer shape 5s of the protrusion 35 is in an elliptical shape and, at the same time, the equivalent diameter d (the equivalent diameter d is defined by the equation $\pi d^2/4 = S$ (area)) of the protrusions 35 is greater than the outer diameter D of the heat transfer tubes 2. For example, the dimensions of the protrusions 35 may be adjusted so that the major axis d_1 of the outer shape 5s that is in an elliptical shape is greater than the outer diameter D of the heat transfer tubes 2 and that the minor axis d_2 is within the range $d_2 \leq d_1 \leq 2d_2$ with respect to the major axis d_1 . The above-mentioned cross-sectional area of the heat transfer tubes 2 is in agreement with the area of the opening of the through holes 3h, which are formed in the fin 30 to place the heat transfer tubes 2 therein.

[0062] In addition, as illustrated in Fig. 14, only one protrusion 35 is formed in between each of the first heat transfer tubes 2A and each the second heat transfer tubes 2B. In other words, the protrusions 35 and the through holes 3h are formed alternately along the row direction. The arrangement of the protrusions 35 is in a staggered manner such that the protrusions 35 meander through the through holes 3h. For example, when a large number of small protrusions are formed between two adjacent heat transfer tubes in such a manner as disclosed in JP 7-239196 A, it is difficult to gain a sufficient height for the protrusions because of limitations in the process. Such small protrusions are weak in terms of the effect of guiding the air A. In addition, the protrusions with an insufficient height yield a low rate of increase of the heat transfer area relative to an unprocessed flat plate, and they are unlikely to achieve a significant effect of hindering development of the boundary layer. In contrast, the protrusions 35 according to the present embodiment make it possible to gain a sufficient height H and therefore achieve a strong effect of guiding the air A toward the heat transfer tubes. Moreover, it is possible to enhance the rate of increase of the heat transfer area relative to an unprocessed flat plate relatively easily. At the same time, the effect of hindering development of boundary layer is strong. Thus, sufficient improvement in heat transfer performance can be expected.

[0063] As has been described earlier, the heat transfer tubes 2 are disposed in a staggered manner in two rows, one of the two rows being the front row that is closer to the leading edge 30p of the fin 30 and the other row being the rear row that is parallel to the front row. Other protrusions 35 are formed between two adjacent heat transfer tubes 2, 2 disposed in the rear row, the other protrusions having the same shape and the same dimensions as those of the protrusions 35 that are formed between two adjacent heat transfer tubes 2, 2 disposed in the front row. Thereby, the effect of improving the heat transfer coefficient can be expected in the rear row as well as in the front row.

[0064] It is preferable that the location and orientation of the protrusions 35 be determined in the following manner. As illustrated in Fig. 14, the upstream edge 5f of each of the protrusions 35 is located closer to a leading edge 30p (the outer edge on the upstream side) of the fin 30 than upstream edges 2f of the heat transfer tubes 2A and 2B, with respect to the direction perpendicular to the principal surface of the fin 30 and the row direction (along the X direction). In this way, the air A that has reached the leading edge 30p of the fin 30 can be guided toward the heat transfer tubes 2 swiftly, which is advantageous in improving the heat transfer performance. On the other hand, the downstream edge 5e of each of the protrusions 35 is more distant from the leading edge 30p of the fin 30 than the centers C1 and C2 of the heat transfer tubes 2A and 2B; i.e., it is located on the downstream side with respect to the flow direction of the air A. In this way, the air A that flows on the principal surface of the fin 30 can be guided to the rear of the heat transfer tubes 2A and 2B efficiently. Thus, it is possible to hinder the formation of the dead fluid zone, which does not contribute to heat transfer. Preferably, the downstream edges 5e of the protrusions 35 should be located downstream of the downstream edges 2e of the heat transfer tubes 2A and 2B, with respect to the flow direction of the air A, as in the present embodiment.

[0065] In addition, the orientation of the protrusions 35, the outer shape 5s of which is in an elliptical shape, are determined so that the minor axis of the ellipse is parallel to the row direction (Z direction) in which the first heat transfer tubes 2A and the second heat transfer tubes 2B are arranged. In other words, the flow direction of the air A and the major axis of the ellipse are parallel to each other. In this way, the air A can be guided more smoothly to the left and right of the protrusions 35, and the degree of increase of the pressure loss originating from the formation of the protrusions 35 can be lessened. Of course, it is also possible to set the major axis of the ellipse to be along a direction parallel to the row direction.

[0066] In addition, each of the protrusions 35 is formed at a location equidistant from the center C1 of the first heat transfer tube 2A and the center C2 of the second heat transfer tube 2B. In other words, the location of the protrusion 35, relative to the first heat transfer tube 2A and the second heat transfer tube 2B, is determined so that the major axis of the image of the elliptical shape in a plane on which the protrusion 35 is orthogonally projected is contained in a virtual plane MD that perpendicularly bisects the line segment C1C2 connecting the center C1 of the first heat transfer tube 2A and the center C2 of the second heat transfer tube 2B at the shortest distance. This makes it possible to allow the air A to flow along both the flat region between the protrusion 35 and the first heat transfer tube 2A and the flat region between the protrusion 35 and the second heat transfer tube 2B uniformly. In other words, both the first heat transfer tube 2A and the second heat transfer tube 2B are allowed to contribute to heat transfer equally, and in this case, the heat transfer performance of the fin-tube heat exchanger 1 can be maximized.

[0067] As illustrated in Fig. 15, the height H of the protrusions 35 is adjusted so as to satisfy the expression $(FP/4) \leq H \leq FP$, where FP is the fin pitch defined as the distance between the centers of the fins 30 with respect to the thickness

direction. The fins 30 are arranged so that the locations of the protrusions 35 formed match between the fins with respect to the height direction. If the height H of the protrusions 35 is less than the fin pitch FP, straight gaps G extending in the X direction are formed between one of the fins 30 and another fin 30 that are adjacent to each other. For this reason, it is preferable that the height H of the protrusions 35 be less than the fin pitch FP, from the viewpoint of reducing pressure loss. On the other hand, it is preferable that the height H of the protrusions 35 be greater, from the viewpoint of improving the heat transfer performance by gaining a sufficient heat transfer area. Thus, the heat transfer performance and the pressure loss are in a trade-off relationship. However, by adjusting the height H of the protrusions 35 within the foregoing range, it is possible to maximize the effect of improving the heat transfer characteristics while minimizing the increase in pressure loss.

[0068] In addition, the height H of each of the protrusions 35 monotonously increases toward the apex TP₁. The apex TP₁ corresponds to the center of the ellipse when the protrusion 35 is viewed in plan. By employing such a shape, the air A is allowed to flow toward the apex TP₁ smoothly, and therefore, an increase in pressure loss can be hindered.

[0069] There are several preferable examples of the surface shape of the protrusion 35. First, the cross-sectional view of the fin 30 shown in Fig. 15 represents the XY cross section that is perpendicular to the principal surface of the flat region of the fins 30 and contains the major axes of the protrusions 35. The XY cross section is also a cross section that is parallel to the flow direction of the air A and is perpendicular to the principal surface of the fin 30. The shape of the protrusion 35 may be adjusted so that the surface 5p (the outer circumferential surface) thereof forms a curve in this XY cross section. The curve may be, for example, a sine curve. In the specific example shown in Fig. 16A, the surface 5p of the protrusion 35 forms a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-90^\circ \leq X \leq 90^\circ$ }.

[0070] It is also possible that the shape of the protrusion 35 may be adjusted so that the surface 5p thereof forms a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-180^\circ \leq X \leq 180^\circ$ }, as shown in Fig. 16B. That is, the surface 5p of the protrusion 35 connects seamlessly with the principal surface of the flat region in the fin 30 at the -180° location, which corresponds to the upstream edge 5f. When such a surface shape is employed, the flow velocity of the air A that flows over the protrusion 35 does not tend to decrease; therefore, the height H of the protrusion 35 is gained sufficiently while inhibiting an increase in pressure loss.

[0071] Another example of the curve that can make the bending seamless is a clothoid curve. It is possible to employ the clothoid curve for the surface shape of the protrusion 35. That is, the shape of the protrusion 35 may be adjusted so that the surface 5p thereof forms a clothoid curve in the XY cross section. Fig. 17A shows a clothoid curve.

[0072] Generally, the way of bending of a curve is represented by a circle of curvature. A curve in which the way of bending does not leap but changes seamlessly from small to large or from large to small is best suited to the line for an expressway. One of the best examples of such a curve is "clothoid." The radius r of the circle of curvature of a clothoid is inversely proportional to the travel distance (the distance s from the origin in Fig. 17 A) on the curve. Specifically, the curve defined by the following polar equation (1) is a clothoid.

$$r = a^2/s \text{ (a: constant)...(1)}$$

[0073] Although it has been described that the surface 5p may form a clothoid curve in the XY cross section, one single clothoid curve does not fit the surface 5p of the protrusions 35. For this reason, the shape of the protrusion 35 may be adjusted so that the ascending section of the surface 5p appearing in the XY cross section, which is from the upstream edge 5f to the apex TP₁, is divided into a plurality of sections and each of the divided sections forms a clothoid curve. It is recommended that adjustment should be done so that the radius of the circle of curvature changes seamlessly at the boundaries of the sections. It is recommended that the descending section from the apex TP₁ to the downstream edge 5e should be symmetrical to the ascending section. In this way, the entire surface 5p forms a clothoid curve in the XY cross section.

[0074] Alternatively, the shape of the protrusion 35 may be adjusted so that a portion of the surface 5p forms a clothoid curve while the rest of the surface forms another type of curve, such as a circular arc. For example, as illustrated in the XY cross section in Fig. 17B, the section in which a clothoid curve is employed is restricted to 5/10 of the height, i.e., from the location corresponding to the upstream edge 5f to the location corresponding to the midpoint of the height H. The upper half, from 5/10 to 10/10, of the height, i.e., from the location corresponding to the midpoint of the height H to the apex TP₁, is configured to form a circular arc. As long as the flat region of the fin 30 and the base of the protrusion 35 are connected smoothly, the effect of hindering an abrupt decrease in the flow velocity at the upstream edge 5f can be obtained when the air A reaches the upstream edge 5f. The clothoid curve section shown in Fig. 17B may be other easement curves (the curves in which the radius of the circle of curvature changes seamlessly), such as a sine curve, as mentioned earlier, or a cubical parabola.

[0075] Of course, there is an advantage for the case in which the shape of the protrusion 35 is adjusted so that the curve formed by the surface 5p contains no inflection point between the upstream edge 5f and the apex TP₁ in the XY

cross section. Although excellent performance is obtained in the case in which an inflection point is contained (Fig. 16B), there is an advantage for the case in which no inflection point is contained (Fig. 16A) in that it can be pressed easily, in other words, production is easier. For example, the surface 5p of the protrusion 35 may be such that the entire section, which is from the location corresponding to the upstream edge 5f through the apex TP_1 to the downstream edge 5e, can be represented by a circular arc in the XY cross section.

[0076] It is desirable that the shape of the protrusion 35 in the cross section that contains the minor axis and is perpendicular to the principal surface of the fin 30, i.e., in the YZ cross section, be adjusted so that the surface 5p forms an easement curve, such as a sine curve or a clothoid curve. More preferably, the shape of the protrusion 35 should be adjusted so that the surface 5p forms an easement curve in an arbitrary cross section that contains the apex TP_1 and is perpendicular to the principal surface of the fin 30. In this way, the effect of hindering a decrease in flow velocity can be maximized, and at the same time, the air A that has reached the protrusions 35 can be guided more smoothly toward the heat transfer tubes 2.

[0077] Thus, the shape of the protrusion 35 may be adjusted so that the surface 5p forms a curve containing an inflection point between the upstream edge 5f and the apex TP_1 in a cross section that is perpendicular to the principal surface of the fin 30 and contains the minor axis or the major axis of the ellipse. When such a configuration is employed, the effect of hindering a decrease in the flow velocity at the protrusions 35 can be expected. It is also possible to adjust the shape of the protrusion 35 so that the surface 5p forms a curve containing an inflection point in an arbitrary cross section that is perpendicular to the principal surface of the fin 30 and contains the apex TP_1 .

[0078] On the other hand, when the surface 5p is configured to form a curve containing no inflection point, production of the fin 30 is easy. That is, it is also possible to adjust the shape of the protrusion 35 so that the surface 5p forms a curve containing no inflection point in an arbitrary cross section that contains the apex TP_1 and is perpendicular to the principal surface of the fin 30.

[0079] Next, the workings of the fin-tube heat exchanger 1 according to the present embodiment will be described below.

[0080] As illustrated in Fig. 18, the air A that has reached the leading edge 30p of the fin 30 is guided onto the principal surface of the fin 30, i.e., to the fin-tube heat exchanger 1, from the direction parallel to the principal surface of the fin 30 and perpendicular to the longitudinal direction of the fin 30. Since the heat transfer tubes 2 are disposed penetrating the fin 30, the air A flows so as to avoid the heat transfer tubes 2. In addition, since the protrusions 35 are formed in the fin 30, the air A tends to flow so as to avoid the protrusions 35. In other words, the protrusions 35 guide the air A toward the heat transfer tubes 2. As a result, airflow AF with an increased flow velocity is produced between the protrusions 35 and the heat transfer tubes 2. When the flow velocity increases, the heat transfer coefficient becomes high. In particular, the heat transfer coefficient increases in the vicinity of the side face of the heat transfer tube 2, which is indicated by the dashed line in Fig. 18. Therefore, it becomes possible to transfer the heat of the refrigerant flowing in the heat transfer tubes 2 to the air A efficiently. In addition, the heat transfer performance of the fin-tube heat exchanger 1 is enhanced due to the following effects: the leading edge effect resulting from the collision of the air A with the upstream edges 5f of the protrusions 35, the effect of hindering development of a boundary layer at the protrusions 35, and the effect of reducing the dead fluid zone DS due to the flow of the air toward the rear of the heat transfer tubes 2. Moreover, the protrusions 35 are adjusted to be in a shape such as to hinder an increase in pressure loss, as already described above. According to the result of a computer simulation carried out by the present inventors, when the pressure loss in the heat exchanger employing a conventional corrugated fin is 1, the pressure loss in the heat exchanger according to the present embodiment is 0.91, about 10% less.

[0081] On the other hand, a fin 203 in which two protrusions 205a and 205b having an elliptical hump shape are formed between a first heat transfer tube 2A and a second heat transfer tube 2B, as illustrated in Fig. 19, is discussed for comparison with the present invention. The equivalent diameter (the diameter of the circle that has the same area as the protrusion) of each of the protrusions 205a and 205b is smaller than the outer diameter D of the heat transfer tubes 2. One protrusion 205a guides the air A toward the first heat transfer tube 2A and produces airflow AF_1 with an increased flow velocity in a region between the protrusion 205a and the first heat transfer tube 2A. Likewise, the other protrusion 205b guides the air A toward the second heat transfer tube 2B and produces airflow AF_2 with an increased flow velocity in a region between the protrusion 205b and the second heat transfer tube 2B. In addition, airflow AF_3 with an increased flow velocity is produced also between the one protrusion 205a and the other protrusion 205b.

[0082] However, because the airflow AF_3 between the protrusion 205a and the protrusion 205b is relatively away from the heat transfer tubes 2A and 2B, the airflow AF_3 does not contribute to improvements in the heat transfer performance as much as the flows AF_1 and AF_2 that are closer to the heat transfer tubes 2A and 2B. If this is the case, it is believed more effective to form a large protrusion 35 as in the present invention than to form two protrusions 205a and 205b.

(Embodiment 5)

[0083] A fin 31 shown in the plan view of Fig. 20 may be suitably employed for the fin-tube heat exchanger 1. The arrangement and dimensions of the heat transfer tubes 2 are common to those in Embodiment 4. The difference is that

protrusions 51 having a circular hump shape are employed in place of the protrusions 35 having an elliptical hump shape.

[0084] As illustrated in Fig. 20, the protrusions 51 formed on the fin 31 have an outer shape 51s in a circular shape in plan view. Specifically, when a protrusion 51 is orthogonally projected onto a virtual plane parallel to the fin 31, the image appearing on the virtual plane is in a circular shape. In addition, the diameter d_3 of the circle formed by the outer shape 51s of the protrusion 51 is greater than the outer diameter D of the heat transfer tubes 2.

[0085] The protrusion 51 in a circular hump shape is free from the issue of orientation, unlike the protrusion 35 in an elliptical hump shape (Fig. 14), but the location may be determined in the same manner as in the case of the elliptical hump. That is, the protrusion 51 may be formed at a location equidistant from the center $C1$ of the first heat transfer tube 2A and the center $C2$ of the second heat transfer tube 2B. Specifically, the location of each of the protrusions 51, relative to the first heat transfer tubes 2A and the second heat transfer tubes 2B, may be determined so that apex TP_2 overlaps with a virtual plane MD that perpendicularly bisects the line segment $C1C2$ connecting the center $C1$ of the first heat transfer tube 2A and the center $C2$ of the second heat transfer tube 2B at the shortest distance. In addition, it is preferable that the upstream edges 51f of the protrusions 51 be located upstream of the upstream edges 2f of the heat transfer tubes 2 and that the downstream edges 51e of the protrusions 51 be located downstream of the downstream edges 2e of the heat transfer tubes 2. These points are common to the case of the protrusion 35 described in Embodiment 4.

[0086] The height and the surface shape of the protrusion 51 are also the same as in the case of the protrusion 35 described in Embodiment 4. For example, the shape of the protrusion 51 may be adjusted so that the surface 51p forms an easement curve, such as a sine curve (see Figs. 16A and 16B) or a clothoid curve (see Fig. 17B), in the XY cross section. Also, the shape of the protrusion 51 may be adjusted so that the surface 51p forms a curve containing an inflection point between the upstream edge 51f and the apex TP_2 . Of course, it is possible to configure the curve formed by the surface 51p so that it contains no inflection point between the upstream edge 51f and the apex TP_2 .

[0087] In Embodiments 4 and 5 as well, when assuming a virtual line extending in the X direction from the upstream edge 5f, 51f of the protrusion 35, 51 and a virtual line extending in the Z direction through the apex TP_1 , TP_2 of the protrusions 35, 51, a first slanted surface that guides the air toward the first heat transfer tube 2A and a second slanted surface that guides the air toward the second heat transfer tube 2B are formed between the two virtual lines, as in Embodiment 1.

[0088] As illustrated in Figs. 14 and 20, the width of the protrusions 35, 51 increases from the upstream edges 5f, 51f to the intermediate portions 5b, 51b, while it decreases from the intermediate portions 5b, 51b to the downstream edges 5e, 51e. The upstream edges 5f, 51f of the protrusions 35, 51 are located upstream of the centers $C1$ and $C2$ of the heat transfer tubes 2A and 2B. The intermediate portions 5b, 51b of the protrusions 35, 51 are located upstream of the downstream edges 2e of the heat transfer tubes 2A and 2B. The equivalent diameter d of the protrusions 35, 51 is equal to or greater than the diameter D of the heat transfer tubes 2. The height of the protrusions 35, 51 may be either greater or less than the fin pitch, or it may be equal to the fin pitch. With such a configuration, the same advantageous effects as in Embodiment 1 can be obtained.

(Embodiment 6)

[0089] A fin 32 shown in the plan view of Fig. 21 suitably may be employed for the fin-tube heat exchanger 1. The arrangement and dimensions of the heat transfer tubes 2 are common to those in Embodiment 4. The difference is that second protrusions 53 are formed between the protrusions 35 formed in the front row and the other protrusions 35 formed in the rear row, the second protrusions 53 having a surface area smaller than the foregoing protrusions 35, 35. Strictly speaking, in Fig. 21, which is a plan view showing the fin 32 viewed in plan from the height direction (Y direction) perpendicular to the principal surface, the diameter d_4 of the second protrusions 53 is smaller than the outer diameter D of the heat transfer tubes 2. In addition, the second protrusions 53 protrude in the same direction as the protrusions 35, 35 in the front row and the rear row.

[0090] In the fin 30 according to Embodiment 4 (see Fig. 14), there is a little space between the protrusions 35 in the front row and the protrusions 35 in the rear row. By forming the second protrusions 53 in this space, the heat transfer area increases. In particular, the region in which the second protrusions 53 are formed serves as a passage of the air A whose flow velocity has been increased by the effect of the protrusions 35 in the front row. Therefore, by allowing the air A with an increased flow velocity to hit the second protrusions 53 intentionally, it is possible to improve the heat transfer performance further. Such second protrusions 53 may have a circular hump shape as in the present embodiment, or they may have an elliptical hump shape.

[0091] In addition, a fin 33 as illustrated in Fig. 22 also can be employed suitably for the same reasons as described above. In the fin 33, protrusions 51, 51 in a circular hump shape are formed in the front row and the rear row in place of the protrusions 35 in an elliptical hump, and second protrusions 53 having a smaller surface area than the protrusions 51, 51 are formed between the protrusions 51 in the front row and the protrusions 51 in the rear row.

(Embodiment 7)

[0092] The protrusions 35, 51, and 53 described in Embodiments 4 to 6 are formed so that all the protrusions protrude in the same direction. However, as has been mentioned in Embodiment 1, this is not essential. Specifically, a fin 34 as illustrated in Figs. 23A and 23B may be suitably employed as the fin for the fin-tube heat exchanger 1. In the fin 34, protrusions 35 protruding from a first principal surface 34j side (the obverse side of the fins 34) and protrusions 35' protruding from a second principal surface 34k side (the reverse side of the fin 34) coexist.

[0093] When the protrusions 35, 35' with different protruding directions are formed to coexist as described above, the following effects are achieved. A fin in which all the protrusions protrude in the same direction is produced through the following steps: the step of cutting a metal plate into a predetermined size, the step of forming through holes for accommodating the heat transfer tubes, and the step of forming the protrusions in the metal plate by a pressing process. When the protruding direction of the protrusions is limited to one direction, the metal plate warps during the step of forming the protrusions, resulting in warpage in the fin obtained. If such warpage occurs, there may be cases in which, when assembling the heat exchanger, the fin pitch becomes non-uniform or the heat transfer tubes cannot be inserted smoothly in the through holes because of misalignment of the through holes.

[0094] In contrast, when producing the fin 34 in which the protrusions 35, 35' with different protruding directions are formed, a metal plate that forms the fin 34 is pressed from both sides. Since the pressing is conducted from both sides, the warpage can be balanced between the obverse side and the reverse side, and the warpage can be prevented.

[0095] Regarding the dimensions and locations, the protrusions 35, 35' are made in the same manner as described in Embodiment 4, except that the protruding directions are made different. In addition, it is preferable that the numbers of the protrusions 35, 35' be the same and that they be formed alternately along the row direction. In this case, a high warpage prevention effect can be obtained. Of course, such a configuration may be combined with any other embodiments.

[0096] Further, a fin 36 shown in Figs. 24A and 24B also may be employed suitably as the fin for the fin-tube heat exchanger 1. The fin 36 is an application of the fin 32 described in Fig. 21, in which the protruding direction of second protrusions 53' formed between the protrusions 35 in the front row and the protrusions 35 in the rear row is opposite the protruding direction of larger protrusions 35, 35 in the front and the rear. All the larger protrusions 35, 35 are formed so as to protrude toward a first principal surface 36j side of the fin 36 while smaller second protrusions 53' are formed so as to protrude toward a second principal surface 36k side of the fin 36. In this way as well, the warpage prevention effect can be obtained sufficiently.

[0097] The above-described embodiments 1 through 7 may be combined freely as long as such combinations do not depart from the scope of the present invention. For example, the second protrusions 53 described in Fig. 21 and so forth may be applied to all the other embodiments.

[0098] The fin-tube heat exchanger 1 described above may be applied to a heat pump apparatus for heating or cooling an object such as air or water. As illustrated in Fig. 25, a heat pump apparatus 70 includes: a compressor 71 for compressing a refrigerant; a radiator 72 for cooling the refrigerant compressed by the compressor 71; an expansion valve 73 for expanding the refrigerant that has been cooled by the radiator 72; and an evaporator 74 for evaporating the refrigerant expanded by the expansion valve 73. The compressor 71, the radiator 72, the expansion valve 73, and the evaporator 74 are connected by pipes 75, whereby a refrigerant circuit is formed. In place of the expansion valve 73, a positive displacement-type expander may be employed. The radiator 72 and the evaporator 74 may be constructed to contain the fin-tube heat exchanger 1 according to the present invention.

[0099] The above-described heat pump apparatus 70 may be applied to an air conditioner 80 or a hot water heater 90, as illustrated in Fig. 26. For example, a heat pump air conditioner 80 has an indoor unit 81, which should be installed indoors, and an outdoor unit 82, which forms the refrigerant circuit together with the indoor unit 81 and should be installed outdoors. This air conditioner 80 has the function to both cooling and heating, and the circulation direction of the refrigerant is reversed when switching between cooling and heating. During cooling, the heat exchanger that constitutes the indoor unit 81 serves as an evaporator, and the heat exchanger that constitutes the outdoor unit 82 serves as a radiator. During heating, the heat exchanger that constitutes the indoor unit 81 serves as a radiator, and the heat exchanger that constitutes the outdoor unit 82 serves as an evaporator. When taking the issue of frost formation during the heating into consideration, the fin-tube heat exchanger 1 according to the present invention can be employed suitably as the heat exchanger for the outdoor unit 82. The heat exchanger that constitutes the outdoor unit 82 works as the evaporator 74 of the heat pump apparatus 70 when the heating function is used. Therefore, frost formation tends to occur easily especially in winter. However, the fin-tube heat exchanger 1 according to the present invention is free from the problem of considerable deterioration in the heat exchange efficiency resulting from clogging in the fins due to the frost formation because it does not have raised parts as described with reference to Fig. 37.

[0100] As illustrated in Fig. 26, the heat pump hot water heater 90 has a heat pump unit 91 and a hot water storage tank unit 92. The heat pump unit 91 includes a heat exchanger 73 serving to heat water and heat exchanger 74 for collecting heat from the outdoors. The fin-tube heat exchanger 1 according to the present invention may be employed

suitably for the latter heat exchanger 74.

EXAMPLES

5 **[0101]** Characteristics of fin-tube heat exchangers employing the fin shown in Fig. 20 were studied by a computer simulation. Specifically, a computer simulation was carried out for heat exchangers that employed the fins having protrusions in a circular hump shape (Examples 1 to 3). In addition, the same computer simulation was carried out for a conventional heat exchanger (Comparative Example 1) employing a corrugated fin. The characteristics studied by the computer simulations were flow velocity of the air, Nusselt number, heat transfer coefficient, and pressure loss. The computer simulations were conducted under the following conditions, using "Fluent Ver.6" made by Fluent Asia Pacific Corp.

<Conditions common to Examples 1 to 3 and Comparative Example 1>

15 **[0102]** Fin size: 16.94 mm (air flow direction) x 7.65 mm (row direction)
 Fin thickness: 0.1 mm
 Fin pitch: 1.06 mm
 Outer diameter of the heat transfer tube: 5.0 mm
 Inner diameter of the heat transfer tube: 4.0 mm
 20 Front velocity V_{air} : 1 m/sec.

<Conditions of Example 1>

25 **[0103]** Shape of the protrusion: Circular hump with a cosine curve ($-90^\circ \leq X \leq 90^\circ$)
 Diameter of the protrusion: 6.0 mm
 Height of the protrusion: 1.0 mm

<Conditions of Example 2>

30 **[0104]** Shape of protrusion: Circular hump with a cosine curve ($-180^\circ \leq X \leq 180^\circ$)
 Diameter of the protrusion: 6.0 mm
 Height of the protrusion: 1.0 mm

<Conditions of Example 3>

35 **[0105]** Shape of the protrusion: Circular hump with a clothoid curve
 Diameter of the protrusion: 6.0 mm
 Height of the protrusion: 1.0 mm

40 <Conditions of Comparative Example 1>

[0106] Shape: Corrugated
 Level difference between the ridge and the valley: 1.0 mm

45 **[0107]** The computer simulation results for Examples 1 through 3 and Comparative Example 1 are shown in Figs. 27 to 30 and Table 2. Fig. 27 shows the results for Example 1, Fig. 28 for Example 2, Fig. 29 for Example 3, and Fig. 30 for Comparative Example 1. In each of Figs. 27 to 30, view A shows Nusselt number distribution, and view B shows flow velocity distribution. The white arrow in each figure indicates a flow direction of air.

50

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[TABLE 2]

	Heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)	Pressure loss ΔP (Pa)	Heat transfer coefficient ratio (vs. Corrugated)	Pressure loss ratio (vs. Corrugated)
Example 1	39.9	9.0	1.04	0.95
Example 2	39.7	8.6	1.03	0.90
Example 3	39.5	9.0	1.03	0.94
Comparative Example 1	38.4	9.5	1.00	1.00

[0108] First, as will be appreciated from Fig. 30A, the heat exchanger of Comparative Example 1, which uses the corrugated fin, shows large Nusselt numbers only in the regions near the leading edge of the fin and near the heat transfer tubes. In contrast, as will be appreciated from Figs. 27A to 29A, the heat exchangers of Examples 1 to 3 show large Nusselt numbers not only in the regions near the leading edges of the fins and near the heat transfer tubes but also over the surfaces of the protrusions. This fact proves the results shown in Table 2, which indicates that the heat transfer coefficients of the heat exchangers of Examples 1 to 3 are higher than that of the heat exchanger using a corrugated fin. The Nusselt number is proportional to the heat transfer coefficient, and a greater Nusselt number means a higher heat transfer coefficient.

[0109] In addition, as will be appreciated from Fig. 30B, high flow velocity regions are distributed near the heat transfer tubes such as to be over the ridges in the heat exchanger of Comparative Example 1, which uses the corrugated fin. It is believed that such a flow velocity distribution has the effect of enlarging the regions with large Nusselt numbers toward the downstream side. These points agree with the results for Examples 1 to 3. Specifically, as will be seen from Figs. 27B to 29B, there are high flow velocity regions between the heat transfer tube and the protrusion in the heat exchangers of Examples 1 to 3 as well. Also in the aspect of the air flowing around to the rear of each of the heat transfer tubes sufficiently, Examples 1 to 3 and Comparative Example 1 are generally similar.

[0110] The flow velocity distribution shown in each view B is represented by the values at midpoints between a fin and another fin. When the fin pitches are equal, no significant difference in flow velocity distribution is seen between the corrugated fin and the fins according to the present invention. However, primary factors responsible for the improvement in heat transfer performance are that the boundary layer along the fin surface is thin and that the flow velocity around the heat transfer tubes is great. These two factors are seen from the Nusselt number distribution.

[0111] Thus, the heat exchanger according to the present invention makes it possible to reduce the thickness of the boundary layer over the surfaces of the protrusions and to increase the flow velocity in a region between the protrusion and the heat transfer tube. Thereby, the heat exchanger according to the present invention achieves a superior heat transfer coefficient to the heat exchanger using a corrugated fin. Moreover, as shown in Table 2, each of the heat exchangers of Examples 1 to 3 shows a less pressure loss than the conventional heat exchanger using a corrugated fin.

[0112] Next, similar computer simulations were conducted for a heat exchanger employing the fin shown in Fig. 14 (Example 4), a heat exchanger employing the fin shown in Fig. 22 (Example 5), and a heat exchanger employing a second corrugated fin (Comparative Example 2) under the following conditions.

<Conditions common to Examples 4 and 5 and Comparative Example 2>

[0113] Fin size: 27.0 mm (air flow direction) x 10.5 mm (row direction)

Fin thickness: 0.1 mm

Fin pitch: 1.49 mm

Outer diameter of the heat transfer tube: 7.0 mm

Inner diameter of the heat transfer tube: 5.8 mm

Front velocity V_{air} : 1 m/sec.

<Conditions of Example 4>

[0114] Shape of the protrusion: Elliptical hump with a cosine curve ($-90^\circ \leq X \leq 90^\circ$)

Orientation of the protrusion: The major axis is parallel to the air flow direction

Major axis of the protrusion: 13.0 mm

Minor axis of the protrusion: 10.0 mm

Height of the protrusion: 0.765 mm

<Conditions of Example 5>

[0115] Shape of the first protrusion: Circular hump with a cosine curve ($-90^\circ \leq X \leq 90^\circ$)

Diameter of the first protrusion: 10 mm

Height of the first protrusion: 0.765 mm

Shape of the second protrusion: Circular hump with a cosine curve ($-90^\circ \leq X \leq 90^\circ$)

Diameter of the second protrusion: 5.7 mm

Height of the second protrusion: 0.765 mm

<Conditions of Comparative Example 2>

[0116] Shape: Corrugated

Level difference between the ridge and the valley: 1.49 mm

[0117] The computer simulation results for Examples 4 and 5 as well as Comparative Example 2 are shown in Figs. 31 to 33 and Table 3. Fig. 31 shows the results for Example 4, Fig. 32 for Example 5, and Fig. 33 for Comparative Example 2.

[TABLE 3]

	Heat transfer coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)	Pressure loss ΔP (Pa)	Heat transfer coefficient ratio (vs. Corrugated)	Pressure loss ratio (vs. Corrugated)
Example 4	32.4	6.3	0.99	0.91
Example 5	32.6	6.6	1.00	0.94
Comparative Example 2	32.6	7.0	1.00	1.00

[0118] As will be appreciated from Figs. 33A and 33B, the heat exchanger of Comparative Example 2, which uses a corrugated fin having different dimensions from that of Comparative Example 1, generally shows the same tendency as the heat exchanger of Comparative Example 1. As will be appreciated from Figs. 31A and 31B, the heat exchanger of Example 4, which employs a fin having protrusions in an elliptical hump shape, shows the same tendency as the heat exchangers of Examples 1 to 3. As shown in Table 3, the heat exchanger of Example 4 exhibits almost the same heat transfer coefficient as that of the heat exchanger of Comparative Example 2, but it is superior to the heat exchanger of Comparative Example 2 in terms of pressure loss. However, it is believed certainly possible to improve the heat transfer coefficient to almost the same degree as those of Examples 1 to 3 by adjusting the shape and dimensions of the protrusions.

[0119] On the other hand, as will be appreciated from Figs. 32A and 32B, the heat exchanger of Example 5 has a high flow velocity region between the heat transfer tube and the protrusion in the front row (i.e., the first protrusion) and shows great Nusselt numbers over the surface of the second protrusion. In other words, the effect of reducing the thickness of the boundary layer over the surface of the second protrusion is obtained. As shown in Table 3, the heat exchanger of Example 5 has about the same heat transfer coefficient as that of the heat exchanger of Comparative Example 2, but it is superior to the heat exchanger of Comparative Example 2 in terms of pressure loss. However, it certainly is believed possible to improve the heat transfer coefficient to almost the same degree as those of Examples 1 to 3 by adjusting the shapes and dimensions of the first protrusions and/or the second protrusions.

Claims

1. A fin-tube heat exchanger for exchanging heat between a first fluid and a second fluid, comprising:

5 a plurality of fins arranged spaced apart from and parallel to each other so as to form a space or spaces for allowing the first fluid to flow therethrough; and
 a plurality of heat transfer tubes penetrating said plurality of fins, for allowing the second fluid to flow therethrough, wherein:
 said plurality of heat transfer tubes include a first heat transfer tube and a second heat transfer tube arranged
 10 in a predetermined row direction that intersects with a flow direction of the first fluid;
 said first heat transfer tube and said second heat transfer tube are adjacent to each other with respect to the row direction;
 each of said fins has a protrusion formed between said first heat transfer tube and said second heat transfer tube, said protrusion guiding the first fluid to said first heat transfer tube side and said second heat transfer tube
 15 side; and
 the equivalent diameter of said protrusion, as viewed in an axis direction of said heat transfer tubes, is equal to or greater than the outer diameter of said heat transfer tubes.

2. The fin-tube heat exchanger according to claim 1, wherein:

20 the width of said protrusion along the row direction increases from an upstream edge of said protrusion to an intermediate portion thereof but decreases from said intermediate portion to a downstream edge thereof, along the flow direction of the first fluid;
 a first slanted surface slanted toward said first heat transfer tube side so as to guide the first fluid toward said
 25 first heat transfer tube side, and a second slanted surface slanted toward said second heat transfer tube side so as to guide the first fluid toward said second heat transfer tube side, are formed between said upstream edge and said intermediate portion of said protrusion;
 said upstream edge of said protrusion is located upstream of centers of said first and second heat transfer tubes; and
 30 said intermediate portion of said protrusion is located upstream of downstream edges of said first and second heat transfer tubes.

3. The fin-tube heat exchanger according to claim 1, wherein said heat transfer tubes and said protrusion are arrayed in a staggered manner, when viewed in an axis direction of said heat transfer tubes.

4. The fin-tube heat exchanger according to claim 1, wherein said protrusion is formed in a pyramidal shape.

5. The fin-tube heat exchanger according to claim 4, wherein said protrusion is formed in a quadrangular pyramidal shape.

6. The fin-tube heat exchanger according to claim 1, wherein said protrusion is formed in a circular conic shape or an elliptic conic shape.

7. The fin-tube heat exchanger according to claim 1, wherein said protrusion is formed in a circular hump shape or an elliptical hump shape.

8. The fin-tube heat exchanger according to claim 1, wherein the ratio L/H is greater than 5.5, where L is the length of said protrusion with respect to the flow direction of the first fluid and H is the protruding height of said protrusion.

9. The fin-tube heat exchanger according to claim 1, wherein the occupied area of said protrusion in each of said fins is from 43% to 73%.

10. The fin-tube heat exchanger according to claim 1, wherein:

55 said protrusion is such that, when said protrusion is orthogonally projected onto a plane parallel to said plurality of fins, an image of said protrusion appearing on the plane shows a circular shape or an elliptical shape; and the area of the image of said protrusion appearing on the plane is greater than the cross-sectional area of each of said heat transfer tubes.

11. The fin-tube heat exchanger according to claim 1, wherein only one said protrusion is formed between said first heat transfer tube and said second heat transfer tube.
- 5 12. The fin-tube heat exchanger according to claim 10, wherein said upstream edge of said protrusion is located, with respect to the flow direction of the first fluid, closer to leading edges of said plurality of fins than upstream edges of said first heat transfer tube and said second heat transfer tube.
- 10 13. The fin-tube heat exchanger according to claim 10, wherein said protrusion is such that the image thereof appearing on the plane is in an elliptical shape, and that the orientation of said protrusion is determined such that the minor axis of the ellipse is parallel to the row direction in which said first heat transfer tube and said second heat transfer tube are arranged.
- 15 14. The fin-tube heat exchanger according to claim 13, wherein said protrusion is such that the location thereof relative to said first heat transfer tube and said second heat transfer tube is determined so that the major axis of the ellipse is contained in a virtual plane that perpendicularly bisects a line segment connecting the center of said first heat transfer tube and the center of said second heat transfer tube at the shortest distance.
- 20 15. The fin-tube heat exchanger according to claim 13, wherein the shape of said protrusion is adjusted so that its surface forms a curve in a cross section that contains the minor axis or the major axis of the ellipse and is perpendicular to principal surfaces of said plurality of fins.
- 25 16. The fin-tube heat exchanger according to claim 15, wherein said curve contains an inflection point between an upstream edge and an apex of said protrusion.
- 30 17. The fin-tube heat exchanger according to claim 15, wherein said curve contains no inflection point between an upstream edge and an apex of said protrusion.
18. The fin-tube heat exchanger according to claim 13, wherein the shape of said protrusion is adjusted so that its surface forms a sine curve in a cross section that contains the minor axis or the major axis of the ellipse and is perpendicular to principal surfaces of said plurality of fins.
- 35 19. The fin-tube heat exchanger according to claim 18, wherein said surface follows a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-180^\circ \leq X \leq 180^\circ$ }.
20. The fin-tube heat exchanger according to claim 18, wherein said surface follows a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-90^\circ \leq X \leq 90^\circ$ }.
- 40 21. The fin-tube heat exchanger according to claim 10, wherein the shape of said protrusion is adjusted so that its surface forms a clothoid curve in a cross section that is parallel to the flow direction of the first fluid and is perpendicular to principal surfaces of said plurality of fins.
- 45 22. The fin-tube heat exchanger according to claim 1, wherein the height of said protrusion is adjusted so as to satisfy the expression $(FP/4) \leq H \leq FP$, where H is the height of said protrusion and FP is the fin pitch, which is a parallel gap distance between said plurality of fins.
- 50 23. The fin-tube heat exchanger according to claim 10, wherein the shape of said protrusion is adjusted so that the image thereof appearing on the plane shows a circular shape and that its surface forms a curve in a cross section that is perpendicular to principal surfaces of said plurality of fins.
- 55 24. The fin-tube heat exchanger according to claim 23, wherein said curve contains an inflection point between said upstream edge and an apex of said protrusion.
25. The fin-tube heat exchanger according to claim 23, wherein said curve contains no inflection point between an upstream edge and an apex of said protrusion.
26. The fin-tube heat exchanger according to claim 10, wherein the shape of said protrusion is adjusted so that the image thereof appearing on the plane shows a circular shape and that its surface forms a sine curve in a cross section that is perpendicular to principal surfaces of said plurality of fins.

27. The fin-tube heat exchanger according to claim 26, wherein said surface follows a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-180^\circ \leq X \leq 180^\circ$ }.

28. The fin-tube heat exchanger according to claim 26, wherein said surface follows a sine curve represented by the equation $Y = K\cos(X)$ {K: constant, $-90^\circ \leq X \leq 90^\circ$ }.

29. The fin-tube heat exchanger according to claim 1, wherein:

said heat transfer tubes are disposed in a staggered manner in two rows, one of the two rows being a front row that is closer to leading edges of said plurality of fins and the other row being a rear row that is parallel to the front row; and

another protrusion is formed between two adjacent ones of said heat transfer tubes disposed in the rear row, said other protrusion having the same shape and the same dimensions as those of said protrusion that is formed between two adjacent ones of said heat transfer tubes disposed in the front row.

30. The fin-tube heat exchanger according to claim 29, wherein a second protrusion is formed between said protrusion formed in the front row and said other protrusion formed in the rear row, said second protrusion having a surface area smaller than that of said protrusion and said other protrusion.

31. A fin used for the fin-tube heat exchanger according to claim 1.

32. A heat pump apparatus comprising:

a compressor for compressing a refrigerant;

a radiator for cooling the refrigerant compressed by said compressor;

an expansion mechanism for expanding the refrigerant cooled by said radiator; and

an evaporator for evaporating the refrigerant expanded by said expansion mechanism, wherein

at least one of said evaporator and said radiator comprises the fin-tube heat exchanger according to claim 1.

33. The heat pump apparatus according to claim 32, wherein at least said evaporator comprises the fin-tube heat exchanger according to claim 1.

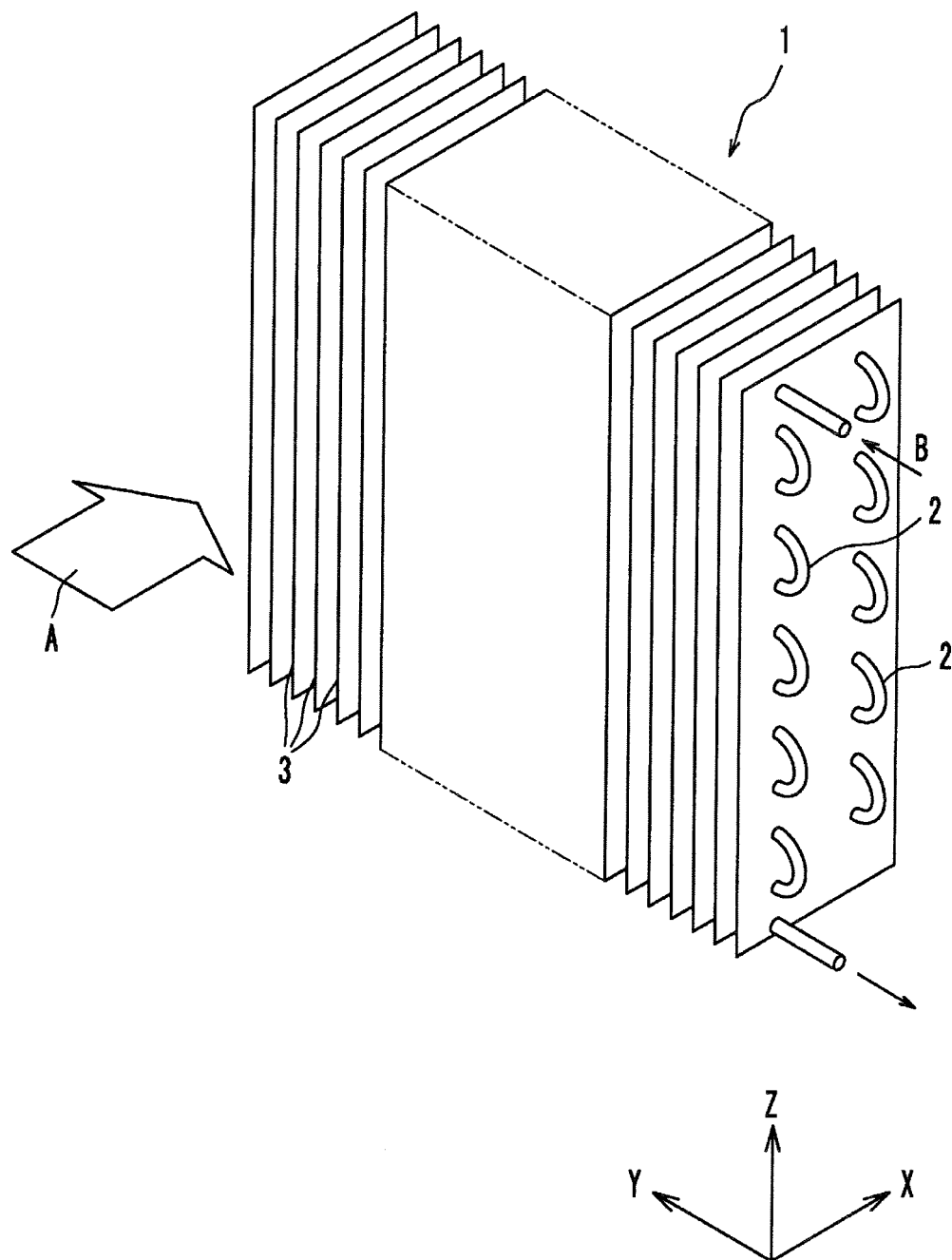


FIG.1

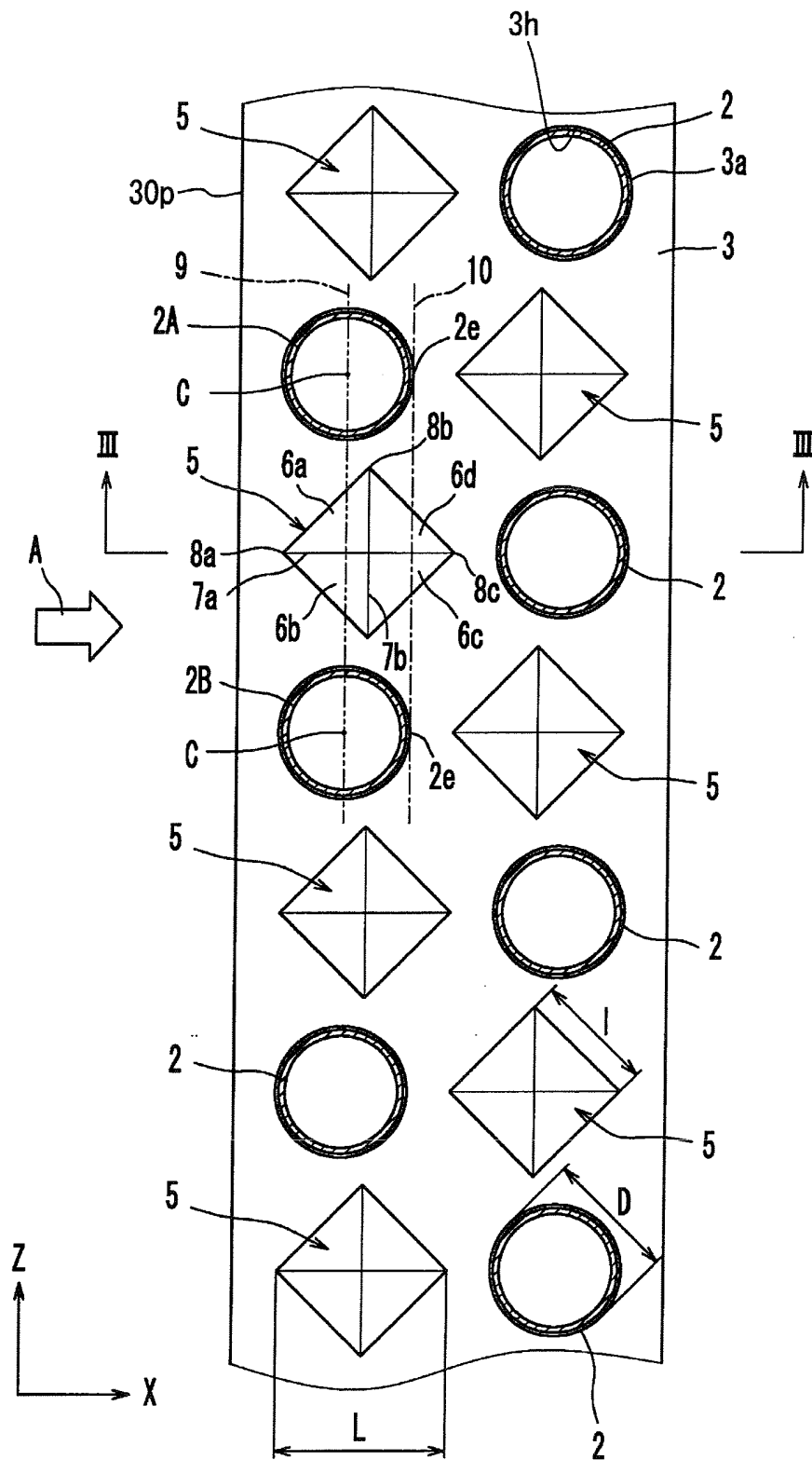


FIG.2

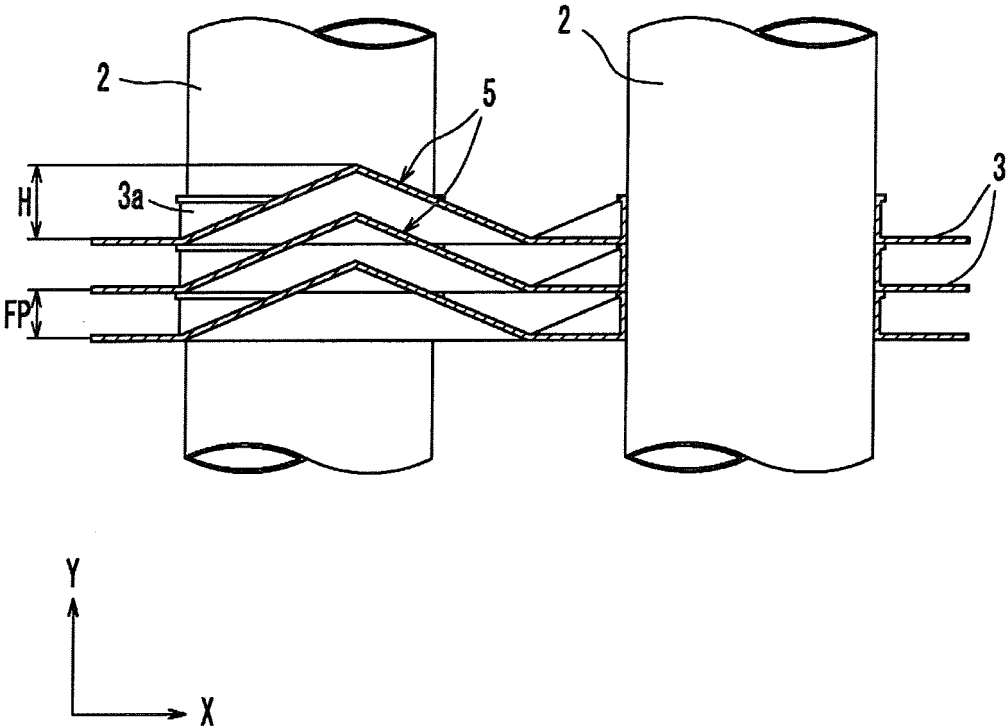


FIG.3

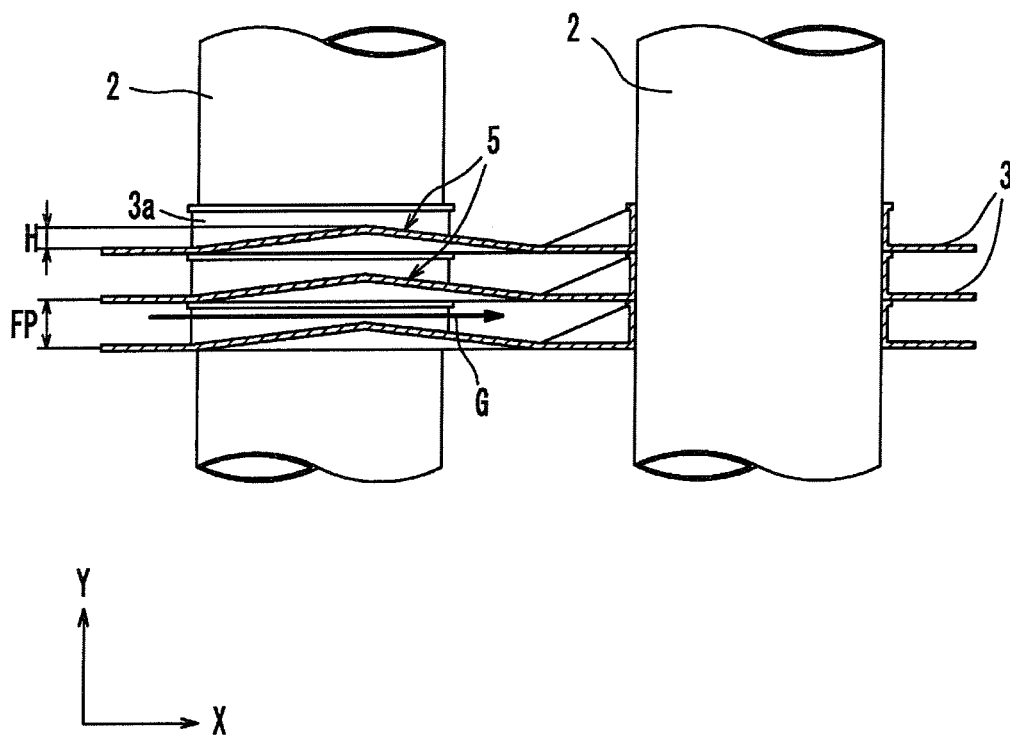


FIG.4

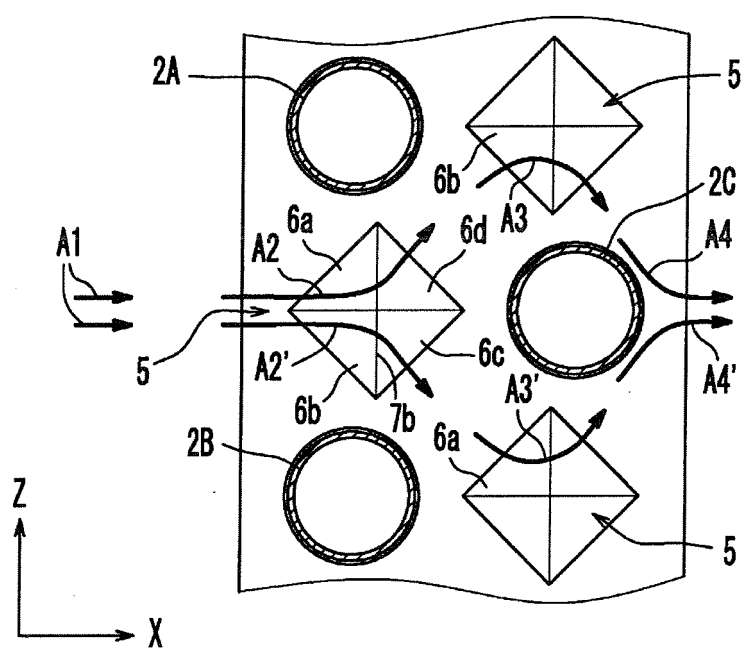


FIG.5

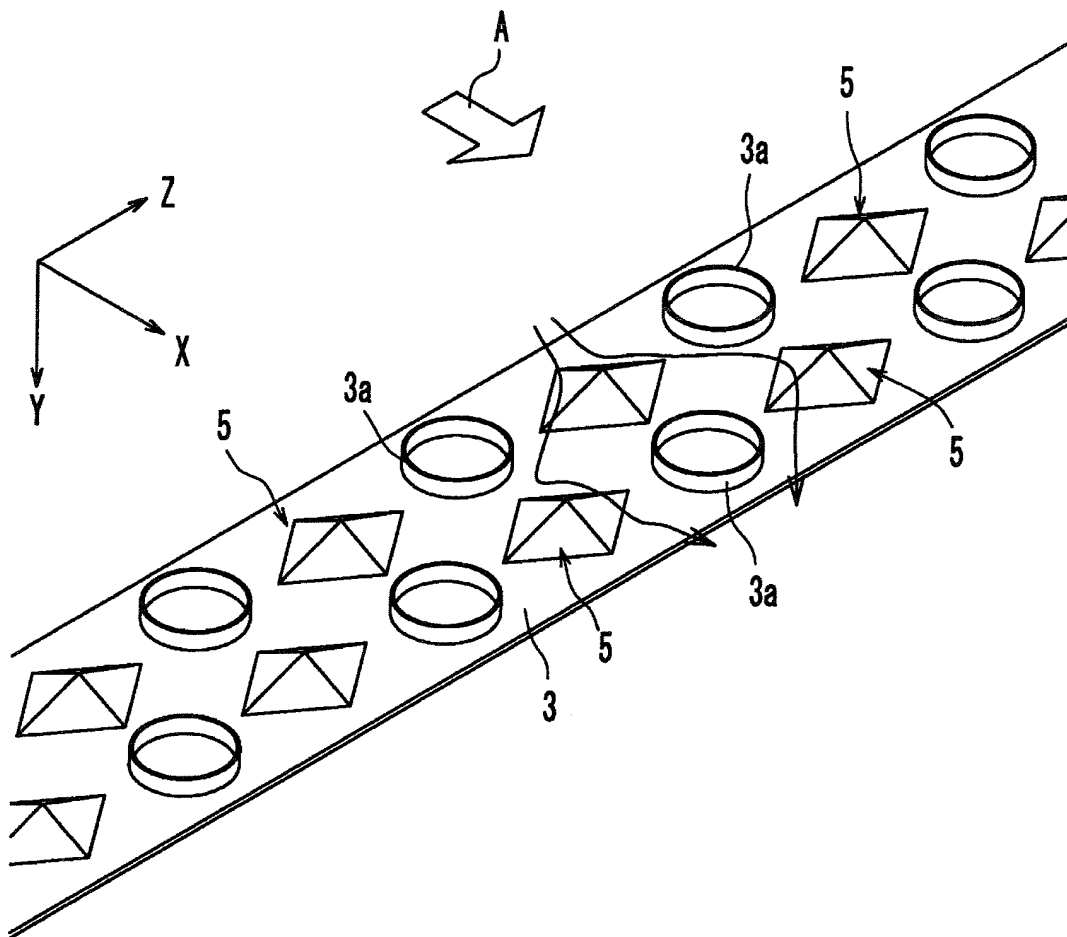


FIG. 6

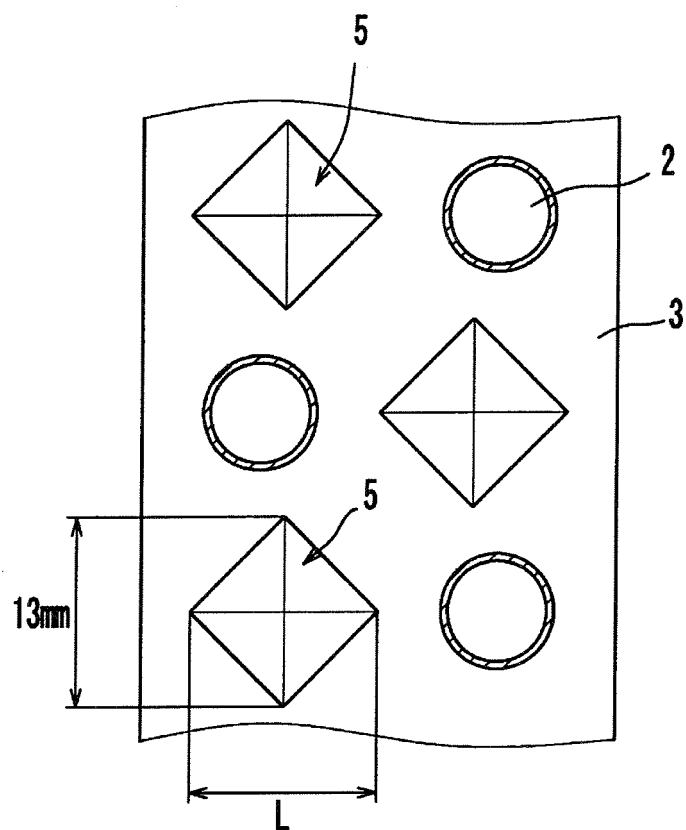


FIG.7

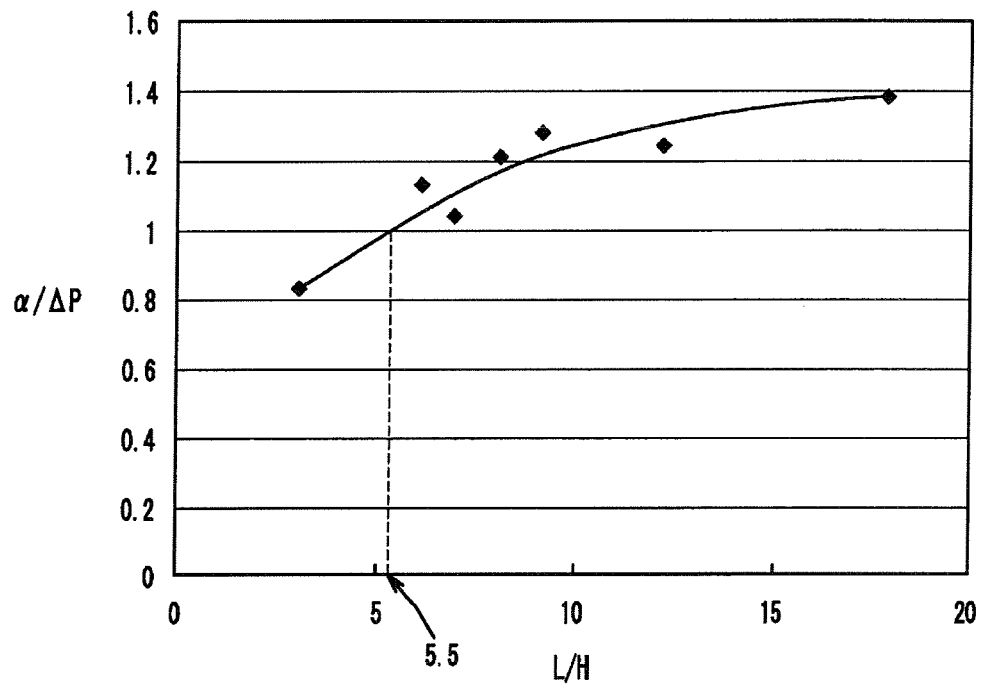


FIG.8

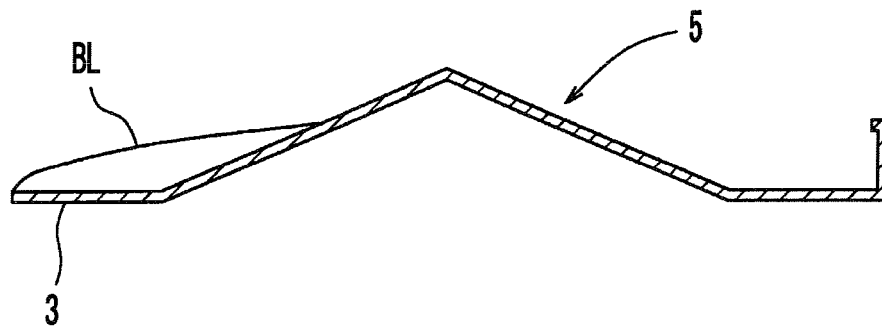


FIG.9

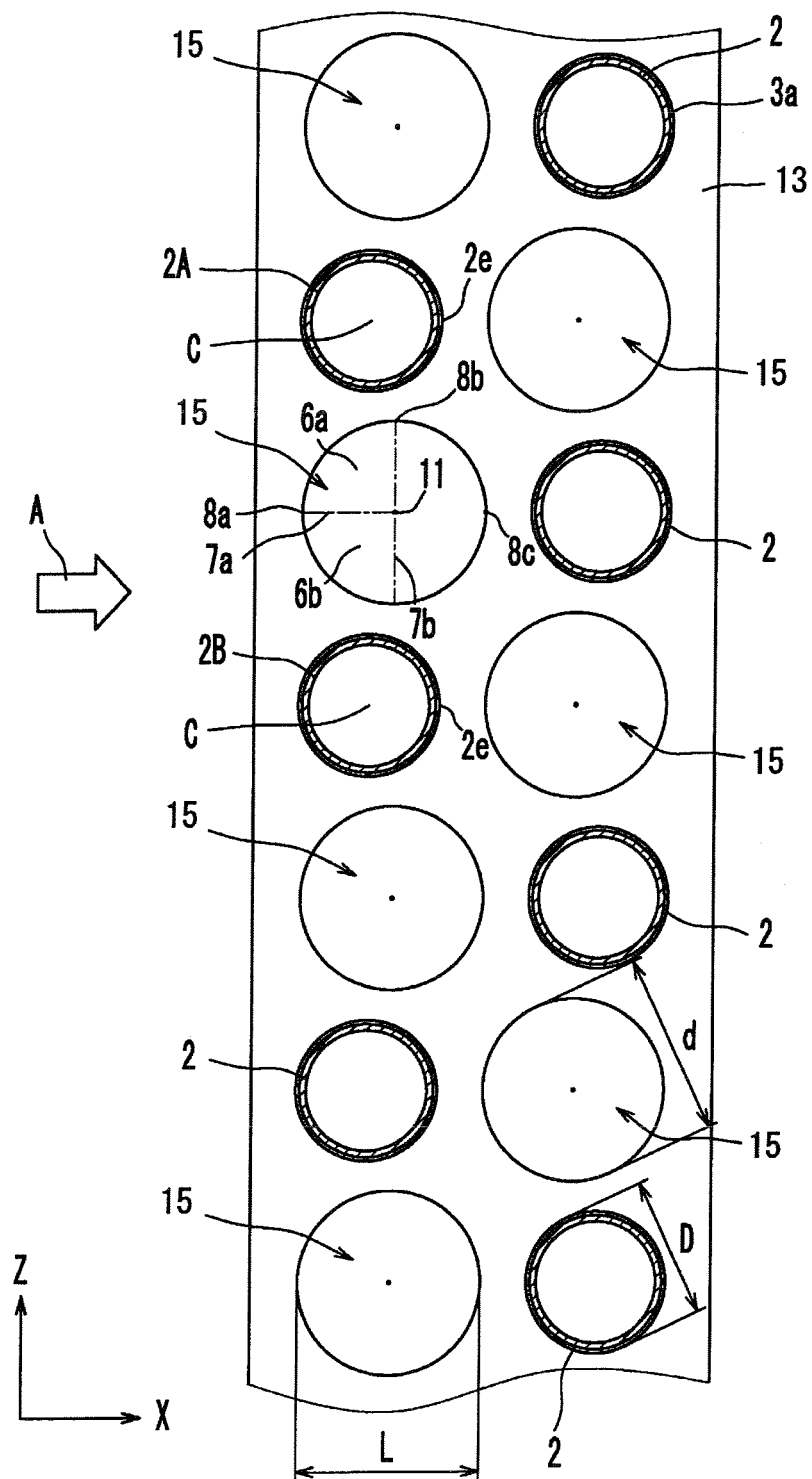


FIG.10

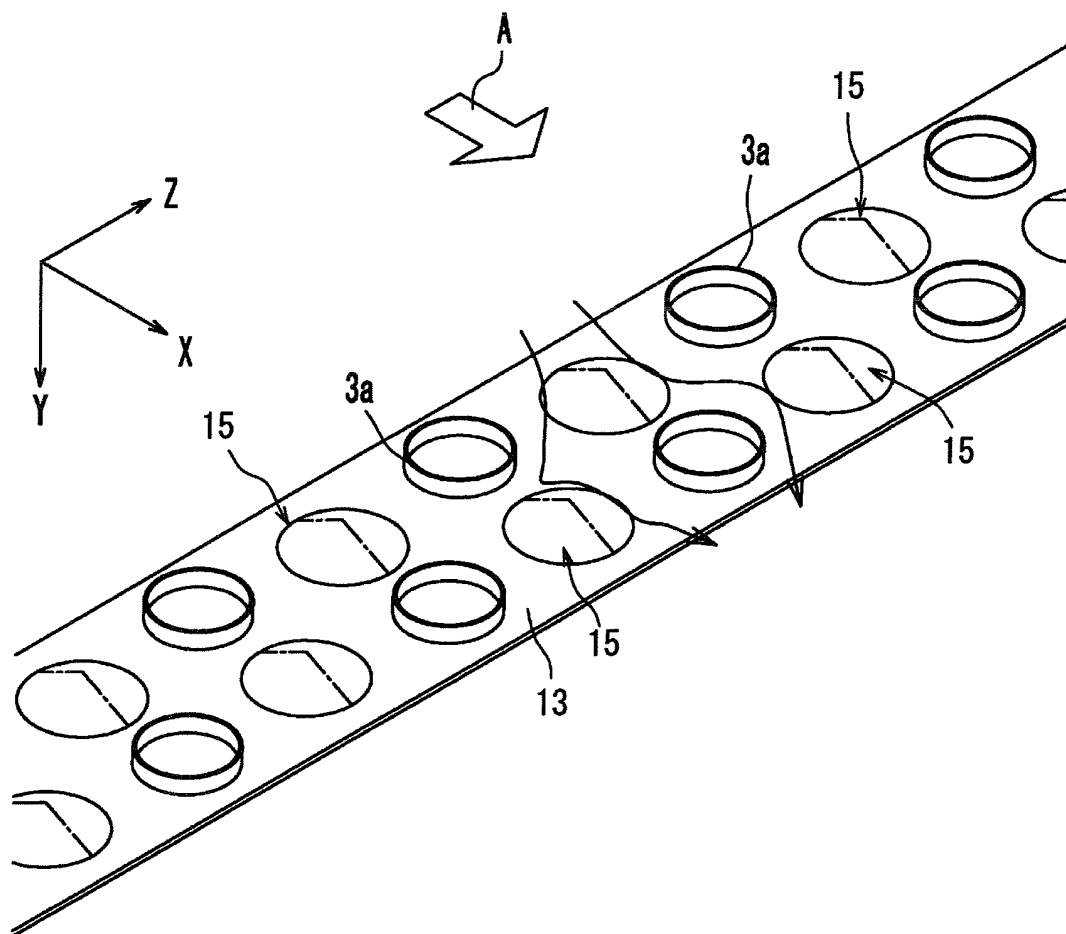


FIG. 11

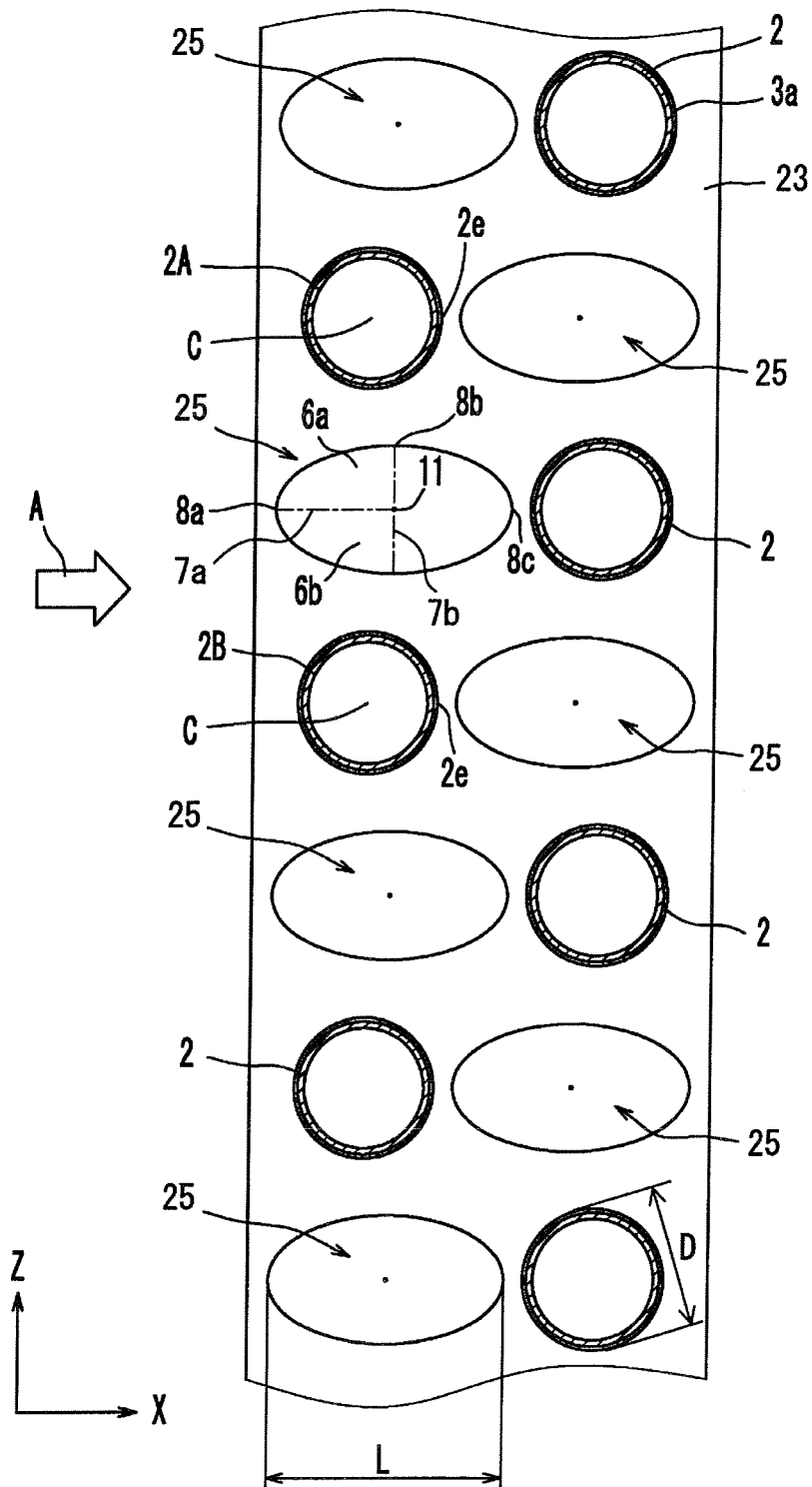


FIG.12

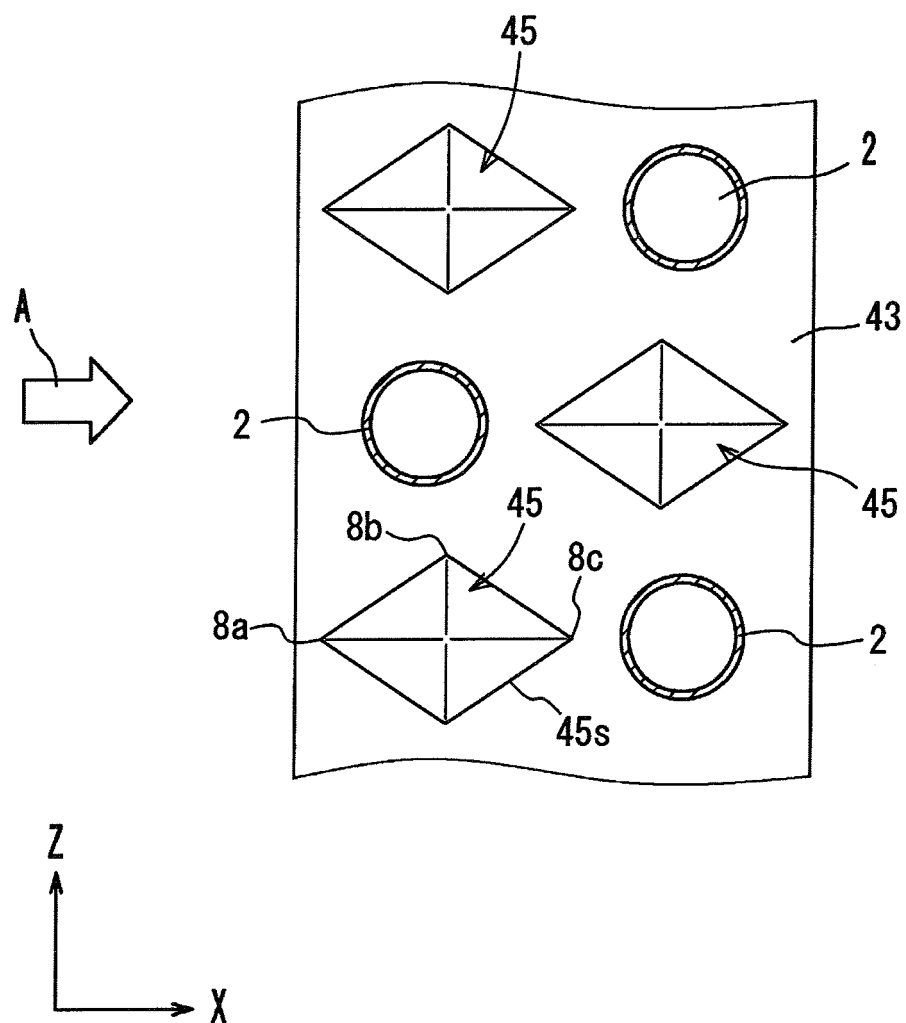


FIG.13

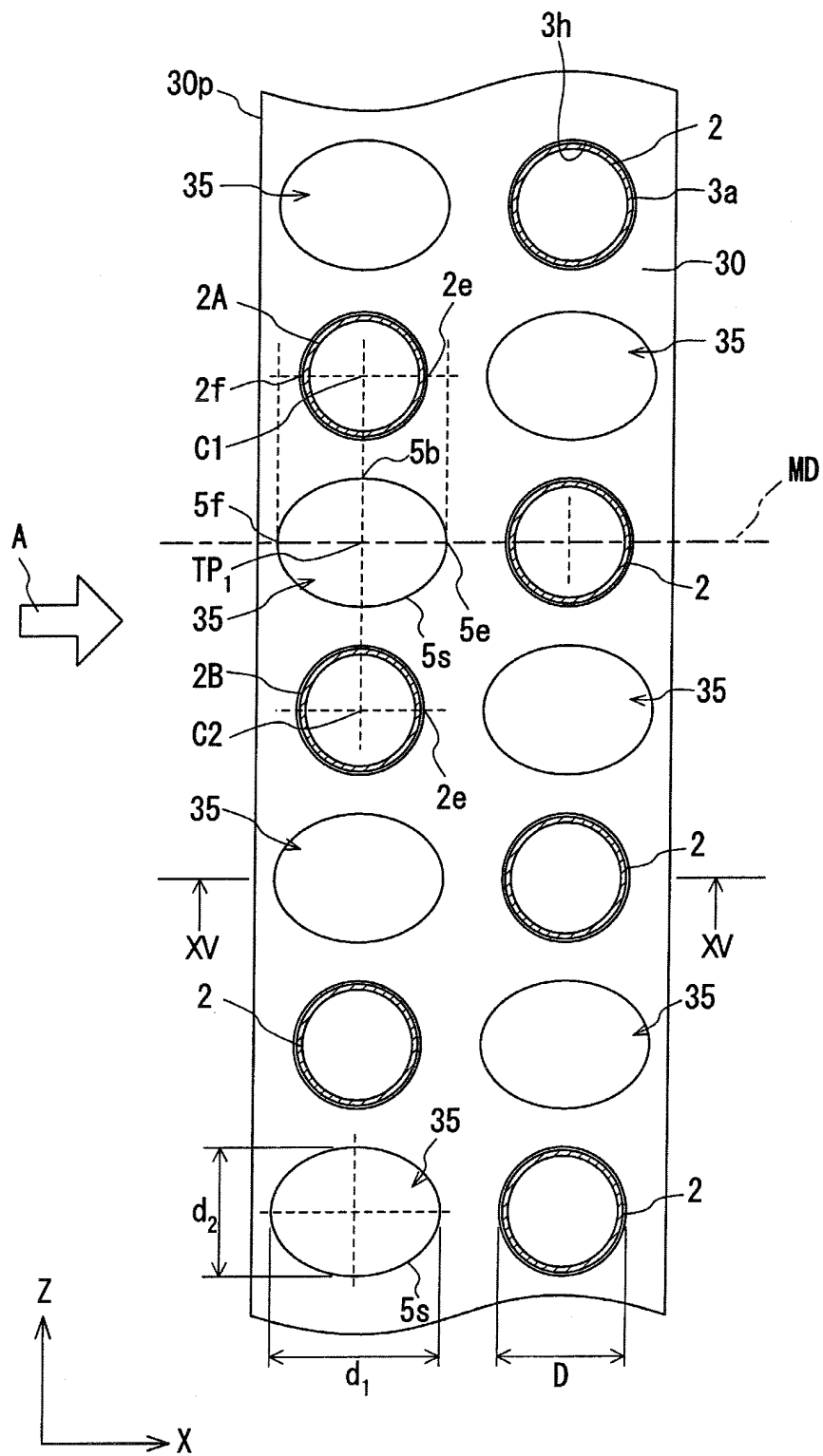


FIG.14

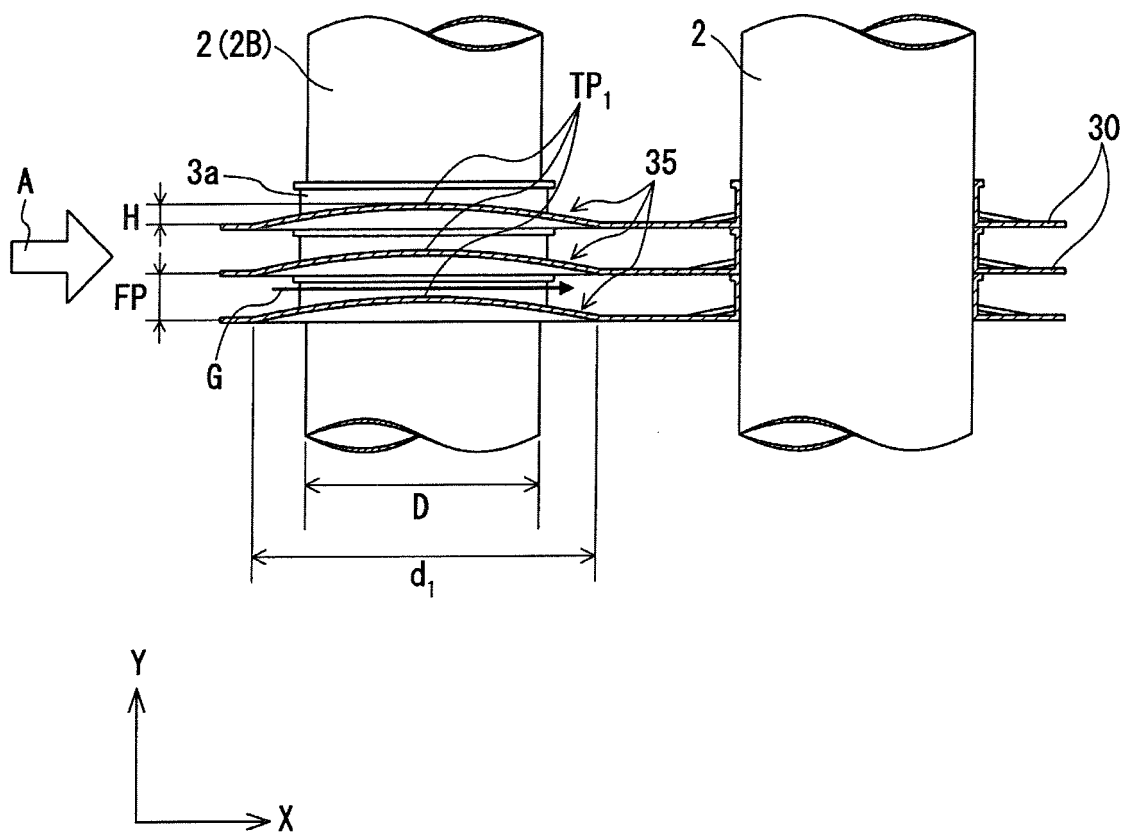


FIG.15

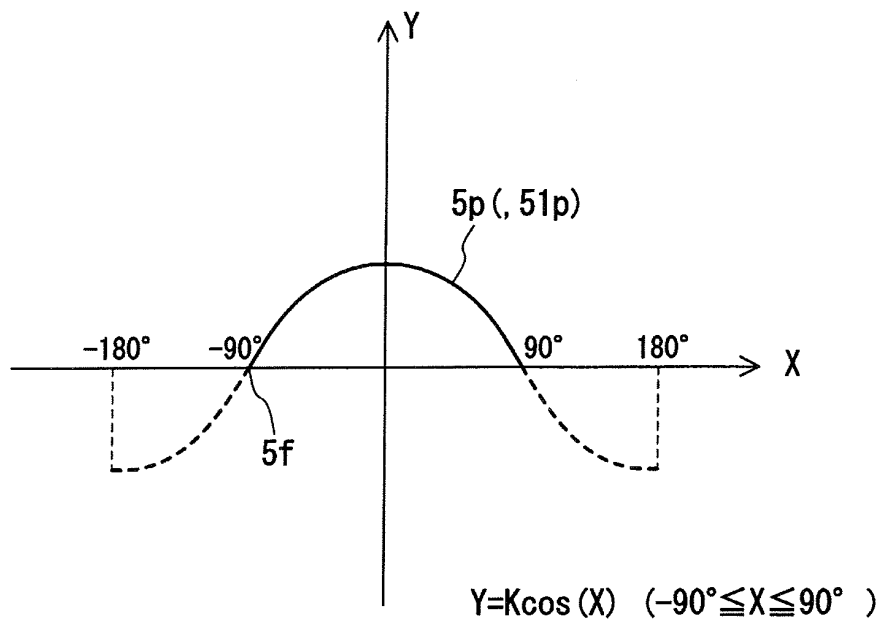


FIG.16A

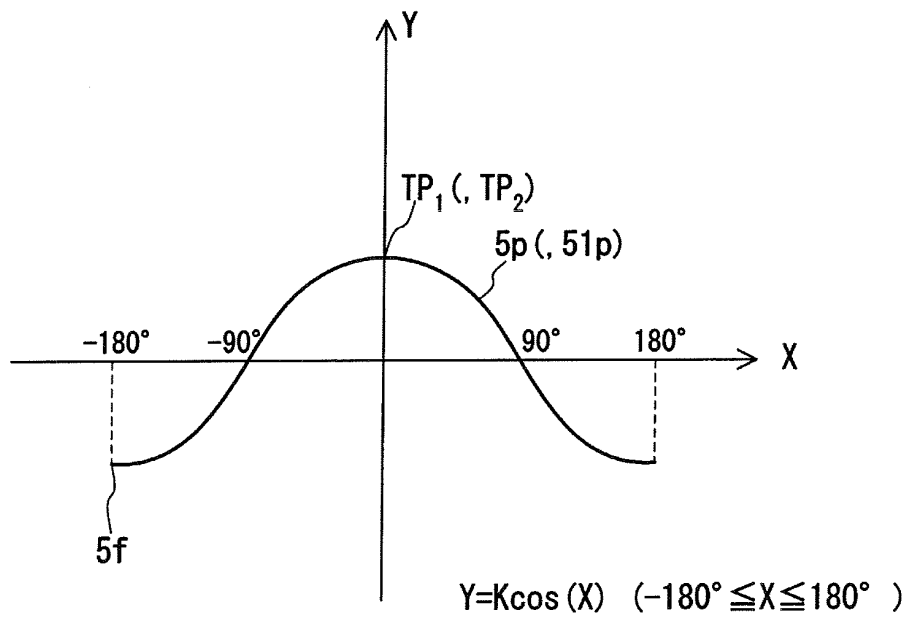


FIG.16B

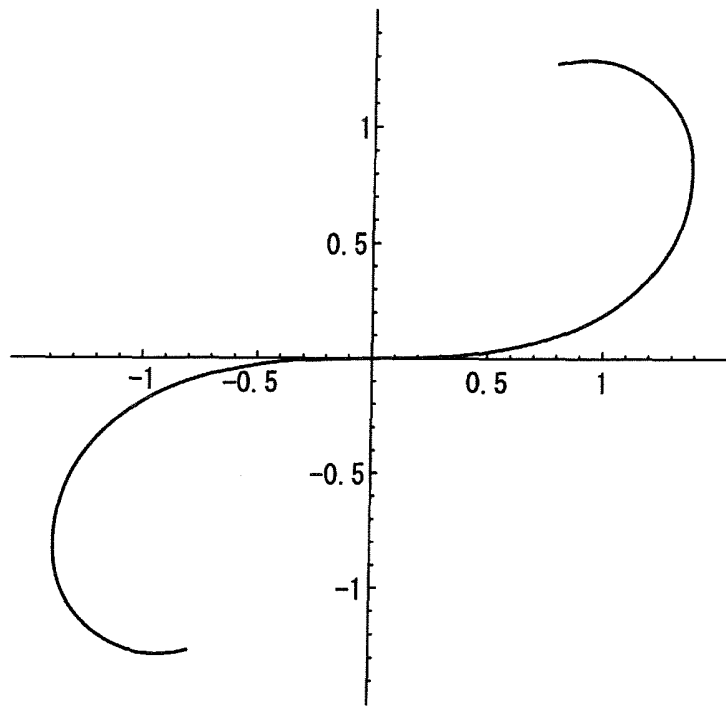


FIG. 17A

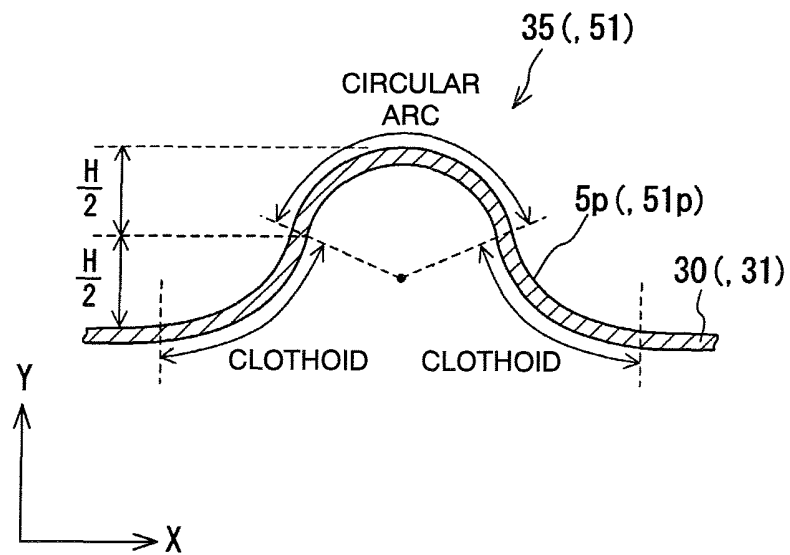


FIG. 17B

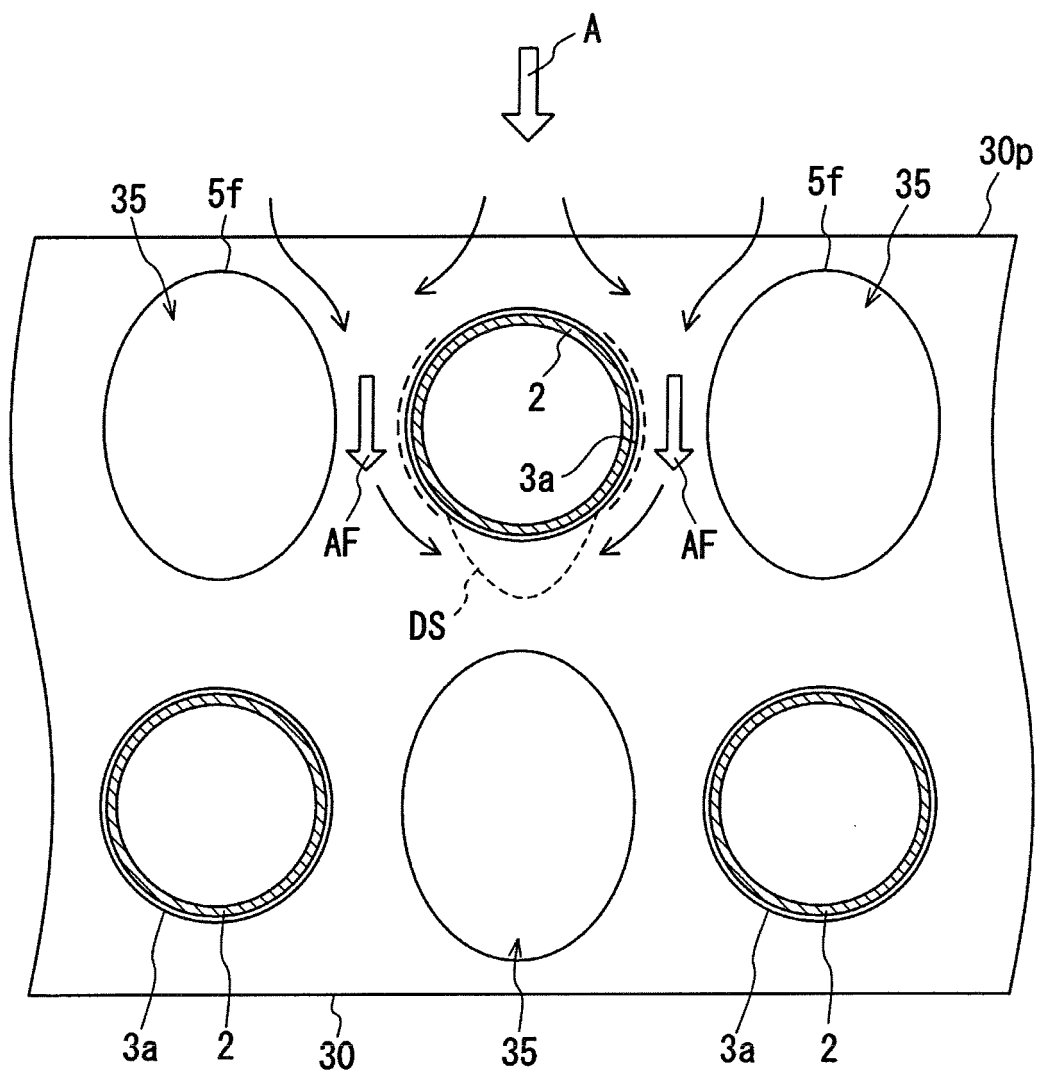


FIG. 18

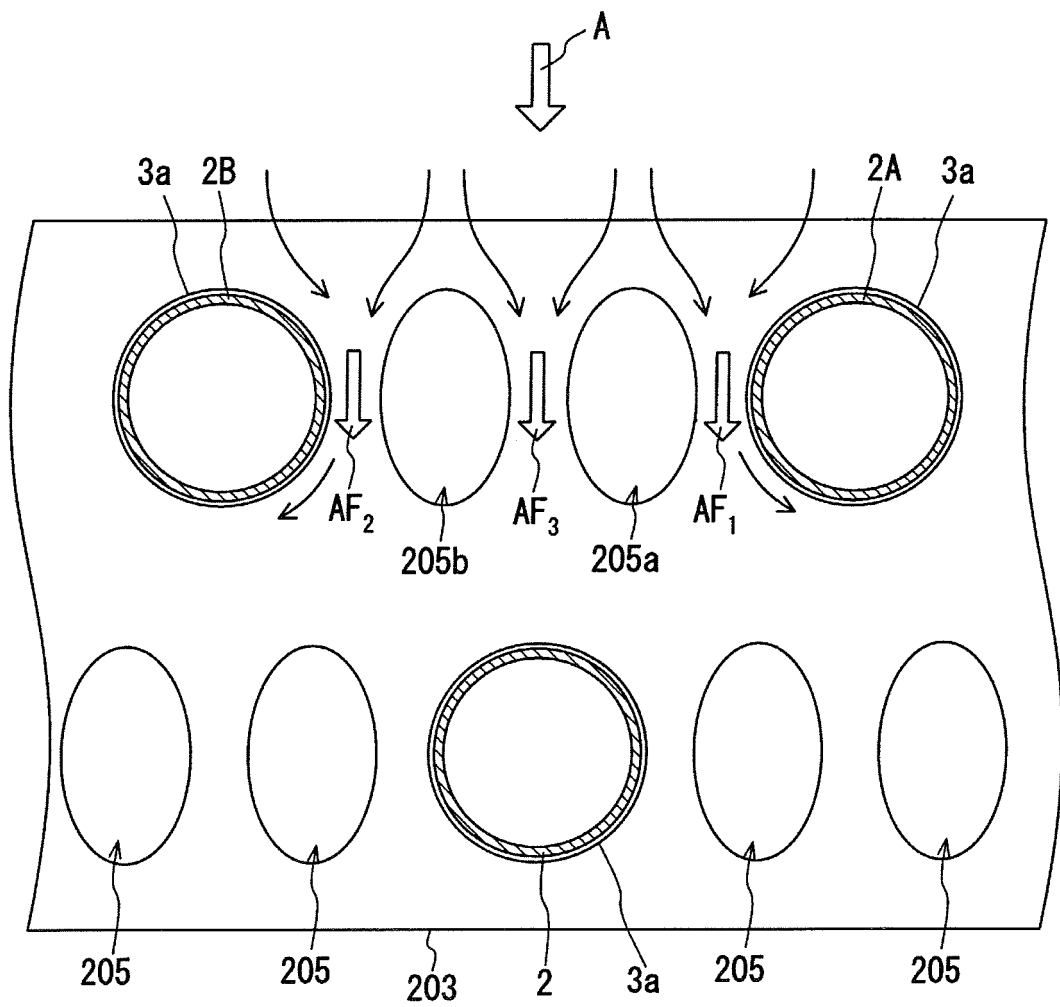


FIG.19

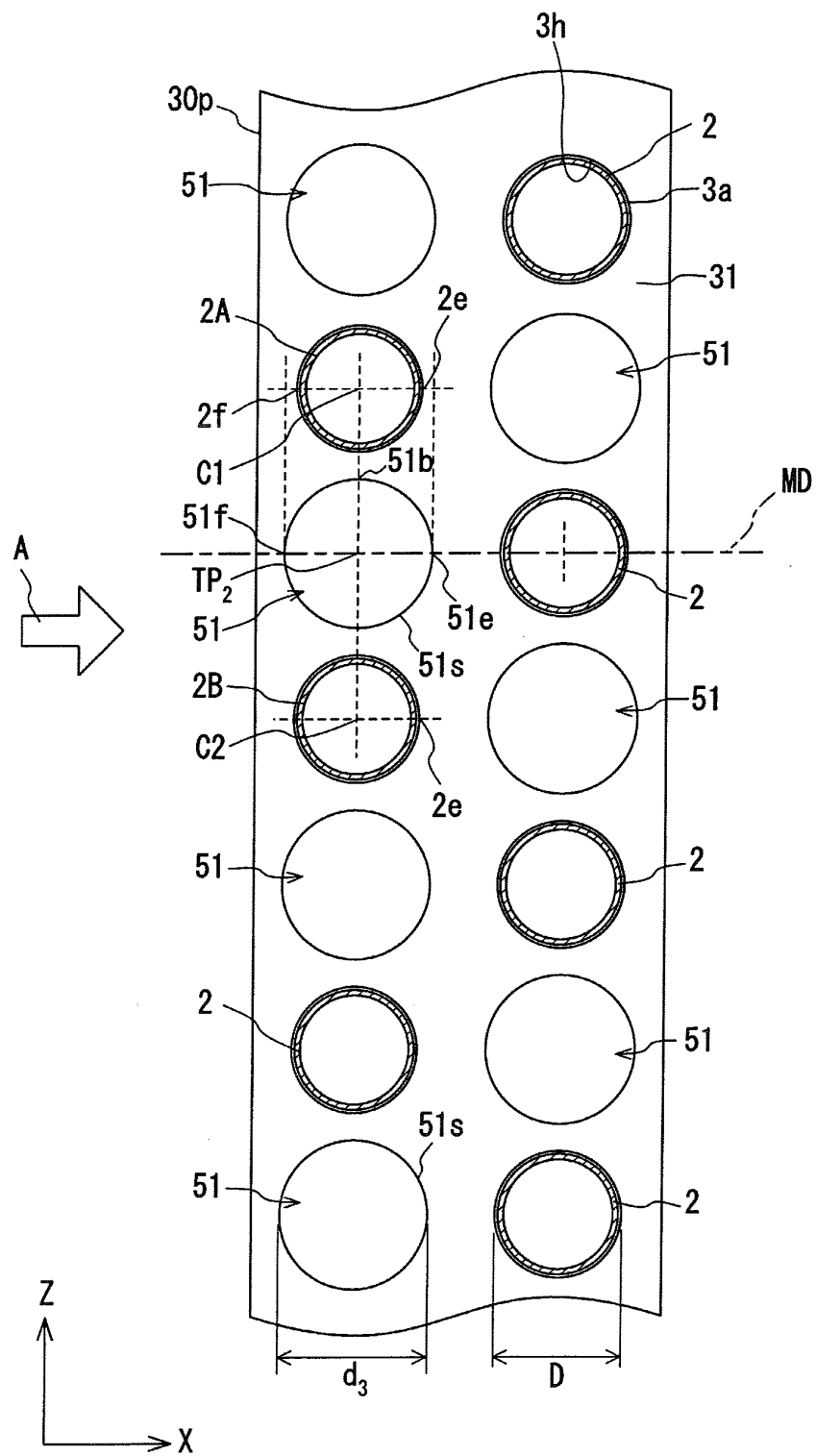


FIG.20

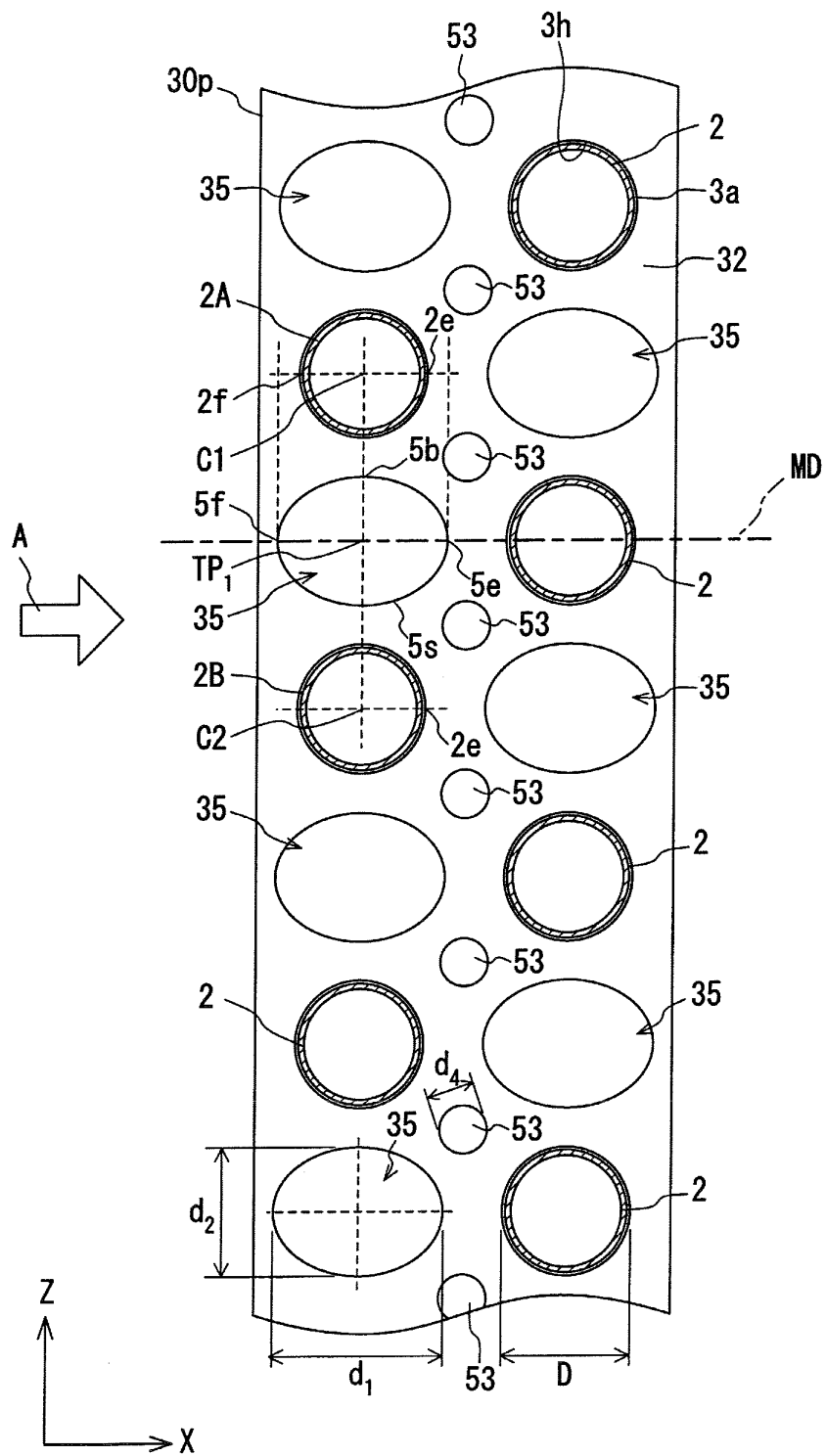


FIG. 21

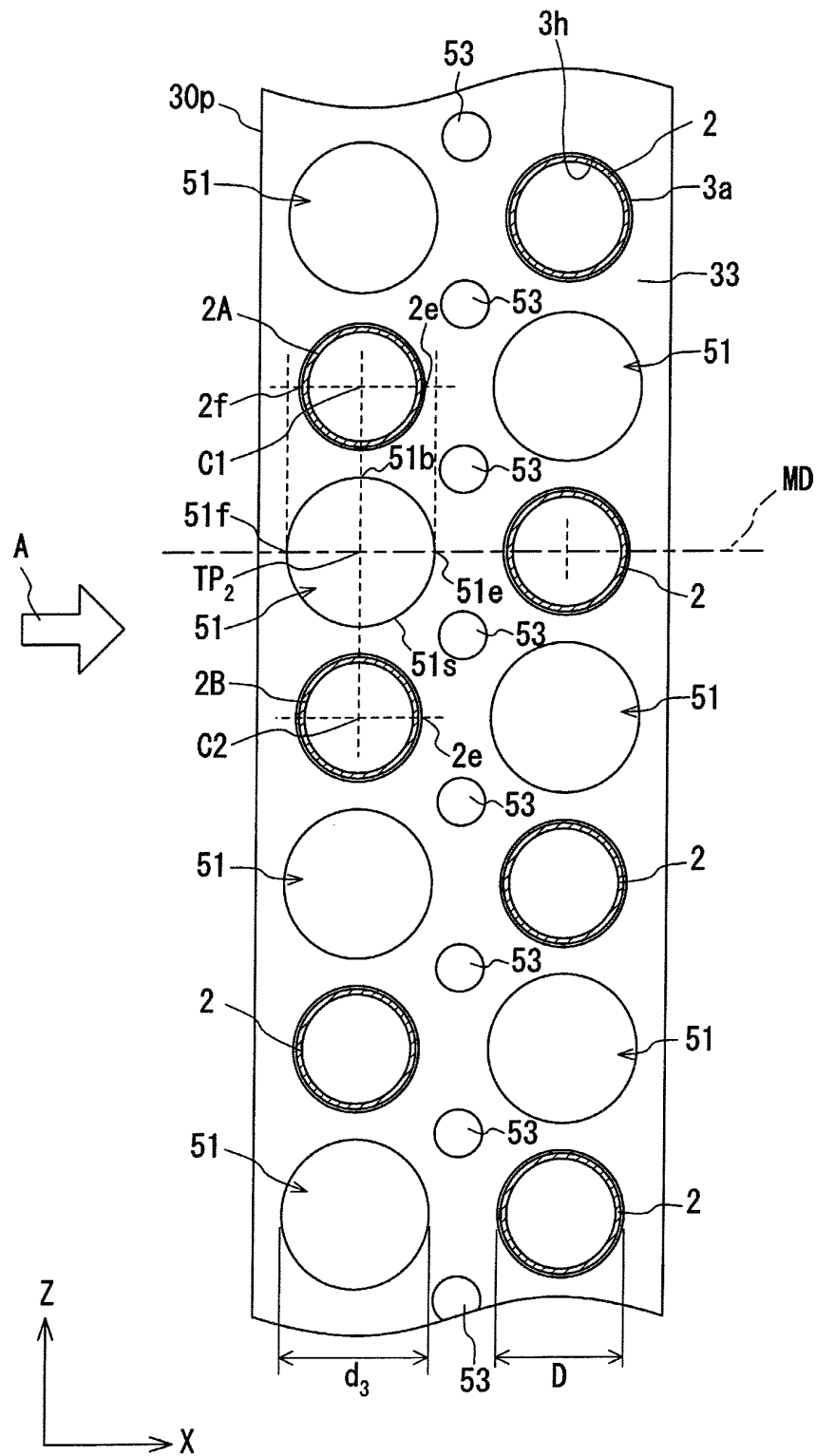


FIG.22

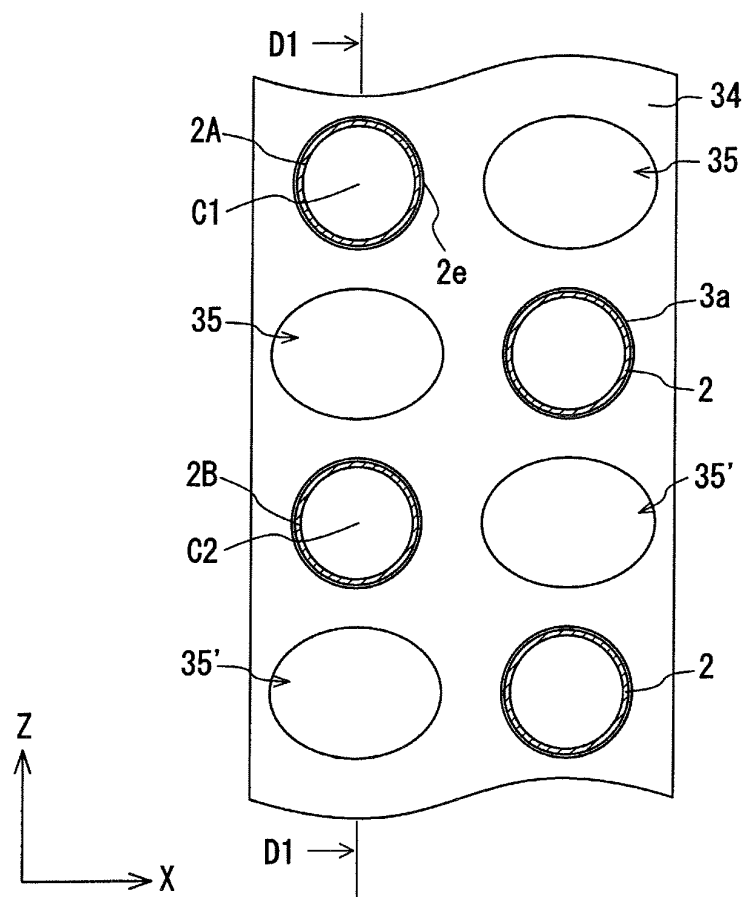


FIG. 23A

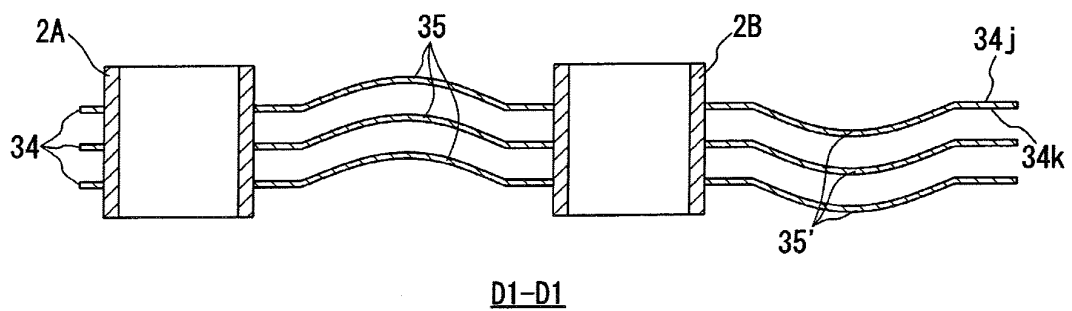


FIG. 23B

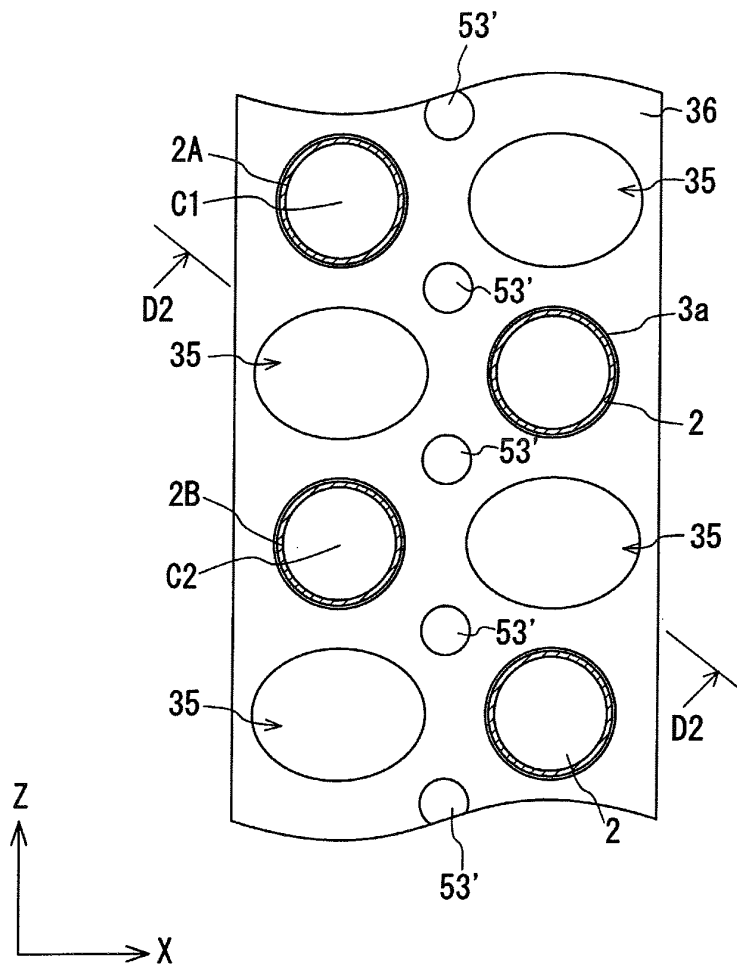


FIG. 24A

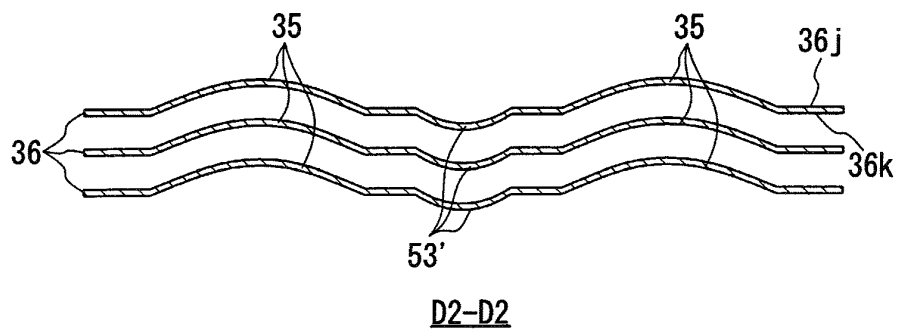


FIG. 24B

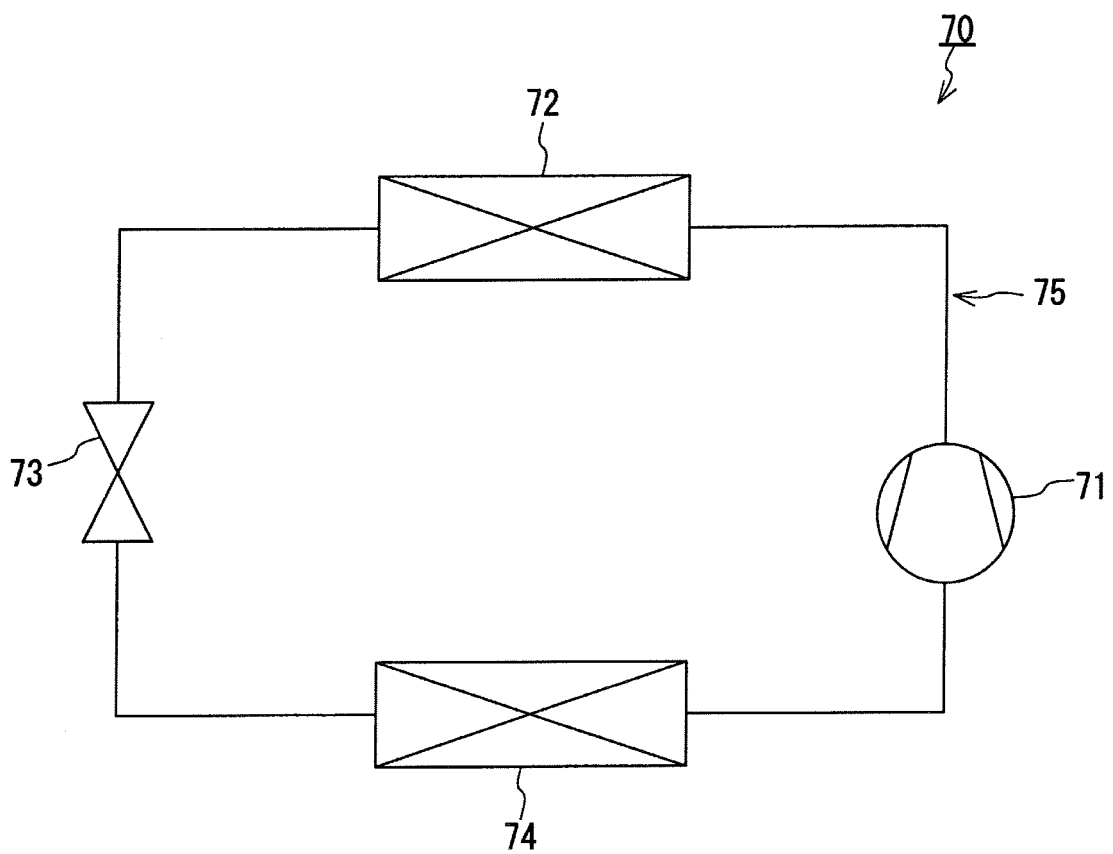


FIG.25

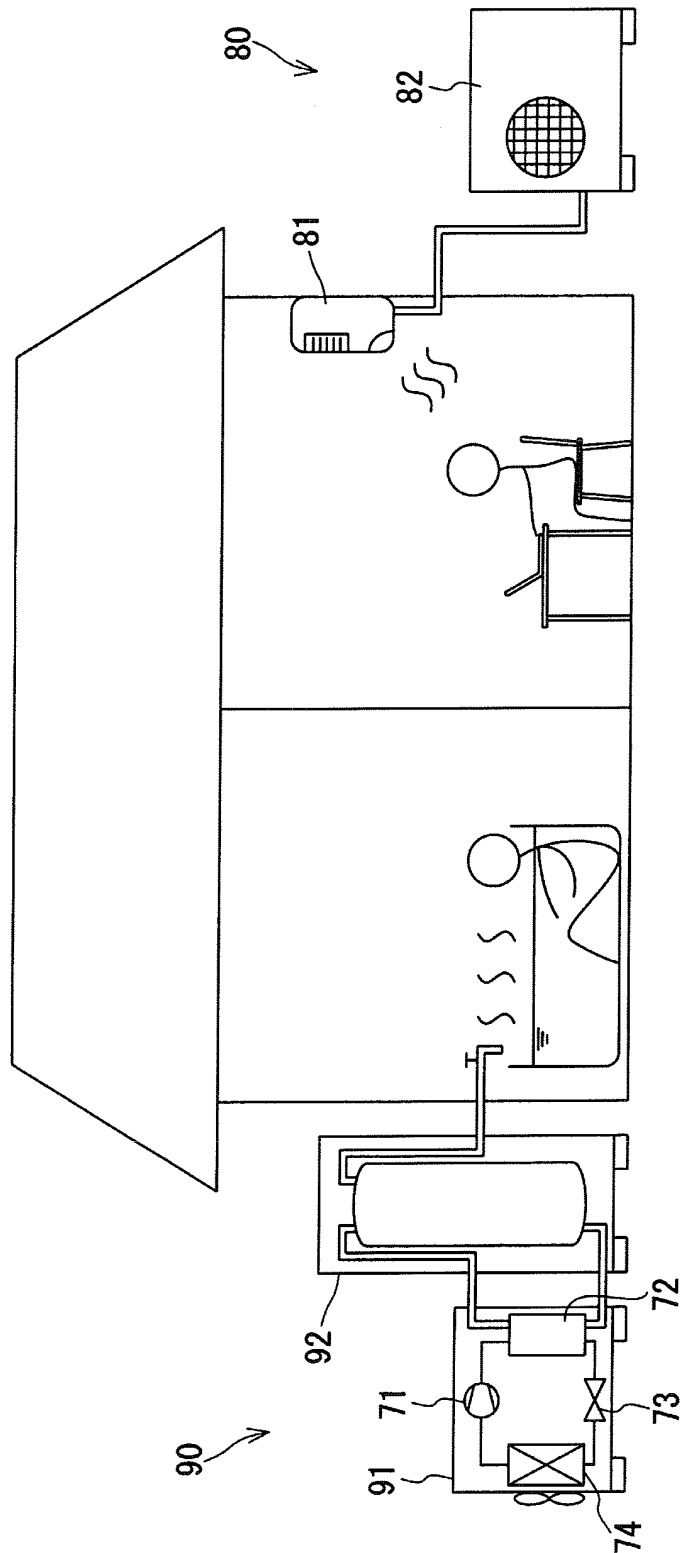


FIG.26

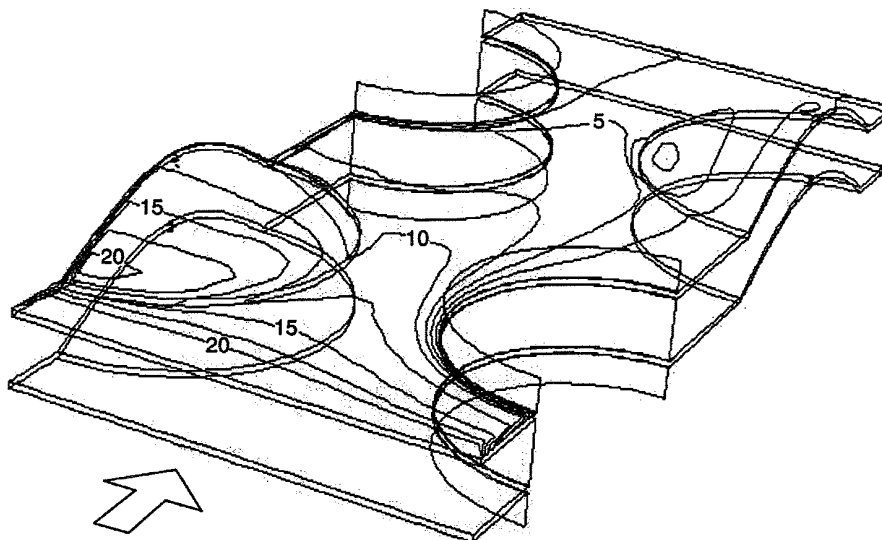
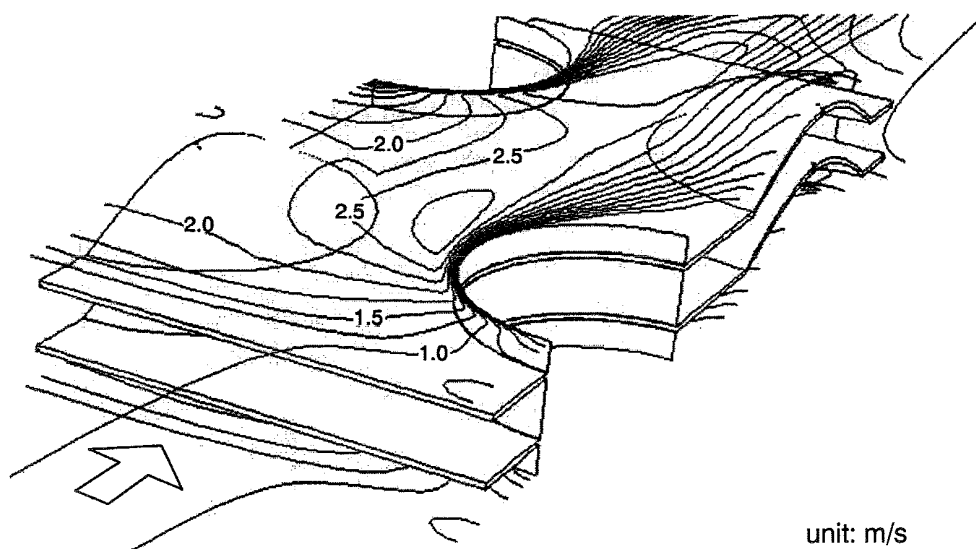


FIG.27A



unit: m/s

FIG.27B

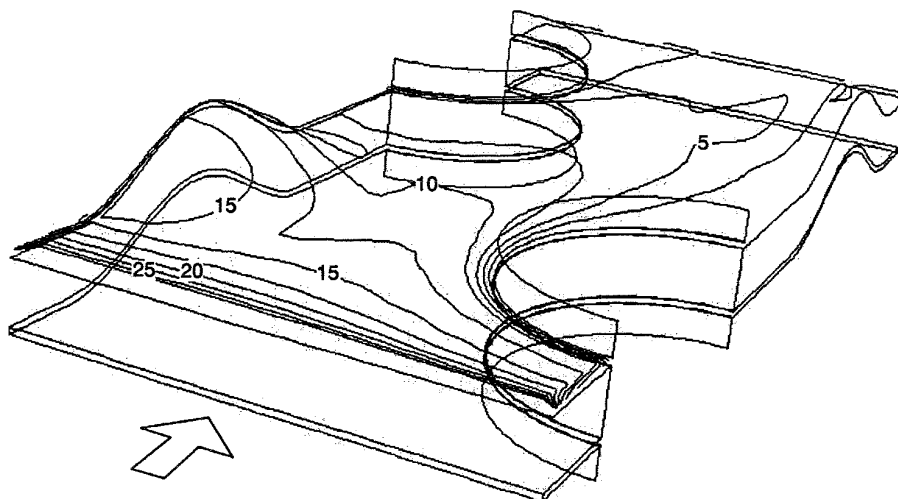


FIG.28A

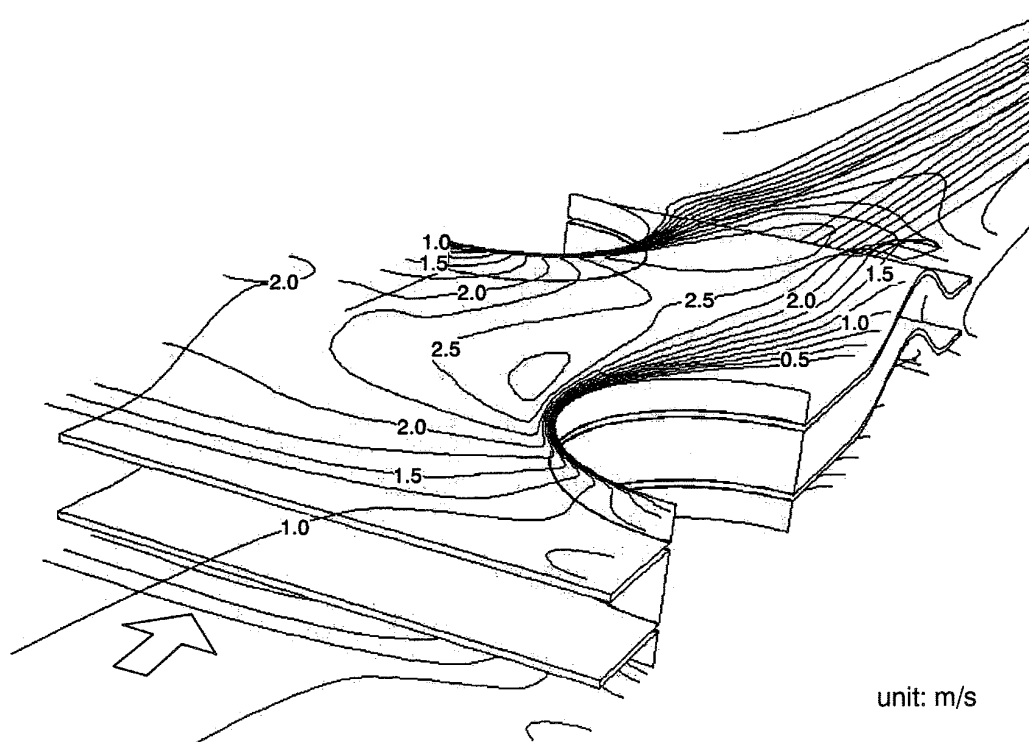


FIG.28B

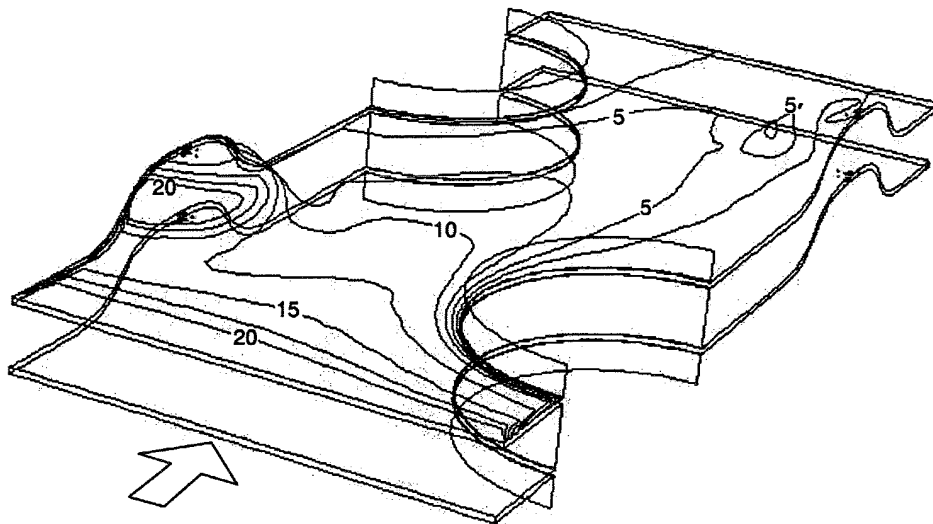


FIG.29A

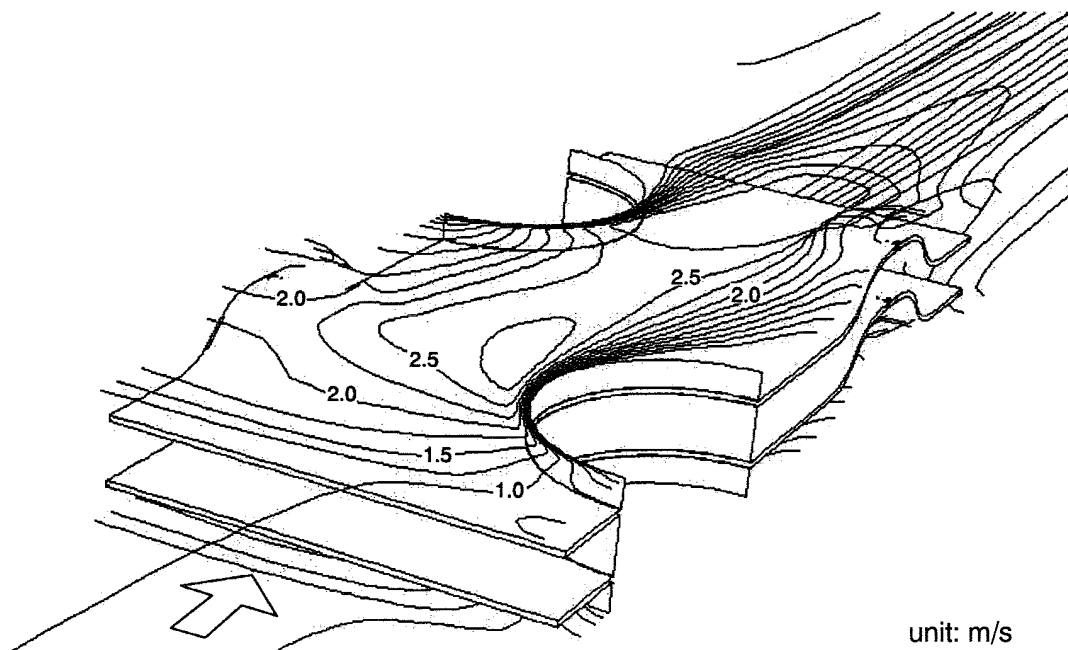


FIG.29B

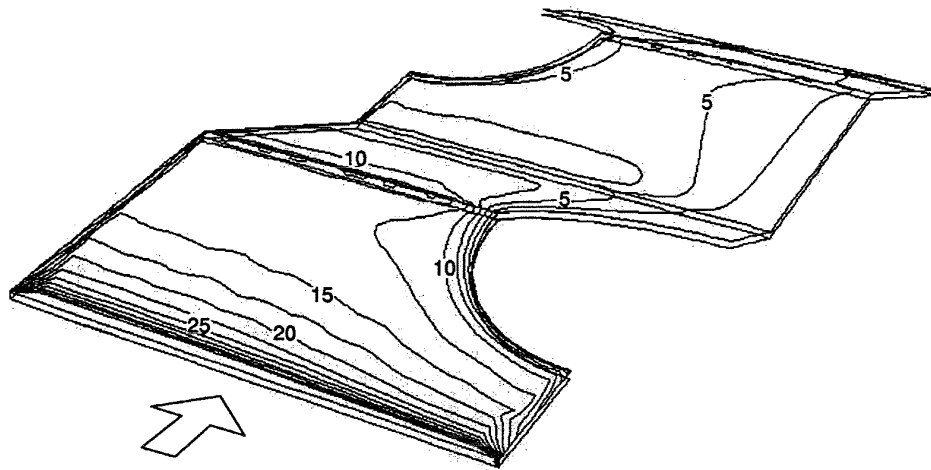


FIG.30A

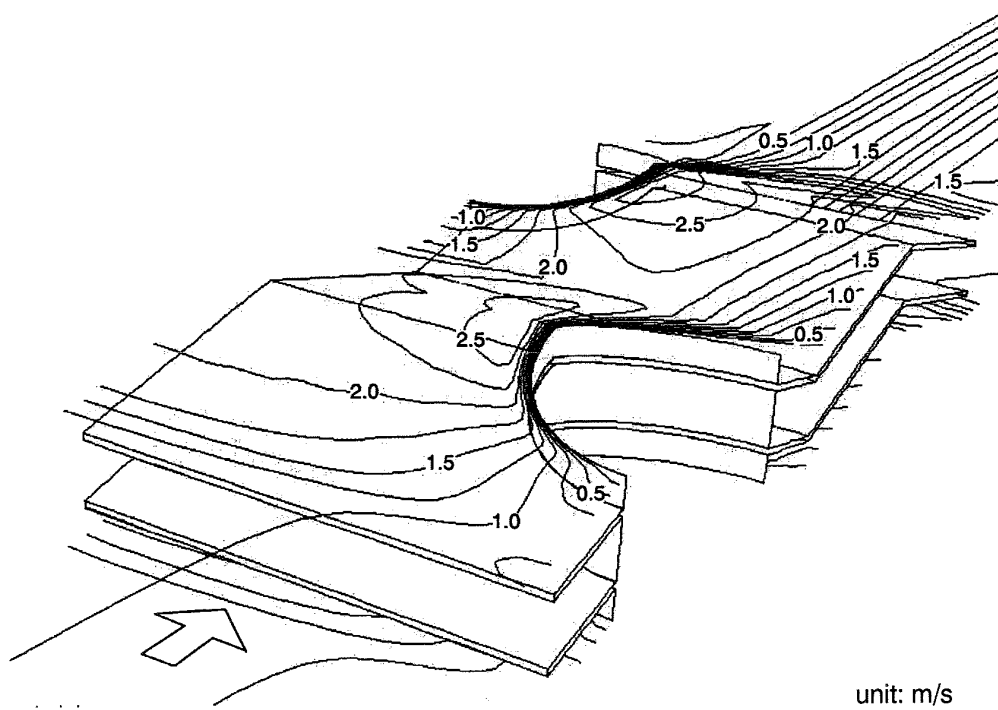


FIG.30B

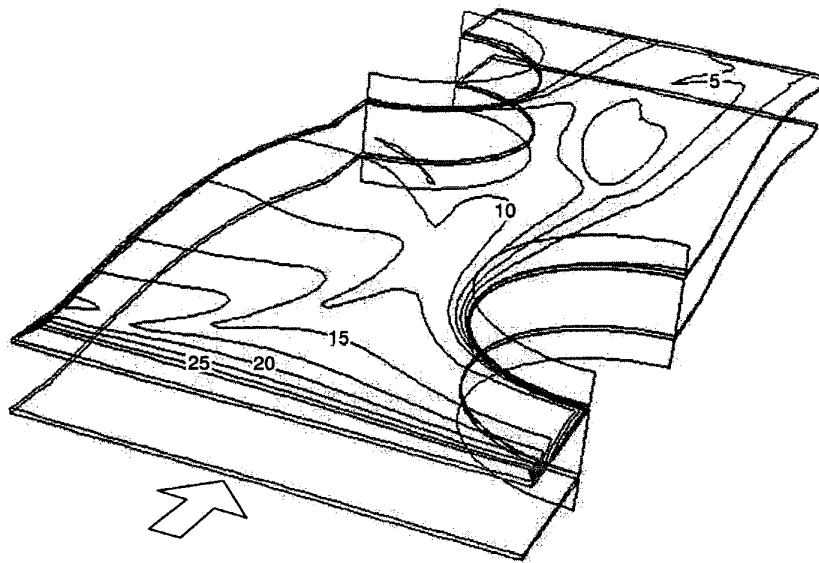


FIG.31A

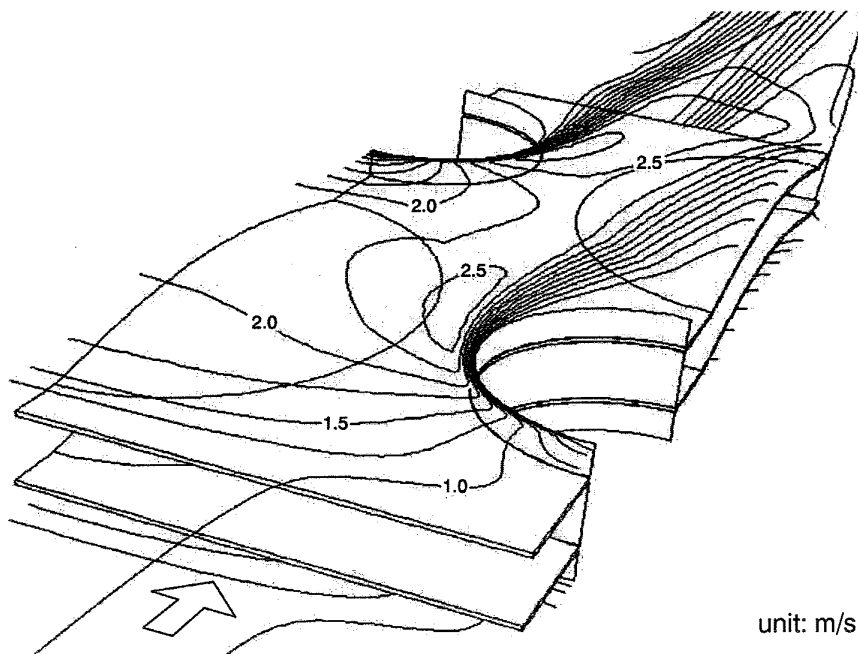


FIG.31B

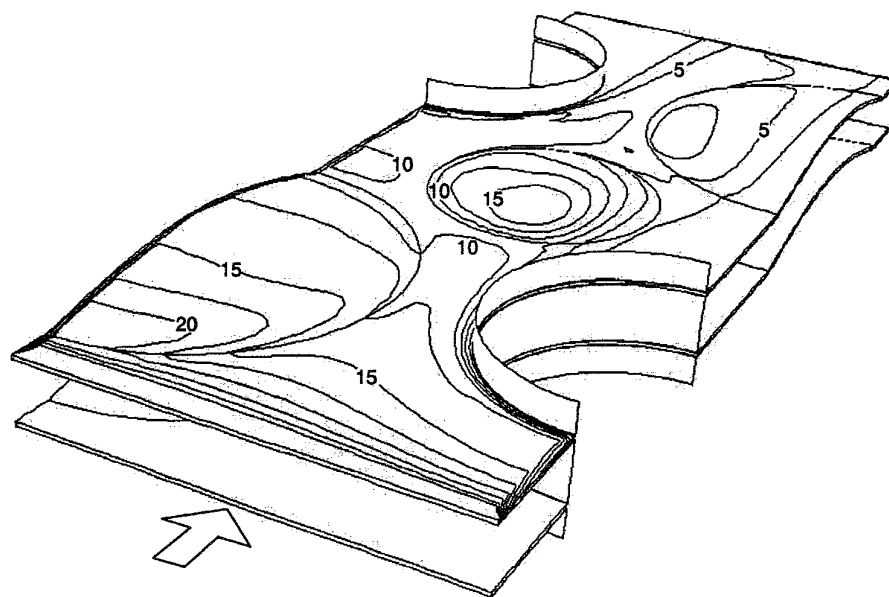
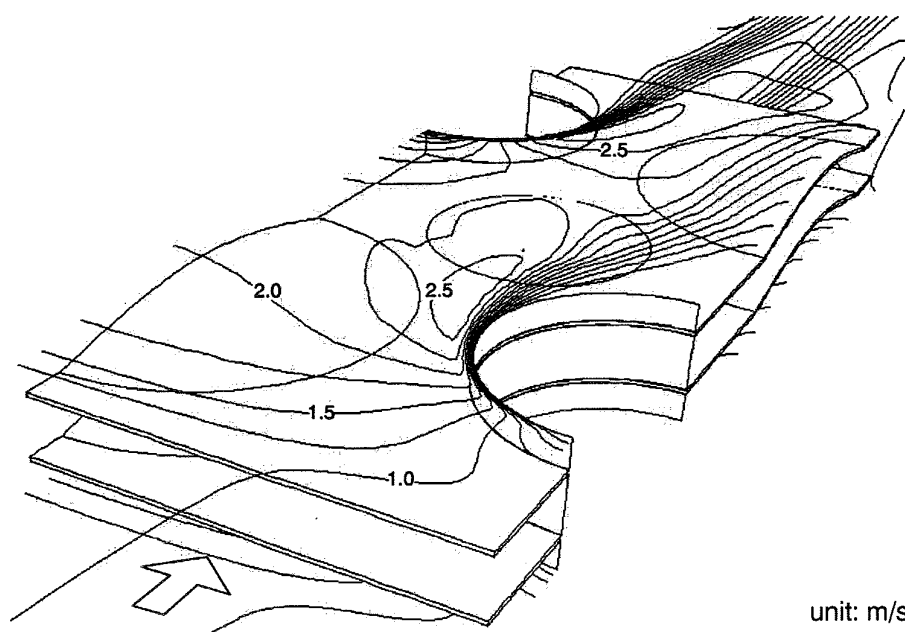


FIG.32A



unit: m/s

FIG.32B

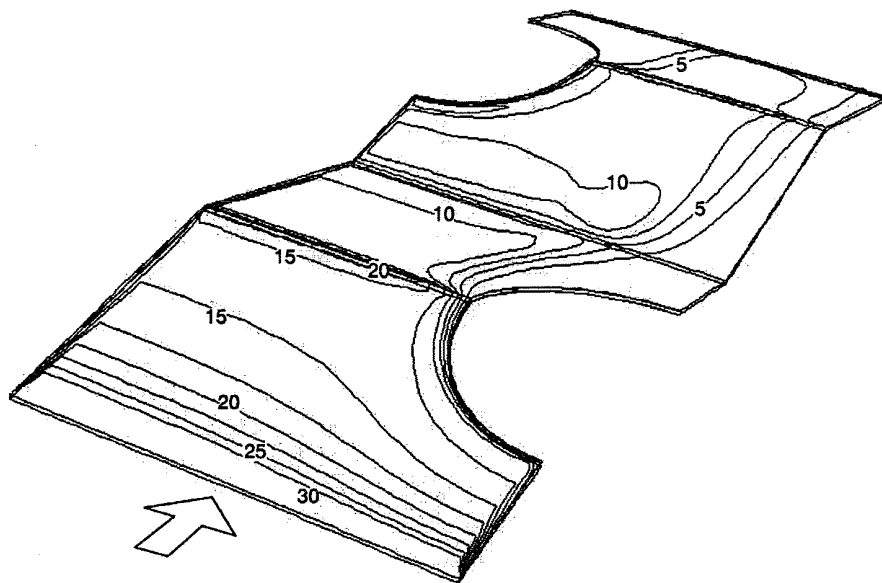


FIG.33A

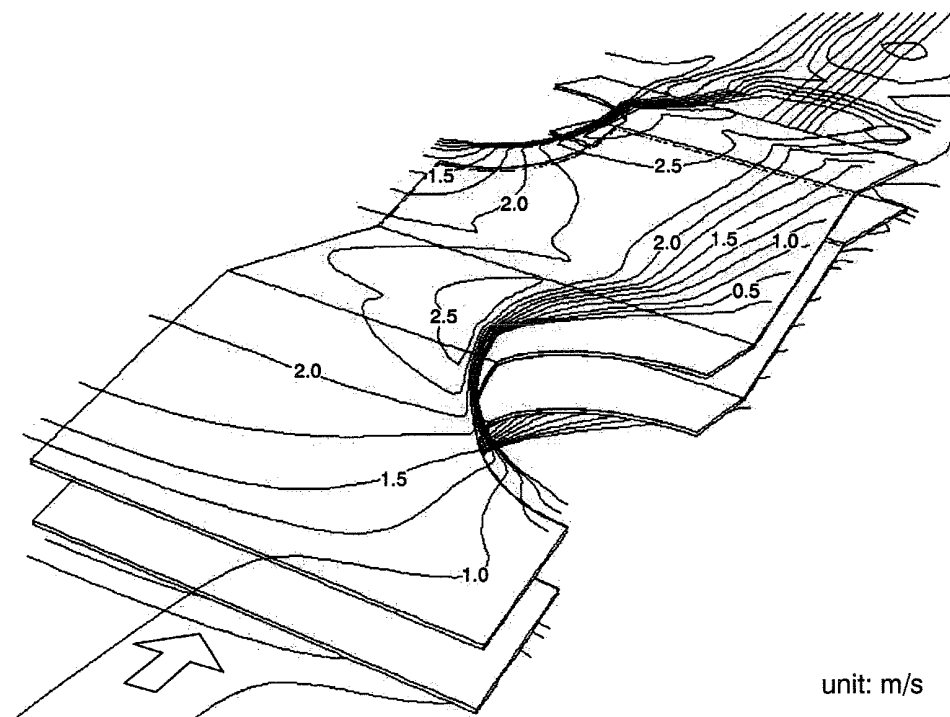


FIG.33B

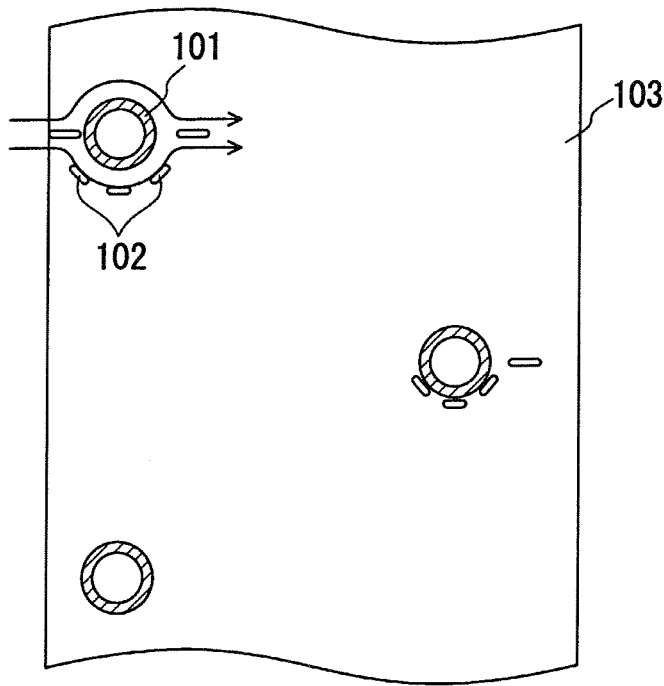


FIG.34

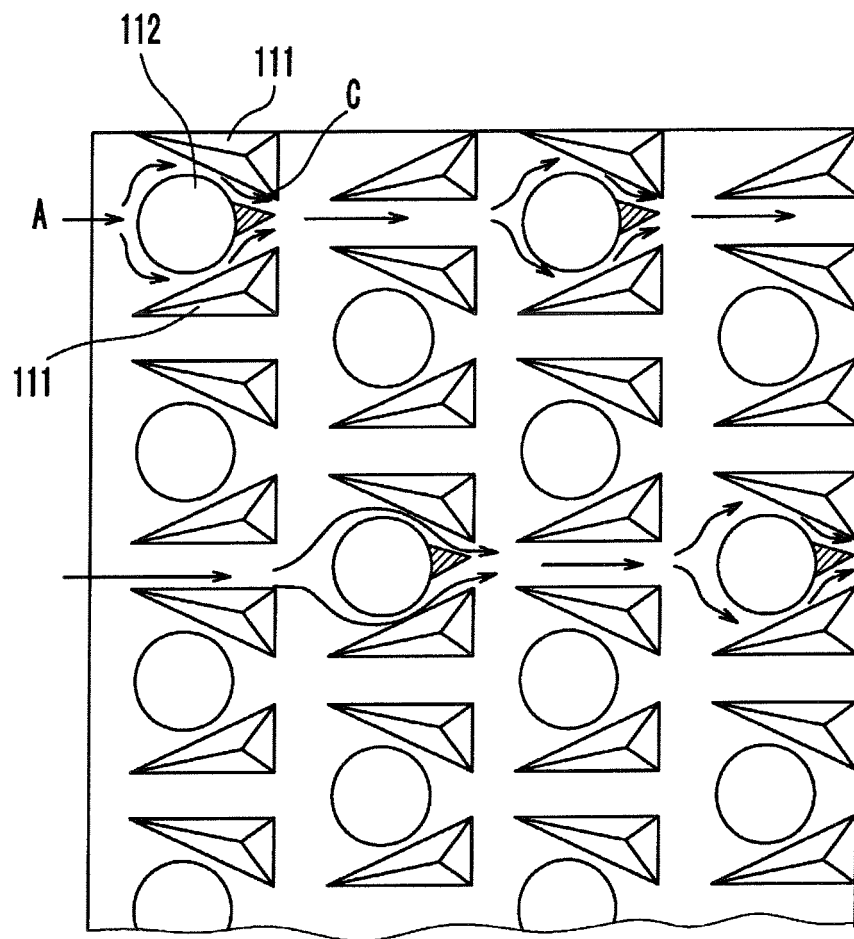


FIG.35

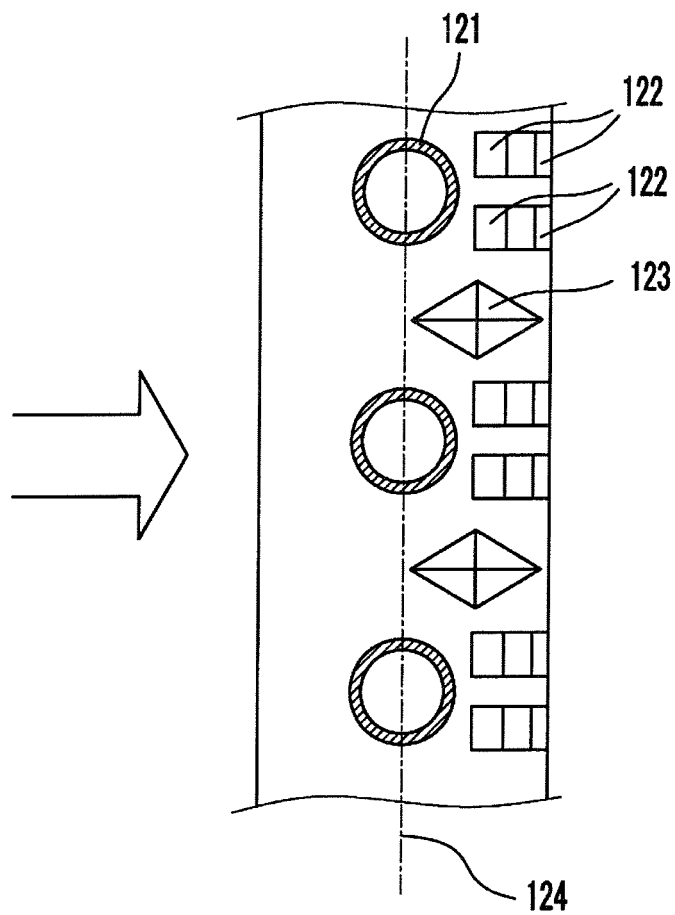


FIG.36

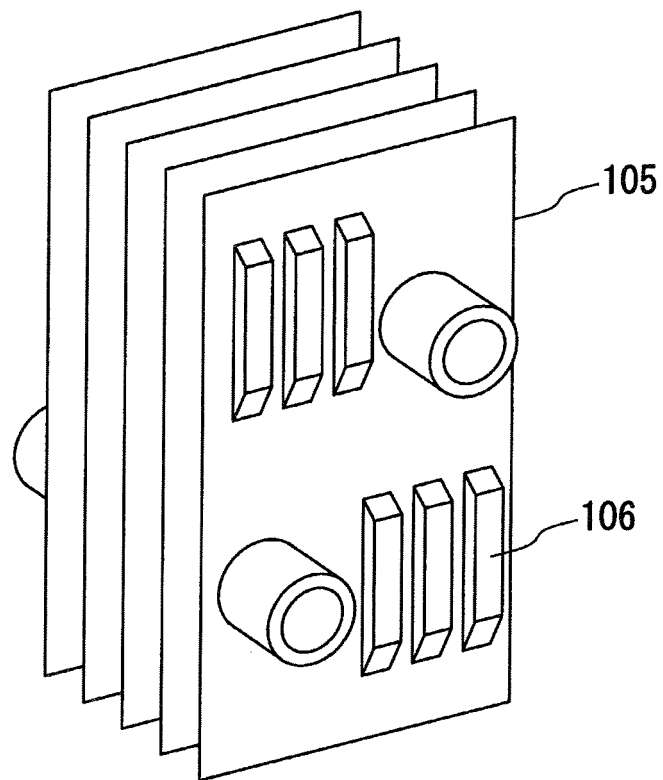


FIG.37

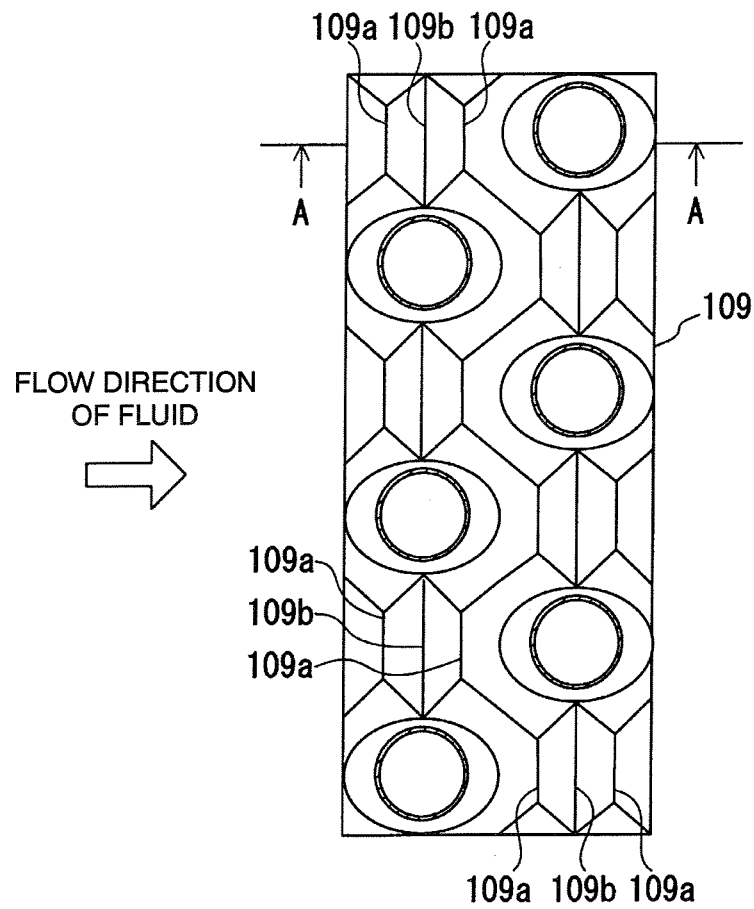


FIG. 38A

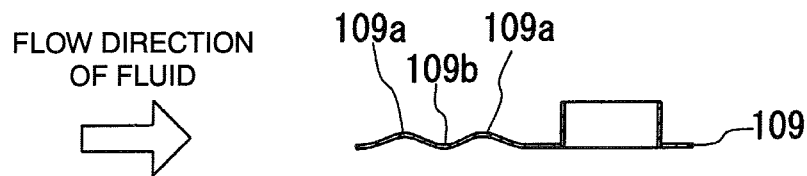


FIG. 38B

REFERENCES CITED IN THE DESCRIPTION

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