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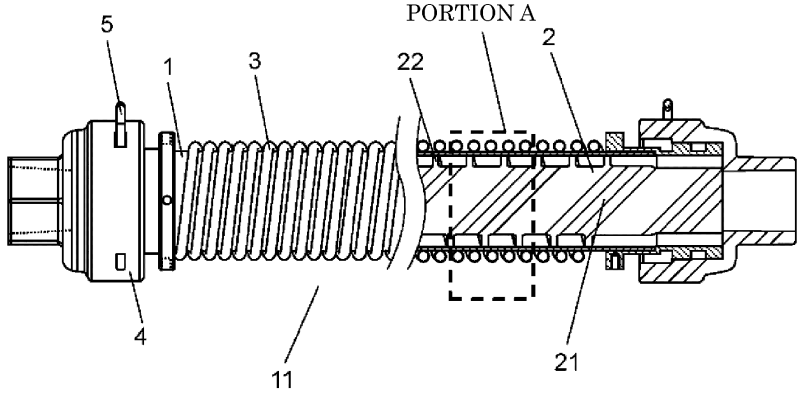
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(54) HEAT EXCHANGER AND REFRIGERATION CYCLE DEVICE USING SAME

(57) A heat exchanger according to the present invention includes an inner pipe in which first fluid flows and an insertion body inserted in the inner pipe. Furthermore, the heat exchanger includes an outer pipe that is provided at an outer periphery of the inner pipe and in which second fluid flows. The insertion body has a shaft portion and a spiral projection portion formed on an outer surface of the shaft portion. The first fluid flows in a spiral flow path formed by an inner surface of the inner pipe, the shaft portion, and the spiral projection portion. This configuration makes it possible to provide a heat exchanger that is compact, is superior in economic performance, and has high quality performance and high heat transfer performance.

FIG. 1



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Description

TECHNICAL FIELD

[0001] The present invention relates to a heat exchanger that exchanges heat between fluids.

BACKGROUND ART

[0002] Conventionally, as such type of a heat exchanger, a heat exchanger has been proposed in which a water pipe and a refrigerant pipe are wound in a double spiral shape (e.g., see PTL 1). Furthermore, a heat exchanger has been proposed in which a refrigerant pipe is wound around a water pipe (e.g., see PTL 2).

[0003] A heat pump hot water dispenser mounting thereon such a heat exchanger is a device that boils water over a predetermined time mainly during night time, and flow speed of water flowing in the heat exchanger equipped in the hot water dispenser is relatively slow during boiling operation.

[0004] Therefore, a flow of the water flowing in the heat exchanger is a laminar flow, so that in order to improve heat transfer performance as a heat exchanger, improving heat transfer performance on a water side is necessary by making the flow of water be disturbed.

[0005] FIG. 11 is a schematic view (partial cross sectional view) of a conventional heat exchanger described in PTL 1. FIG. 12 is an enlarged view illustrating a cross section of the heat exchanger in FIG. 11.

[0006] Heat exchanger 201 includes water pipe 202 and one or more of refrigerant pipe(s) 203 for one water pipe 202. Water pipe 202 is formed in a substantially cylindrical shape by being spirally wound. Refrigerant pipe 203 is spirally wound around an outer periphery of water pipe 202 formed in the substantially cylindrical shape at predetermined pitches. Furthermore, at least one portion of refrigerant pipe 203 is jointed across substantially entire length of water pipe 202. A direction of water flowing in water pipe 202 and a direction of refrigerant flowing in refrigerant pipe 203 are opposed directions.

[0007] Winding water pipe 202 in a spiral shape as described above makes centrifugal force act on the water flowing in the water pipe, causing a secondary flow as illustrated by arrows in FIG. 12 in a cross section perpendicular to a pipe axis. Herein, largeness of the centrifugal force acting on the water flowing in the spiral flow path is described by a following formula based on balance of force.

Formula 1

$$F = M\frac{v^2}{r}$$

[0008] Note that in Formula (1), F denotes centrifugal force, M ($M = V \times \rho$) denotes mass, V denotes cubic volume, ρ denotes density, v denotes rotation speed, and r denotes radius of rotation.

[0009] As is understood from Formula (1), larger centrifugal force exerts as fluid density becomes large due to lower temperature, making the fluid move toward outside of the spiral flow path. Consequently, a temperature difference between water and refrigerant at a heat transfer surface increases, facilitating heat transfer.

[0010] Therefore, a temperature field in the cross section perpendicular to main stream is improved by the secondary flow even when the flow of water is the laminar flow, making it possible to drastically improve heat transfer performance as compared with a heat exchanger having a straight pipe shape in which a water pipe and a refrigerant pipe are jointed.

[0011] FIG. 13 is a schematic view of a conventional heat exchanger described in PTL 2.

[0012] Heat exchanger 301 includes water pipe 302 having a straight portion and one or more refrigerant pipe(s) 303 for one water pipe 302. Refrigerant pipe 303 is wound around water pipe 302, and a twisted tape as a heat transfer facilitating means is inserted inside water pipe 302.

[0013] Making the twisted tape be inserted in the water pipe to generate a swirl flow makes a flow on a water side be disturbed to improve heat transfer performance.

[0014] However, the configuration in above PTL 1 forms a heat exchanger by winding the pipe in a spiral shape, so that the water pipe can be flattened or buckled depending on a material of the pipe, a diameter of the pipe, and a thickness of the pipe.

[0015] Therefore, curvature diameter D of the spiral tube needs to be made large to prevent buckling by increasing a thickness of the water pipe in consideration of thickness reduction due to flattening. This results in increase in cost and unfortunately makes a volume of the heat exchanger large. Furthermore, there is a problem in that heat transfer facilitation effect due to the secondary flow caused by centrifugal force becomes small.

[0016] Furthermore, when a winding pitch of the pipe is set wide, although risk of buckling is reduced, the heat exchanger unfortunately becomes a long one having a large dead space, so that there is also a problem in that a volume of the heat exchanger becomes needlessly large.

[0017] Furthermore, although the configuration in above PTL 2 improves temperature distribution near a heat transfer surface by the swirl flow generated due to the twisted tape, an effect of improving temperature distribution on a center axis of the water pipe that is the farthest portion from the heat transfer surface is small as compared with that near the heat transfer surface.

[0018] That is, a dead water area is unfortunately generated in which contribution to heat transfer is small on the center axis of the water pipe. Furthermore, making a diameter of the water pipe small to reduce the dead water area makes water pressure loss too large, increasing power for a water sending pump. Therefore, there is a problem in that a running coast of equipment mounting thereon the heat exchanger increases.

Citation List

Patent Literature

[0019]

PTL 1: Japanese Patent No. 4805179
PTL 2: Japanese Patent No. 4501446

SUMMARY OF THE INVENTION

[0020] The present invention solves the above conventional problems, and aims to provide a heat exchanger that is compact, is superior in economic performance, and has high quality performance and high heat transfer performance.

[0021] In order to achieve the above object, a heat exchanger according to the present invention includes an inner pipe in which first fluid flows, an insertion body inserted in the inner pipe, and an outer pipe in which second fluid flows, the outer pipe being provided at an outer periphery of the inner pipe. The insertion body has a shaft portion and a spiral projection portion formed on an outer surface of the shaft portion. The first fluid flows in a spiral flow path formed by an inner surface of the inner pipe, the shaft portion, and the spiral projection portion.

[0022] This makes it possible to form the spiral flow path in which the first fluid flows by two parts that are the inner pipe and the insertion body having the spiral projection portion, eliminating the need of winding the inner pipe to form a flow path. This prevents the inner pipe from being buckled and flattened, enabling a thickness of the pipe to be the minimum thickness, making it possible to provide a heat exchanger that is superior in economic efficiency and lightweight.

[0023] Furthermore, a curvature diameter of the spiral flow path can be set smaller than a curvature diameter of a conventional spiral flow path, making it possible to provide a heat exchanger that has a high heat transfer facilitating effect due the secondary flow and that is compact.

[0024] In addition, the maximum distance from a heat transfer surface of the first fluid is determined by a shaft diameter of the insertion body and a projection portion height of the spiral projection portion. This allows a flow path cross sectional area to be designed by changing a pitch of the spiral projection portion so as to be water pressure loss that can be allowed by a water sending pump. This makes it possible to provide a heat exchanger that drastically reduces a dead water area within a water pressure loss limitation range and that has high heat transfer performance.

[0025] The present invention makes it possible to provide a heat exchanger that is compact, is superior in economic performance, and has high quality performance and high heat transfer performance.

BRIEF DESCRIPTION OF DRAWINGS

[0026]

FIG. 1 is a schematic view of a heat exchanger according to a first exemplary embodiment of the present invention.
FIG. 2A is a perspective view illustrating a flow of fluid in an outer pipe of the heat exchanger according to the first exemplary embodiment of the present invention.
FIG. 2B is a perspective view illustrating a flow of fluid in an inner pipe of the heat exchanger.
FIG. 3 is an enlarged view of portion A in FIG. 1.
FIG. 4 is a diagram illustrating preliminary calculation results of a heat-transfer coefficient in the spiral circular pipe.
FIG. 5A is an appearance view of an insertion body of a heat exchanger according to a second exemplary embodiment of the present invention.
FIG. 5B is an enlarged view of portion B in FIG. 5A.

FIG. 6A is a cross sectional view of the heat exchanger according to the second exemplary embodiment of the present invention.

FIG. 6B is an enlarged view of portion C in FIG. 6A.

FIG. 7 is a perspective view of a joint and an insertion body of the heat exchanger according to the second exemplary embodiment of the present invention.

FIG. 8 is a detailed cross sectional view of a heat exchanger according to a third exemplary embodiment of the present invention.

FIG. 9 is a diagram illustrating a relationship between a distal end width of an insertion body and heat exchange capability.

FIG. 10 is a schematic configuration diagram of a refrigeration cycle device according to a fourth exemplary embodiment of the present invention.

FIG. 11 is a schematic view of a conventional heat exchanger.

FIG. 12 is an enlarged view illustrating a cross section of the heat exchanger of FIG. 11.

FIG. 13 is a schematic view of another conventional heat exchanger.

DESCRIPTION OF EMBODIMENTS

[0027] A heat exchanger according to a first invention includes an inner pipe in which first fluid flows, an insertion body inserted in the inner pipe, and an outer pipe in which second fluid flows, the outer pipe being provided at an outer periphery of the inner pipe. The insertion body has a shaft portion and a spiral projection portion formed on an outer surface of the shaft portion.

The first fluid flows in a spiral flow path formed by an inner surface of the inner pipe, the shaft portion, and the spiral projection portion.

[0028] This makes it possible to form the spiral flow path in which the first fluid flows by two parts that are the inner pipe and the insertion body having the spiral projection portion, making it possible to provide a heat exchanger that prevents the inner pipe from being buckled and flattened, that is superior in economic performance and is lightweight by making a thickness of the pipe be a requisite minimum.

[0029] Furthermore, a curvature diameter of the spiral flow path can be made smaller as compared with a curvature diameter of a conventional spiral flow path, making it possible to provide a heat exchanger that has a large heat transfer facilitation effect due to the secondary flow and that is compact.

[0030] In addition, the maximum distance from a heat transfer surface of the first fluid is set to a shaft diameter of the insertion body and a height of the spiral projection portion, and a flow path cross sectional area can be set so as to be a pressure loss that can be allowed by a sending pump, making it possible to provide a heat exchanger having high heat exchange performance in which a dead water area is drastically reduced within a pressure loss limitation range as compared with a conventional heat exchanger.

[0031] A second invention is configured such that a winding direction of the outer pipe and a spiral direction of the spiral projection portion are same directions, and a flow of the first fluid and a flow of the second fluid are configured to be opposed flows, specifically in the first invention.

[0032] This enables the first fluid and the second fluid to exchange heat by the opposed flow, making it possible to provide a heat exchanger having high heat exchange performance.

[0033] A third invention is configured such that the outer pipe is disposed at the outer periphery of the inner pipe and at an opposing portion of the spiral flow path, specifically in the first or second invention.

[0034] This enables the first fluid and the second fluid to exchange heat in almost the whole area of the heat exchanger, making it possible to provide a heat exchanger having higher heat exchange performance.

[0035] A fourth invention includes a joint for fixing the inner pipe and the insertion body, specifically in any one of the first to third inventions.

[0036] This makes a disposed position of the insertion body having the spiral projection portion in the inner pipe be fixed in any installation state (vertical placement, lateral placement, or oblique placement), making it possible to provide a heat exchanger in which installation freedom is improved.

[0037] In a fifth invention, the spiral projection portion includes a plurality of projections in contact with the inner pipe, specifically in any one of the first to fourth inventions. The plurality of projections is sequentially aligned along a shaft direction.

[0038] This makes a gap between the spiral projection portion and the inner pipe excluding the projection portions, making it possible to form a flow path communicated along a heat exchanger shaft direction in addition to the spiral flow path. This increases a bypass amount of the first fluid that flows in the gap during large flow rate in which centrifugal force acting on the first fluid is large while agitating the flow by the secondary flow.

[0039] This makes it possible to suppress increase of a pressure loss in the heat exchanger also during large flow rate to reduce a power required by a sending pump for sending the first fluid, improving energy saving performance of

equipment.

[0040] Furthermore, when an inflow temperature of the first fluid is high, it is necessary to increase flow rate of the first fluid to prevent an outflow temperature of the first fluid from becoming abnormally high. The present invention increases a flow rate capable of being sent by a pump having a same lifting height, making it possible to assure a flow rate enough to keep an output flow temperature of the first fluid at not more than a predetermined temperature, improving reliability of equipment.

[0041] A sixth invention satisfies, given that a distal end width and a proximal end width of the spiral projection portion are respectively t1 and t2, a relationship of $t1 < t2$ is satisfied, specifically in any one of the first to fifth inventions.

[0042] This enlarges a heat transfer area of the first fluid that flows in the spiral flow path formed between the inner pipe and the insertion body with respect to the second fluid that flows inside the outer pipe, making it possible to provide a heat exchanger having high heat exchange performance.

[0043] A seventh invention is a refrigeration cycle device including a refrigerant circuit in which at least a compressor, the heat exchanger according to any one of the first to sixth inventions, a decompressor, and an evaporator are circularly connected, and a controller. The refrigeration cycle device has a defrosting operation mode for defrosting frost formation of the evaporator, and the insertion body is made of a resin.

[0044] Making a portion of the flow path for the first fluid be formed by a resin having specific heat larger than that of a metal (copper: 0.04 $J/m^3 \cdot K$, PPS: 0.65$J/m^3 \cdot K$) increases the accumulated heat quantity of the heat exchanger, making it possible to use more heat quantity during defrosting from the heat exchanger.

This makes it possible to terminate defrosting operation within a short period, improving defrosting performance of equipment.

[0045] Hereinafter, exemplary embodiments of the present invention will be descried with reference to the drawings. Note that the present invention is not limited by the exemplary embodiments.

FIRST EXEMPLARY EMBODIMENT

[0046] FIG. 1 is a schematic view (partial cross sectional view) of heat exchanger 11 according to a first exemplary embodiment of the present invention.

[0047] Heat exchanger 11 according to the first exemplary embodiment of the present invention includes inner pipe 1, outer pipe 3 spirally wound around an outer surface of inner pipe 1 to be in close contact therewith, and insertion body 2 to be inserted inside inner pipe 1. Insertion body 2 includes insertion body shaft portion 21 and spiral projection portion 22.

[0048] A spiraled winding direction of outer pipe 3 and a spiral direction of spiral protection portion 22 are the same directions, and winding pitches thereof are also same.

[0049] Hereinafter, operation of the heat exchanger configured as described above will be described.

[0050] Heat exchanger 11 makes water that is first fluid and carbon dioxide that is second fluid exchange heat via inner pipe 1 and outer pipe 3.

[0051] In heat exchanger 11, a flow path in which water flows is a spiral flow path formed by an inner surface of inner pipe 1, an outer surface of the insertion body shaft portion 21 and adjacent spiral projection portions 22, and is formed by two parts that are inner pipe 1 and the insertion body 2 to be inserted in inner pipe 1.

[0052] Thus, it is not necessary to perform a bent process to form a water flow path, preventing inner pipe 1 from being buckled and flattened and enabling the thickness of inner pipe 1 to be the minimum thickness based on a design concept (thickness in consideration for pressure resistance + corrosion margin). This makes it possible to provide a heat exchanger that is superior in economic performance and that is light weight.

[0053] Next, curvature diameter D of the spiral flow path and heat-transfer coefficient in the pipe will be described.

[0054] For heat-transfer coefficient in a developed area in a curved circular pipe such as a spiral pipe, "Japan Society of Mechanical Engineers, heat-transfer engineering document, the revised fifth edition" describes the following.

Formula 2

$$Nu = \frac{\mathrm{Pr}^{0.3}}{24}\mathrm{Re}^{\frac{4}{5}}\left(\frac{d}{D}\right)^{\frac{1}{10}}\left[1+\frac{0.098}{\left\{\mathrm{Re}\left(\frac{d}{D}\right)^{2}\right\}^{\frac{1}{5}}}\right]$$

$$0.6 < \mathrm{Pr} \leq 1, \mathrm{Re}\left(\frac{d}{D}\right)^{2} > 0.1$$

Formula 3

$$Nu = \frac{\mathrm{Pr}^{0.4}}{41}\mathrm{Re}^{\frac{5}{6}}\left(\frac{d}{D}\right)^{\frac{1}{12}}\left[1+\frac{0.061}{\left\{\mathrm{Re}\left(\frac{d}{D}\right)^{2.5}\right\}^{\frac{1}{6}}}\right]$$

$$\mathrm{Pr} > 1, \mathrm{Re}\left(\frac{d}{D}\right)^{2.5} > 0.4$$

[0055] Herein, Nu denotes Nusselt number, Pr denotes Prandtl number, and Re denotes Reynolds number. Then, D denotes a curvature diameter of a center axis of the spiral flow path, and d is an equivalent diameter of pipe. FIG. 4 is an estimation of Nusselt number Nu using above Formula (3) when (d/D) is changed under the condition where Reynolds number Re = 2000, and water temperature is 40°C. The vertical axis denotes Nusselt number Nu and the lateral axis denotes d/D.

[0056] As is understood from above Formula (2), Formula (3), and FIG. 4, Nusselt number in the circular pipe becomes larger as equivalent diameter d of the pipe becomes larger or as curvature diameter D becomes smaller under the condition where Reynolds number and Prandtl number are constant.

[0057] That is, the heat-transfer coefficient in the pipe becomes high, improving heat-transfer performance of the heat exchanger. (d/D) of the of the heat exchanger as described in PTL 1 that is mounted on an existing heat pump hot water dispenser is not more than 0.2. To the contrary, in heat exchanger 11 of the present invention, the spiral flow path is structured by two parts, enabling curvature diameter D of the spiral flow path in which water flows to be drastically smaller as compared with a curvature diameter of a conventional spiral flow path. This increases (d/D), increasing agitation effect due to the secondary flow. This improves heat-transfer facilitation effect and makes it possible to provide a compact heat exchanger.

[0058] FIGS. 2A and 2B each are a perspective view illustrating a flow of fluid flowing in heat exchanger 11 according to the first exemplary embodiment of the present invention.

[0059] Water that is the first fluid flows in the spiral flow path formed by the inner surface of inner pipe 1, the outer surface of insertion body shaft portion 21, and adjacent spiral projection portions 22. Pitches of the spiral projection portion 22 of the insertion body 2 and the wounding direction are synchronized, and carbon dioxide that is the second fluid that flows inside outer pipe 3 wound around an opposing portion of the spiral flow path and water that is the first fluid exchange heat.

[0060] Herein, the water that flows in the spiral flow path between inner pipe 1 and insertion body 2 and the carbon dioxide that flows inside outer pipe 3 are inverse in their flowing directions, making it possible to exchange heat by the opposed flow across the substantially whole area of heat exchanger 11 as indicated by the flows illustrated in FIGS. 2A and 2B, making it possible to provide a high efficient heat exchanger.

[0061] Note that all of the portion of outer pipe 3 is not necessarily wound around the opposing portion of the spiral flow path as long as heat exchange efficiency required by equipment on which the heat exchanger is mounted can be provided. Furthermore, a plurality of outer pipes 3 in which the second fluid flows may be included and the plurality of outer pipes 3 may be alternately wound around the opposing portion of the spiral flow path.

[0062] FIG. 3 is a cross sectional view of heat exchanger 11 according to the first exemplary embodiment of the present invention. The water flow path of the heat exchanger includes two parts that are inner pipe 1 and insertion body 2, so that the maximum distance from a water side heat-transfer surface can be designed on the basis of diameter "a" of insertion body shaft portion 21 and projection portion height "th" of spiral projection portion 22.

[0063] Furthermore, flow path cross sectional area S can be designed by changing winding pitch P of spiral projection portion 22 of insertion body 2 so as to be water pressure loss that can be allowed by a water sending pump for sending water in equipment. This makes it possible to drastically reduce a dead water area within a water pressure loss limitation range. Herein, it is preferable that diameter "a" of insertion body shaft portion 21 and projection portion height "th" of spiral projection portion 22 be designed such that heat exchange performance satisfies a predetermined performance within the range of the following (Formula 4).

Formula 4

$$1.0\times10^{-2} < \frac{th}{a} < 5$$

[0064] Furthermore, in the first exemplary embodiment of the present invention, the flow path cross section of the spiral flow path that is a water flow path is formed to be a rectangular cross section by the inner surface of inner pipe 1, insertion body shaft portion 21, and spiral projection portion 22, readily generating eddy as compared with the case where the cross section is circular shape, increasing effect by the secondary flow.

[0065] As described above, in the first exemplary embodiment, the water flow path is structured by the two parts that are inner pipe 1 and insertion body 2 having spiral projection portion 22, forming the spiral flow path without winding inner pipe 1. This makes it possible to provide a heat exchanger that is lightweight and superior in economic efficiency in which a thickness of inner pipe 1 is made to be a requisite minimum.

[0066] Furthermore, curvature diameter D of the spiral flow path can be drastically reduced as compared with a curvature diameter of a conventional spiral flow path, making it possible to provide a heat exchanger that is compact and that has high heat-transfer performance.

[0067] In addition, the maximum distance from the heat transfer surface of the water side flow path can be designed by diameter "a" of insertion body shaft portion 21 and height "th" of the projection portion of spiral projection portion 22, and flow path cross sectional area S can be designed by changing winding pitch P of spiral projection portion 22 such that water pressure loss becomes within a limitation. This makes it possible to provide a heat exchanger having high heat transfer performance in which the dead water area is drastically reduced within a limitation range of water pressure loss.

SECOND EXEMPLARY EMBODIMENT

[0068] FIGS. 5A and 5B each are an enlarged view of spiral projection portion 22 of insertion body 2 of heat exchanger 11 according to a second exemplary embodiment. FIGS. 6A and 6B each are a cross sectional view of the heat exchanger according to the second exemplary embodiment. FIG. 7 is a perspective view of a joint and an insertion body of the heat exchanger according to the second exemplary embodiment.

[0069] Note that the same numeral references are assigned to the same parts as those in the first exemplary embodiment of the present invention, and their detailed description will be omitted.

[0070] As illustrated in FIG. 5B, projections 25 that are sequentially aligned are provided along a shaft direction of heat exchanger 11, that is, along a shaft direction of insertion body 2 on the outer surface of spiral projection portion 22 of insertion body 2 forming heat exchanger 11 of the second exemplary embodiment. Furthermore, as illustrated in FIG. 7, an end in the shaft direction of insertion body 2 has convex portions 23, and joint 4 has concave portions 24 to be respectively engaged with convex portions 23 at the end of insertion body 2.

[0071] Insertion body 2 is fixed such that convex portions 23 of the end in the shaft direction of insertion body 2 and concave portions 24 of joint 4 are respectively fitted and projections 25 on the outer surface of spiral projection portion 22 are in contact with inner pipe 1.

[0072] Note that, although shapes of fitting portions of insertion body 2 and joint 4 are respectively the convex portion

and the concave portion in the second exemplary embodiment, the shapes thereof may be any shapes as long as the portions can be fitted.

[0073] Hereinafter, operation of the above configured heat exchanger will be described.

[0074] In the present exemplary embodiment, a gap exists between spiral projection portion 22 excluding projections 25 and inner pipe 1, so that a flow path (bypass flow path 50) communicated along the shaft direction of heat exchanger 11, that is, along the shaft direction of insertion body 2 is formed in addition to the spiral flow path described in the first exemplary embodiment.

[0075] Also in heat exchanger 11 of the second exemplary embodiment, like the first exemplary embodiment, water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 and carbon dioxide that is the second fluid that flows inside outer pipe 3 exchange heat by the opposed flow via inner pie 1 and outer pipe 3.

[0076] Herein, when a temperature of influent water that flows in heat exchanger 11 is high, heated water can be disadvantageously boiled in heat exchanger 11, so that adjustment is performed such that a temperature of output hot water becomes not more than a predetermined temperature by increasing the flow rate of the water to be sent to heat exchanger 11.

[0077] However, in the conventional heat exchanger described in above PTL 1, a spiral flow path is formed by winding a pipe, increasing a flow path length as compared with a straight flow path. Thus, water pressure loss in the heat exchanger becomes large during large flow rate, so that there is a problem in that pump power of equipment that sends water becomes too large to disadvantageously diminish energy saving performance.

[0078] Furthermore, when the water pressure loss in heat exchanger 11 exceeds sending capability of the pump, the temperature of output hot water fails to be kept at not more than a predetermined temperature, disadvantageously diminishing reliability of the equipment.

[0079] In contrast, heat exchanger 11 of the second exemplary embodiment has bypass flow path 50 communicated along the shaft direction of heat exchanger 11, that is, along the shaft direction of insertion body 2 between an inner surface of inner pipe 1 and spiral projection portion 22 excluding inner pipe 1 and projections 25 as illustrated in FIGS. 6A and 6B.

[0080] Consequently, a bypass amount of water that flows in the flow path communicated in the shaft direction of heat exchanger 11, that is, along the shaft direction of insertion body 2 increases during large flow rate when centrifugal force acting on the water is large while the flow is agitated by secondary flow.

[0081] Increase of pressure loss during large flow rate can be thus suppressed as compared with the conventional heat exchanger described in above PTL 1, which reduces power required by a sending pump, improving energy saving performance of the equipment.

[0082] Furthermore, increase of water pressure loss can be suppressed, which increases flow rate capable of being sent by a pump having a same lifting height, making it possible to secure flow rate enough to keep the temperature of output hot water to be flown away at not more than a predetermined temperature, improving reliability of the equipment.

[0083] Furthermore, joint 4 is fitted with insertion body 2, and joint 4 covers inner pipe 1 from outside to be fixed by a fastening body such as insertion pin 5 (see FIG. 1), fixing positions of insertion body 2 and inner pipe 1. This makes it possible to secure the flow path that is communicated along the shaft direction of the heat exchanger 11, that is, along the shaft direction of insertion body 2 between spiral projection portion 22 and inner pipe 1 in any installation state (vertical placement, lateral placement, or oblique placement).

[0084] This makes it possible to provide a heat exchanger improved in installation freedom as well as suppressing increase of pressure loss.

[0085] As described above, the second exemplified embodiment has projections 25 sequentially aligned along the shaft direction of heat exchanger 11 on the outer surface of spiral projection portion 22 of insertion body 2, and inner pipe 1 and insertion body 2 are fixed by joint 4 such that projections 25 and the inner surface of inner pipe 1 are in contact. This makes it possible to form a flow path also in the shaft direction of heat exchanger 11 in addition to the spiral flow path, making it possible provide heat exchanger 11 that suppresses increase of water pressure loss also in the case where water that flows in heat exchanger 11 is large flow rate. This improves energy saving performance of equipment mounting thereon heat exchanger 11 of the second exemplary embodiment.

[0086] Note that, also when there are no projections 25, making joint 4 fit with insertion body 2 to cover inner pipe 1 from outside and be fixed by a fastening body (see FIG. 5A) makes it possible to secure the flow path (bypass flow path 50) communicated along the shaft direction of heat exchanger 11, that is, along the shaft direction of insertion body 2 between spiral projection portion 22 and inner pipe 1 in any installation state (vertical placement, lateral placement, or oblique placement). Therefore, setting a distance between spiral projection portion 22 and inner pipe 1 at an appropriate distance makes it possible to provide heat exchanger 11 improved in installation freedom as well as suppressing increase of water pressure loss.

THIRD EXEMPLARY EMBODIMENT

[0087] FIG. 8 is a cross sectional view of a heat exchanger according to a third exemplary embodiment. Note that the same numeral references are assigned to the same parts as those in the first and second exemplary embodiments of the present invention, and their detailed description will be omitted.

[0088] The heat exchanger according to a fourth exemplary embodiment of the present invention is configured such that the relationship between distal end width t1 and proximal end width t2 of spiral projection portion 22 of insertion body 2 satisfies t1 < t2.

[0089] Hereinafter, operation of the heat exchanger configured as described above will be described.

[0090] Like the first and second exemplary embodiments, also in heat exchanger 11 according to the fourth embodiment, water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 and carbon dioxide that is the second fluid that flows inside outer pipe 3 exchange heat by the opposed flow via inner pie 1 and outer pipe 3.

[0091] Width L of a heat transfer surface for water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 with respect to carbon oxide that is the second fluid that flows inside outer pipe 3 of heat exchanger 11 is P - t1 obtained by subtracting distal end width t1 of spiral projection portion 22 from spiral pitch P of spiral projection portion 22 as illustrated in FIG. 8.

[0092] In the present exemplary embodiment, as illustrated in FIG. 8, the shape of spiral projection portion 22 of insertion body 2 is formed to satisfy t1< t2. This makes it possible to increase width L of the heat transfer surface for water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 with respect to carbon dioxide that is the second fluid that flows inside outer pipe 3 as compared with the case where a thickness of the spiral projection portion 22 is constant while keeping water side flow path cross sectional area S same as that in the case where a thickness of the spiral projection portion 22 is constant as illustrated in FIG. 3 for the first exemplary embodiment.

[0093] That is, heat transfer area for water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 with respect to carbon dioxide that is the second fluid that flows inside outer pipe 3 increases, making it possible to provide a heat exchanger having higher transfer performance.

[0094] FIG. 9 illustrates a relationship between insertion projection distal end width t1 and heat exchange capability Q under the conditions in which a length of the spiral flow path formed between inner pipe 1 and insertion body 2 and water side flow path cross sectional area S are constant, that is, under the condition in which water side pressure loss is equivalent.

[0095] As is apparent from FIG. 9, the heat transfer area for water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 with respect to carbon oxide that is the second fluid that flows inside outer pipe 3 becomes larger as distal end width t1 of spiral projection portion 22 becomes smaller. This improves heat exchange performance.

[0096] Alternatively, a proximal shape of spiral projection 22 may be R-character shape to suppress separation of the secondary flow at a proximal portion and reduce water side pressure loss. This makes it possible to reduce friction loss of water due to eddy, making it possible to improve energy efficiency of the heat exchanger of the present exemplary embodiment or equipment mounting thereon the heat exchanger of the present exemplary embodiment.

[0097] As described above, in the third exemplary embodiment, the relationship between distal end width t1 and proximal end width t2 of the spiral projection 22 of insertion body 2 satisfies t1 < t2. This makes it possible to lengthen the length of heat transfer surface for water that is the first fluid that flows in the spiral flow path formed between inner pipe 1 and insertion body 2 with respect to carbon oxide that is the second fluid that flows inside outer pipe 3 without changing water side flow path conditions (length of the spiral flow path formed between inner pipe 1 and insertion body 2 and water side flow path cross sectional area S), that is, under an equivalent water side pressure loss condition. This makes it possible to increase the heat transfer area, making it possible to provide a heat exchanger having high heat exchange performance.

FOURTH EXEMPLARY EMBODIMENT

[0098] FIG. 10 is a configuration diagram of a refrigeration cycle device according to a fourth exemplary embodiment.

[0099] Note that the same numeral references are assigned to the same parts as those in the first to third exemplary embodiments of the present invention, and their detailed description will be omitted.

[0100] FIG. 10 is a refrigeration cycle device to be mounted on, for example, a heat pump hot water dispenser. The refrigeration cycle device includes compressor 101, radiator 102 that is heat exchanger 11 according to any of first to third exemplary embodiments of the present invention, decompressor 103 that is an electronic expansion valve, and evaporator 104, which are circularly connected to form refrigerant circuit 105.

[0101] The refrigerant circuit includes evaporator outlet port temperature detection means 107 for detecting a temper-

ature of refrigerant flown away from evaporator 104, and the refrigeration cycle device has controller 110 and a defrosting operation mode.

[0102] Carbon oxide as refrigerant is enclosed in refrigerant circuit 105, and a high-pressure side of compressor 101 is operated in a super criticality state during operation of compressor 101.

[0103] Furthermore, insertion body 2 having spiral projection portion 22 structuring radiator 102 (heat exchanger 11 according to the first exemplary embodiment or the second exemplary embodiment of the present invention) is made of a resin having volumetric specific heat larger than that of a metal (copper: 0.04 $J/m^3 \cdot K$, PPS: 0.65$J/m^3 \cdot K$).

[0104] Hereinafter, operation and behavior of the refrigeration cycle device configured as described above will be described.

[0105] Upon operation of compressor 101, the refrigerant compressed to be a high pressure to be ejected is sent to radiator 102 to release heat by exchanging heat with low temperature water sent by water sending pump 113 via inflow water pipe 111. This makes the heated low temperature water become high temperature water, and the high temperature water is sent to a hot water storage tank (not shown) via hot water outflow pipe 112 to be stored as high temperature hot water.

[0106] The refrigerant flown away from radiator 102 is supplied to decompressor 103 to be decompressed and expanded to be sent to evaporator 104, and the refrigerant exchanges heat with air introduced by air blower 106 to be evaporated to be gasified. The gasified refrigerant is suctioned in compressor 101.

[0107] Next, defrosting operation of the heat pump hot water dispenser will be described.

[0108] When hot water storage operation is performed in a state where an outside air temperature is low, frost is attached to evaporator 104, disadvantageously drastically lowering heat exchange performance of evaporator 104.

[0109] Therefore, controller 110 performs defrosting operation for defrosting the frost attached to evaporator 104 to recover heat exchange performance of evaporator 104. The defrosting operation is performed when frost is attached to evaporator 104 and the temperature detected by evaporator outlet port temperature detection means 107 falls below a predetermined temperature. Hereinafter the defrosting operation will be specifically described.

[0110] First, controller 110 makes water sending pump 113 for sending water to radiator 102 and air blower 106 stop their operation to reduce a flow path friction of decompressor 103. The high temperature refrigerant compressed by compressor 101 passes through radiator 102 and decompressor 103, flows in evaporator 104 to perform defrosting by heat owned by the refrigerant, and suctioned in compressor 101.

[0111] Then, when the temperature detected by evaporator outlet port temperature detection means 107 exceeds a predetermined temperature, the defrosting operation is terminated and boiling operation is performed.

[0112] During the defrosting operation, evaporator 104 is defrosted by utilizing heat quantity accumulated in radiator 102 in addition to heat quantity of the refrigerant ejected from compressor 101.

[0113] Making insertion body 2 that is a portion of the flow path for radiator 102 be made of a resin having specific heat larger than that of a metal (copper: 0.04 $J/m^3 \cdot K$, PPS: 0.65$J/m^3 \cdot K$) increases heat quantity accumulated in radiator 102, making it possible to utilize larger heat quantity from radiator 102 during defrosting. This makes it possible to terminate the defrosting operation in a short period, improving defrosting performance of equipment.

[0114] Note that in the fourth exemplary embodiment of the present invention, insertion body 2 having spiral projection portion 22 shall be made of a resin (PPS), but the same function effect can be expected as long as a resin other than PPS or a material having large volumetric specific heat is used.

[0115] In the first to third exemplary embodiments of the present invention, the refrigerant that flows in outer pipe 3 shall be carbon oxide, but the same function effect can be expected by using refrigerant of hydrocarbon system or HFC system (R410A, etc.) or substitute refrigerant thereof.

[0116] Note that not only each of the above exemplary embodiments but also any combination of the above exemplary embodiments is included in the scope of the present invention.

INDUSTRIAL APPLICABILITY

[0117] As described above, the heat exchanger according to the present invention makes it possible to provide a heat exchanger that is compact, superior in economic performance, and high in quality performance and heat exchange performance. Therefore, the present invention is applicable to equipment mounting thereon a heat exchanger that exchanges heat between fluids.

REFERENCE MARKS IN THE DRAWINGS

[0118]

1: inner pipe
2: insertion body

3: outer pipe
4: joint
5: stop pin (insertion pin)
11: heat exchanger
21: insertion body shaft portion
22: spiral projection portion
23: convex portion
24: concave portion
25: projection
50: bypass flow path
101: compressor
102: radiator
103: decompressor
104: evaporator
105: refrigerant circuit

Claims

1. A heat exchanger comprising:

an inner pipe in which first fluid flows;
an insertion body inserted in the inner pipe; and
an outer pipe in which second fluid flows, the outer pipe being provided at an outer periphery of the inner pipe, wherein
the insertion body has a shaft portion and a spiral projection portion formed on an outer surface of the shaft portion, and
the first fluid flows in a spiral flow path formed by an inner surface of the inner pipe, the shaft portion, and the spiral projection portion.

2. The heat exchanger according to claim 1, wherein
a winding direction of the outer pipe and a spiral direction of the spiral projection portion are same directions, and
a flow of the first fluid and a flow of the second fluid are configured to be opposed flows.

3. The heat exchanger according to claim 1 or 2, wherein the outer pipe is disposed at the outer periphery of the inner pipe and at an opposing portion of the spiral flow path.

4. The heat exchanger according to any one of claims 1 to 3 comprising a joint for fixing the inner pipe and the insertion body.

5. The heat exchanger according to any one of claims 1 to 4, wherein the spiral projection portion includes a plurality of projections in contact with the inner pipe, and
the plurality of projections is sequentially aligned along a shaft direction.

6. The heat exchanger according to any one of claims 1 to 5, wherein given that a distal end width and a proximal end width of the spiral projection portion are respectively t1 and t2, a relationship of $t1 < t2$ is satisfied.

7. A refrigeration cycle device comprising:

a refrigerant circuit in which at least a compressor, the heat exchanger according to any one of claims 1 to 6, a decompressor, and an evaporator are circularly connected; and
a controller, wherein
the refrigeration cycle device has a defrosting operation mode for defrosting frost formation of the evaporator, and the insertion body is made of a resin.

Amended claims under Art. 19.1 PCT

1. (Currently Amended) A heat exchanger comprising:

an inner pipe in which first fluid flows;
an insertion body inserted in the inner pipe; and
an outer pipe in which second fluid flows, the outer pipe being provided at an outer periphery of the inner pipe, wherein
the insertion body has a shaft portion and a spiral projection portion formed on an outer surface of the shaft portion,
the first fluid flows in a spiral flow path formed by an inner surface of the inner pipe, the shaft portion, and the spiral projection portion, and
a flow of the first fluid and a flow of the second fluid are configured to be opposed flows.

2. (Currently Amended) The heat exchanger according to claim 1, wherein
a winding direction of the outer pipe and a spiral direction of the spiral projection portion are same directions.

3. The heat exchanger according to claim 1 or 2, wherein
the outer pipe is disposed at the outer periphery of the inner pipe and at an opposing portion of the spiral flow path.

4. The heat exchanger according to any one of claims 1 to 3 comprising a joint for fixing the inner pipe and the insertion body.

5. The heat exchanger according to any one of claims 1 to 4, wherein
the spiral projection portion includes a plurality of projections in contact with the inner pipe, and
the plurality of projections is sequentially aligned along a shaft direction.

6. The heat exchanger according to any one of claims 1 to 5, wherein given that a distal end width and a proximal end width of the spiral projection portion are respectively t1 and t2, a relationship of $t1 < t2$ is satisfied.

7. A refrigeration cycle device comprising:

a refrigerant circuit in which at least a compressor, the heat exchanger according to any one of claims 1 to 6, a decompressor, and an evaporator are circularly connected; and
a controller, wherein
the refrigeration cycle device has a defrosting operation mode for defrosting frost formation of the evaporator, and
the insertion body is made of a resin.

FIG. 1

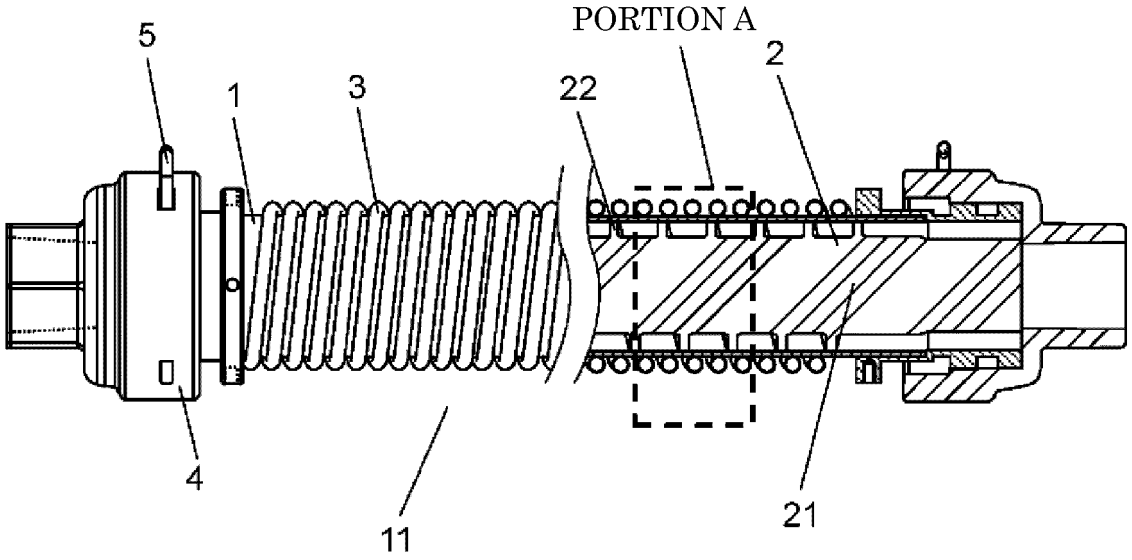


FIG. 2A

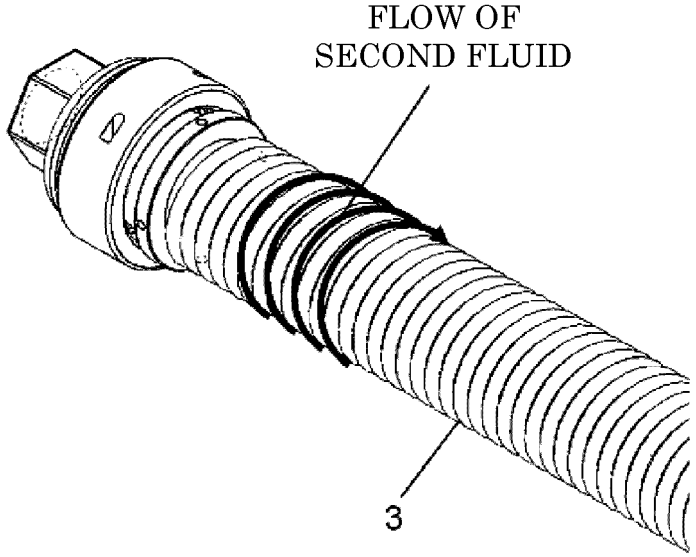


FIG. 2B

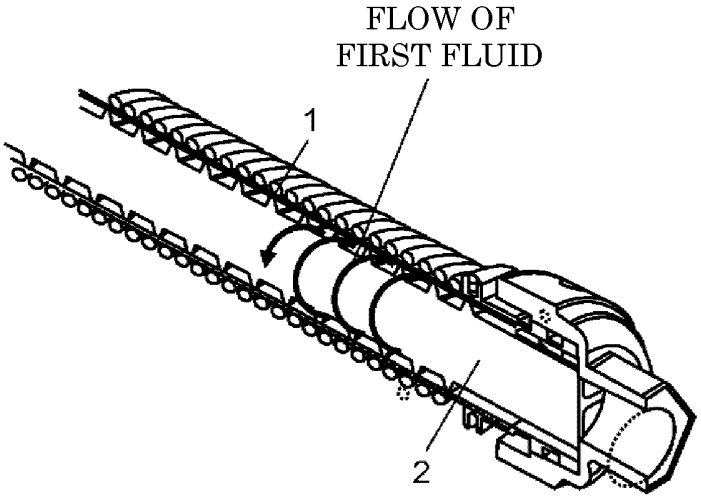

FLOW OF
FIRST FLUID
1
2

FIG. 3

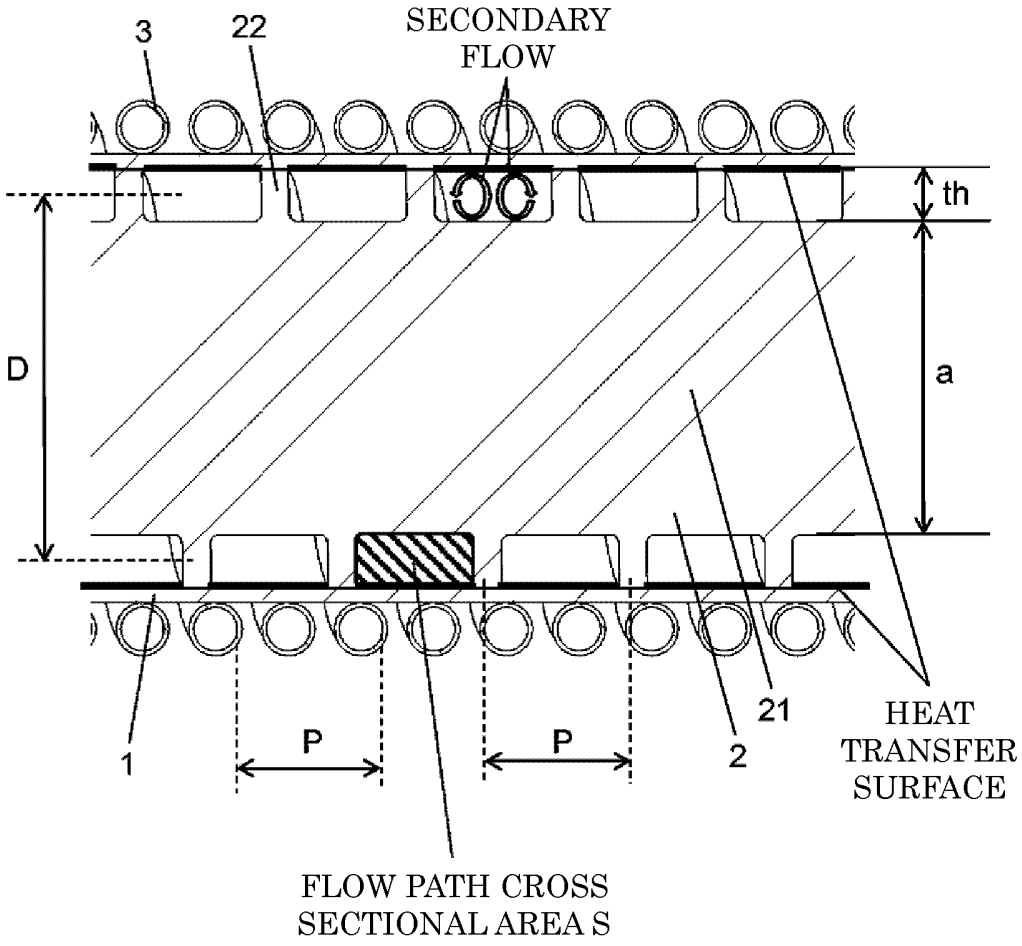

3
22
SECONDARY
FLOW
th
D
a
1
P
P
2
21
HEAT
TRANSFER
SURFACE
FLOW PATH CROSS
SECTIONAL AREA S

FIG. 4

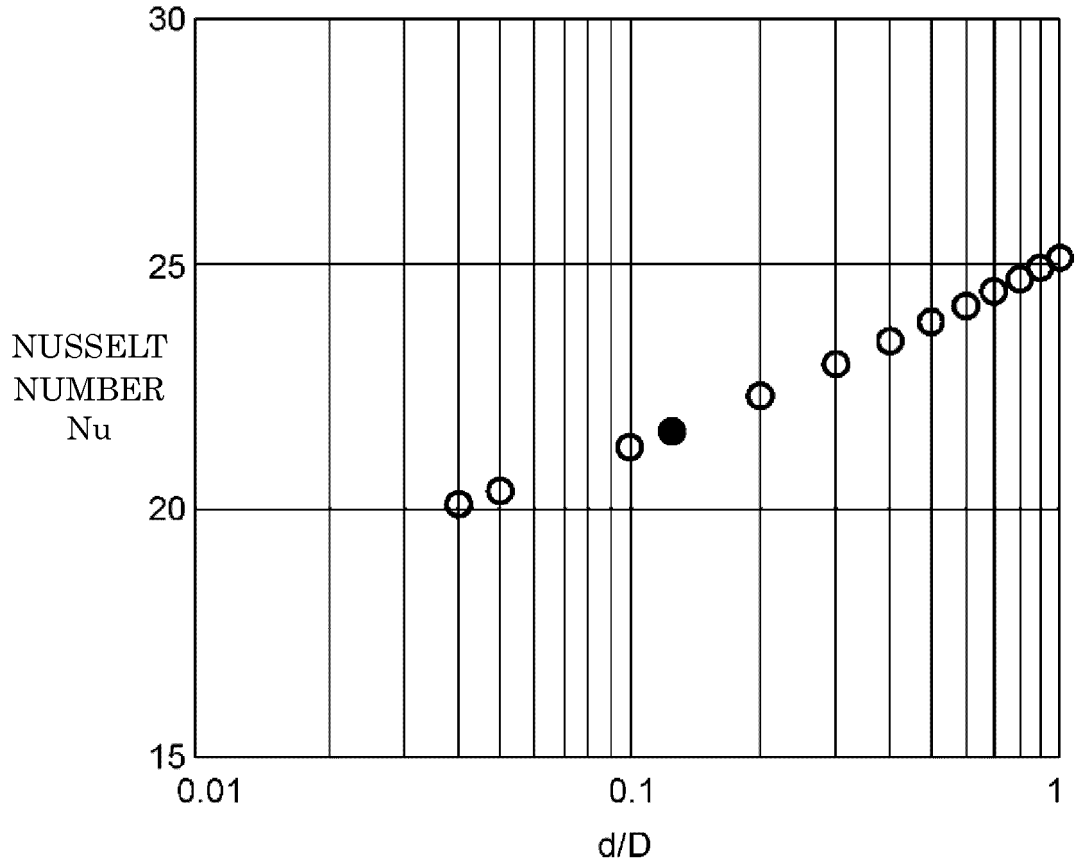


FIG. 5A

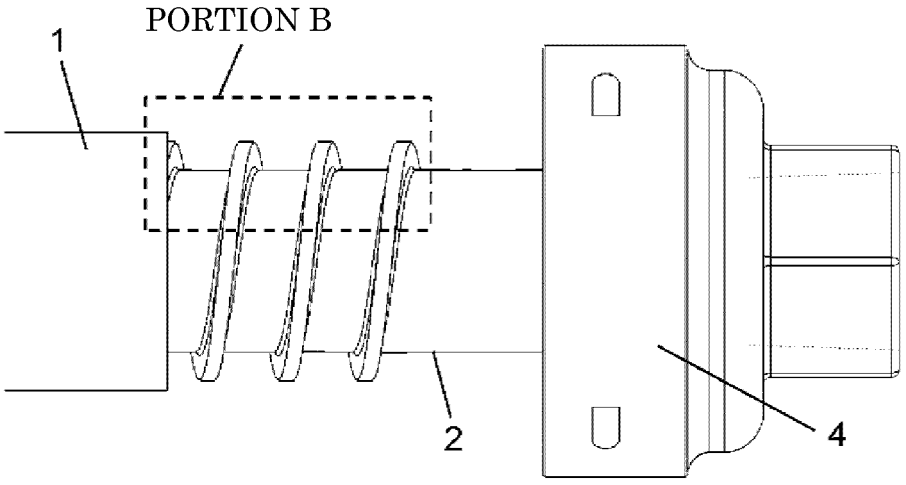


FIG. 5B

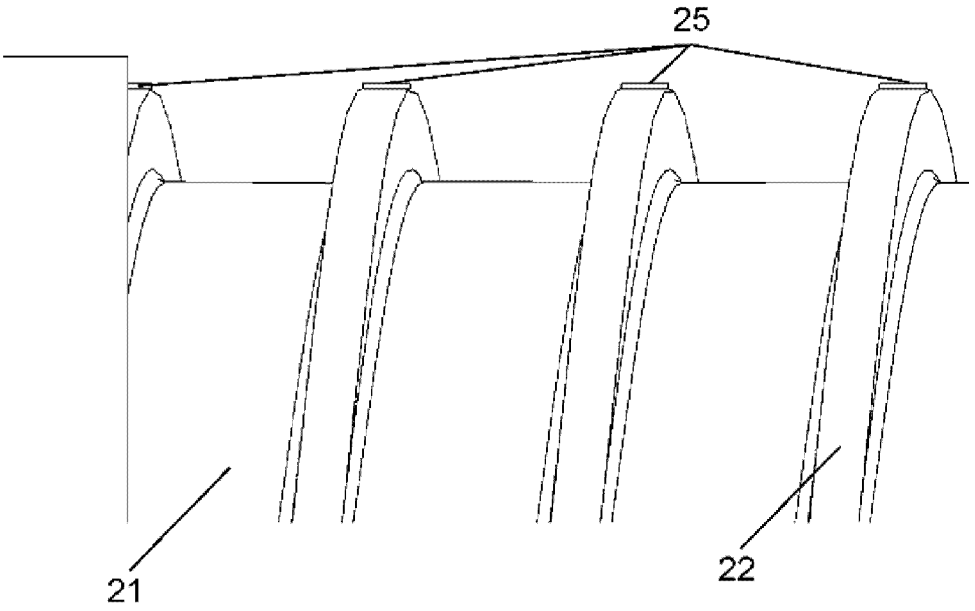


FIG. 6A

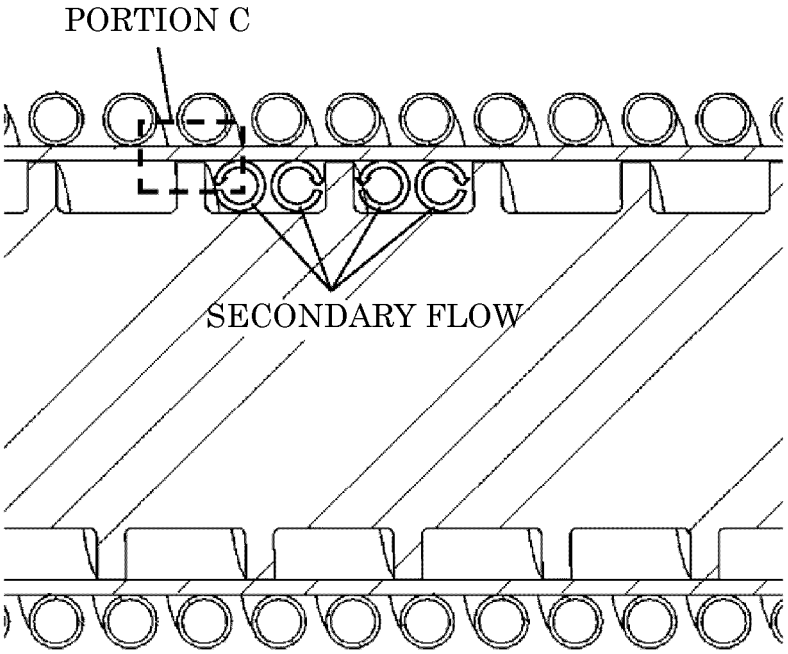


FIG. 6B

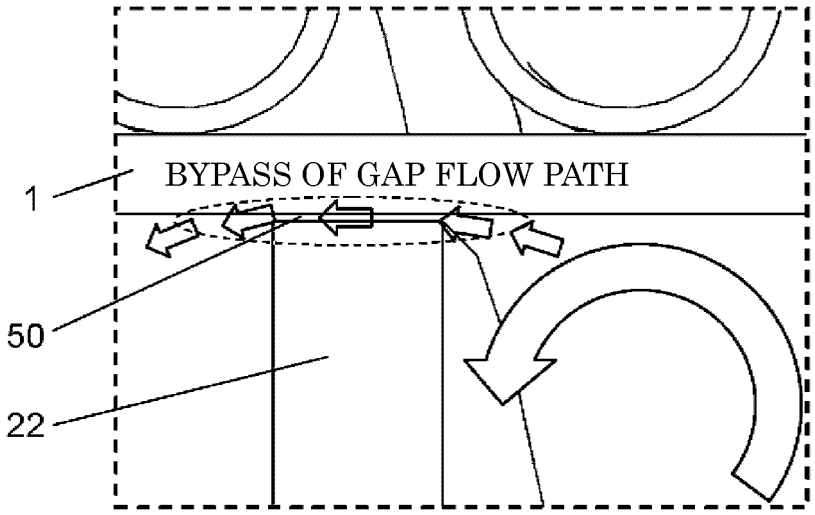


FIG. 7

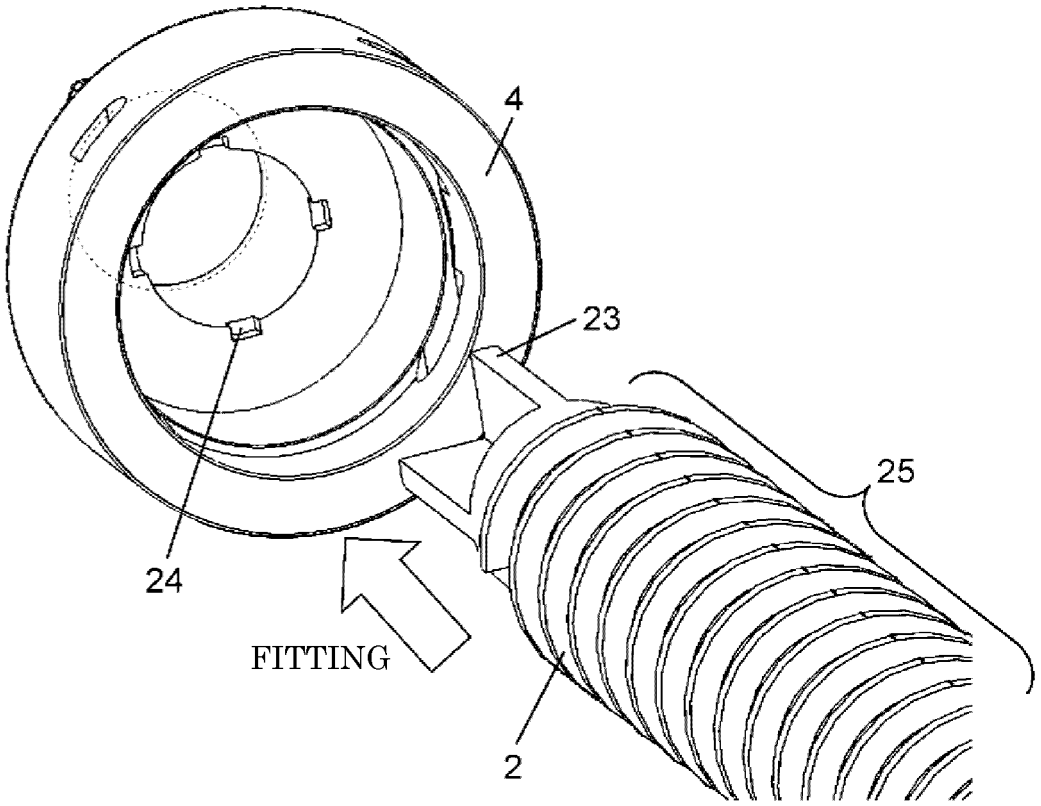


FIG. 8

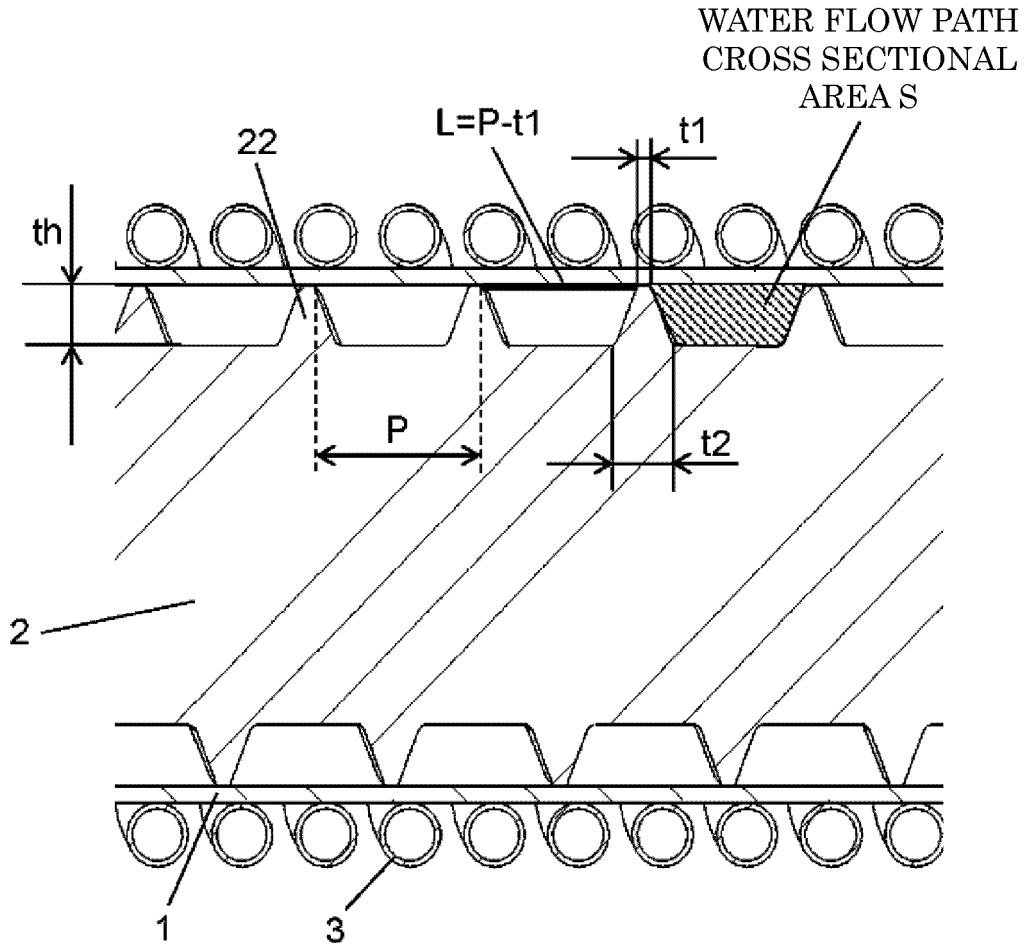


FIG. 9

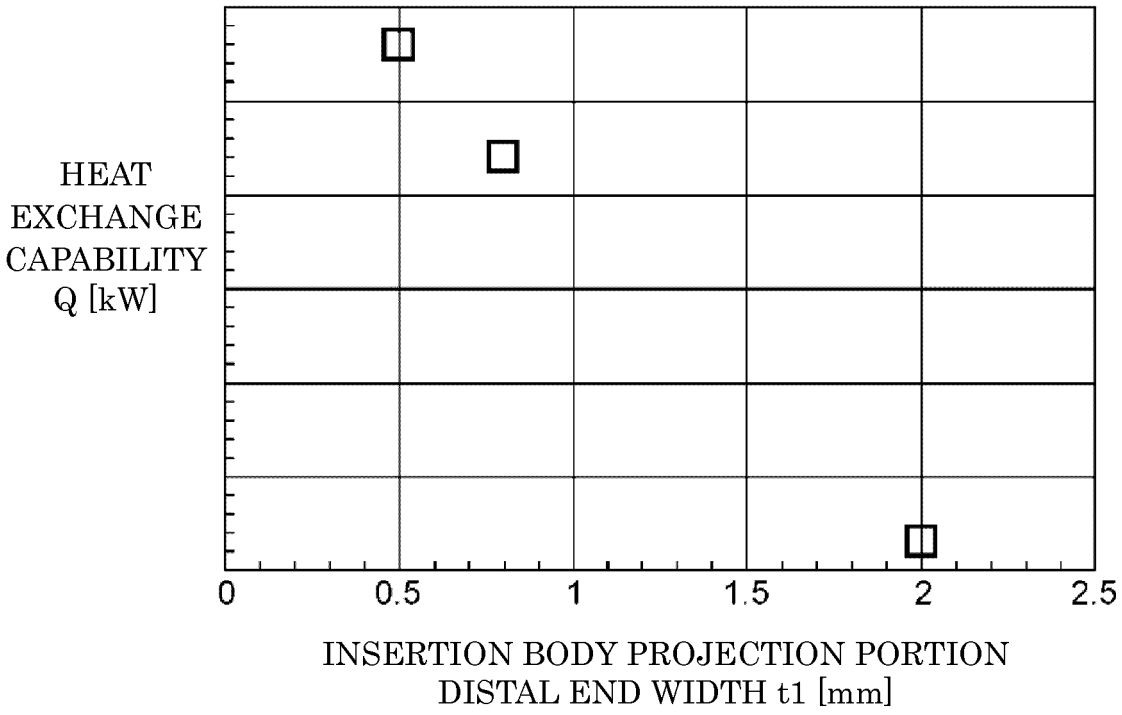

HEAT
EXCHANGE
CAPABILITY
Q [kW]
0
0.5
1
1.5
2
2.5
INSERTION BODY PROJECTION PORTION
DISTAL END WIDTH t1 [mm]

FIG. 10

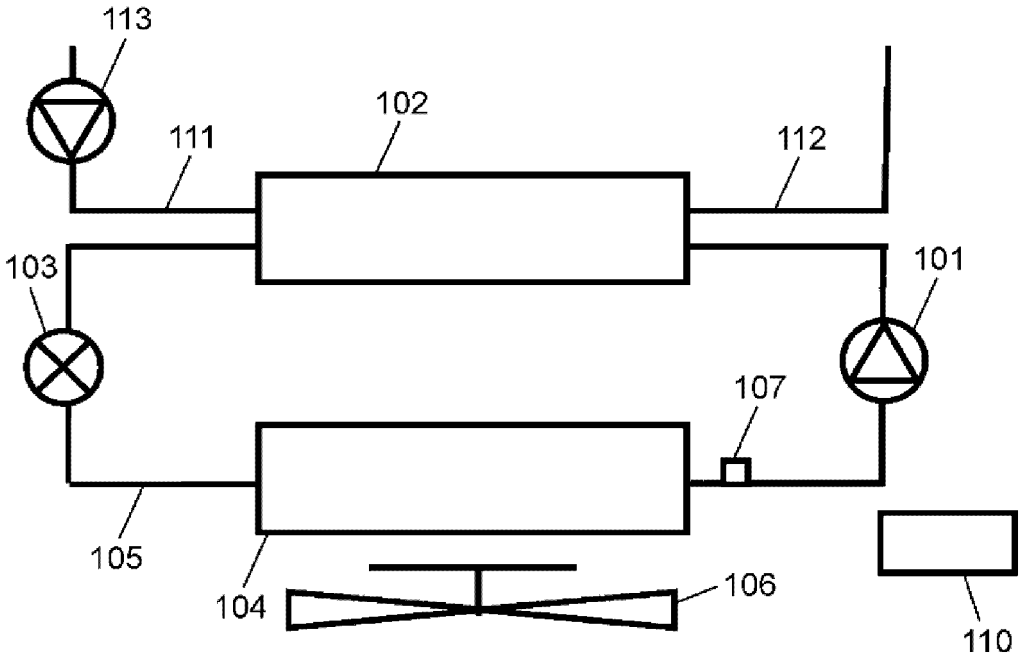


FIG. 11

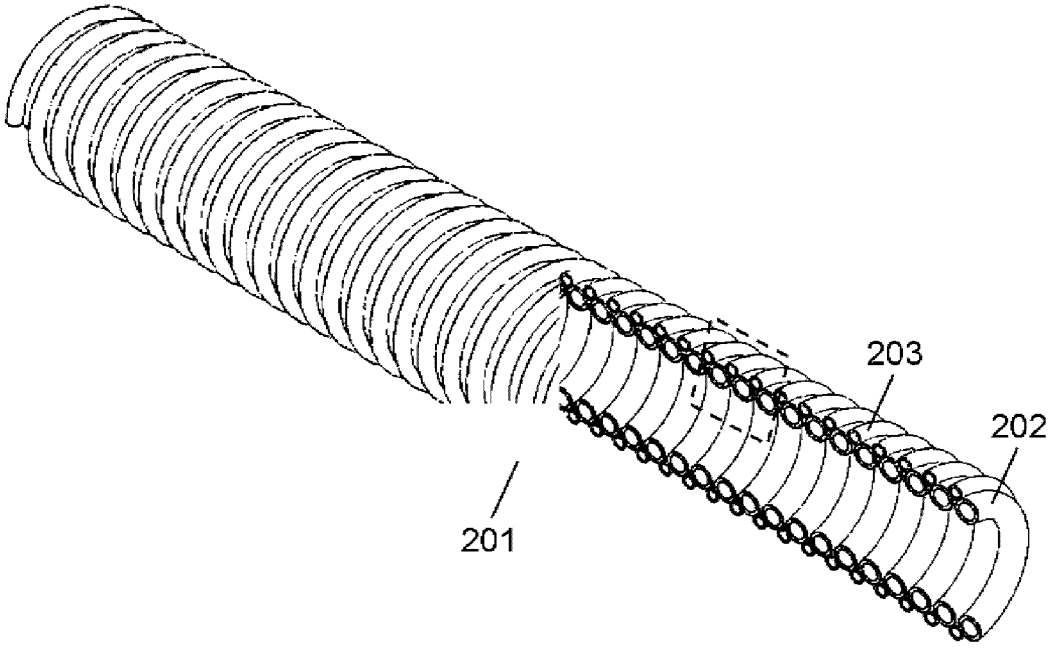


FIG. 12

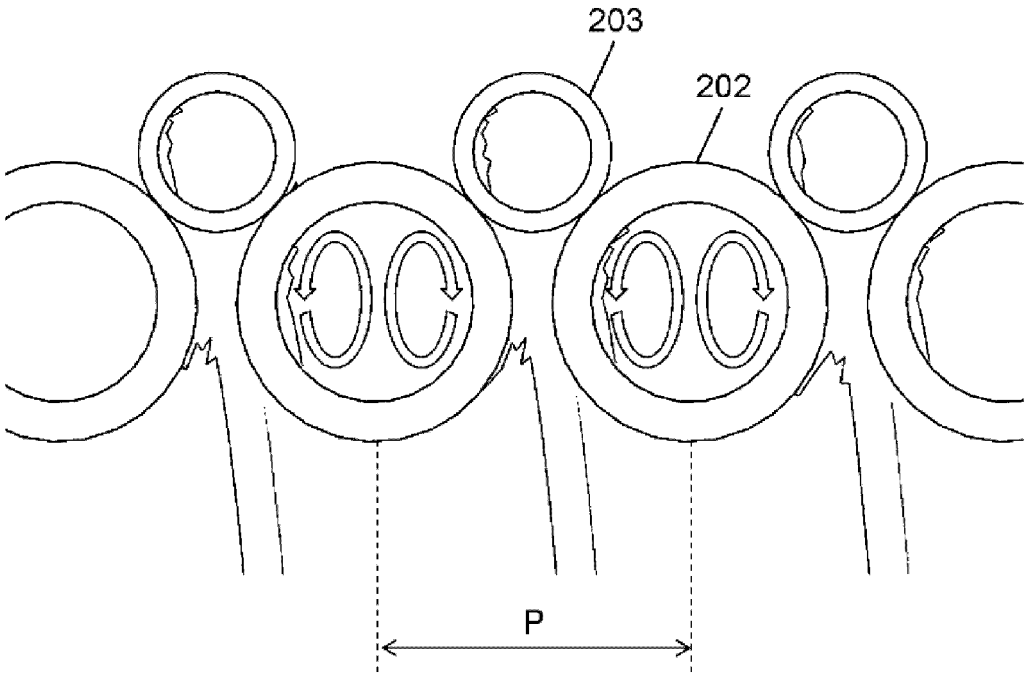
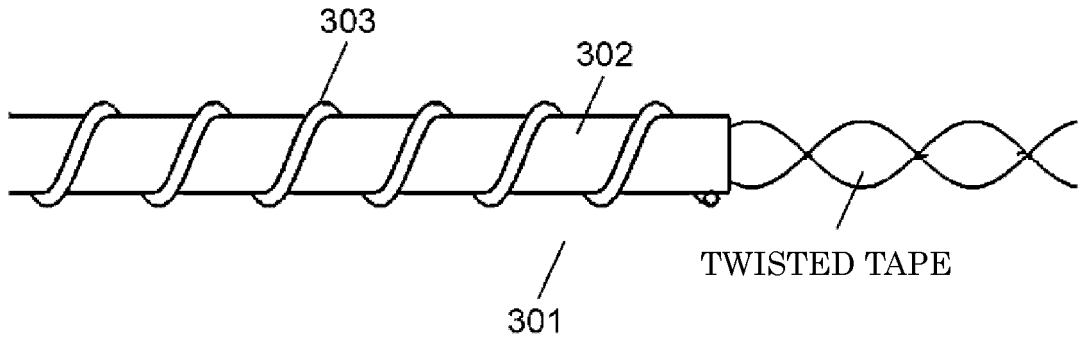


FIG. 13



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/001909

A. CLASSIFICATION OF SUBJECT MATTER
F28F13/12(2006.01)i, *F24H9/00*(2006.01)i, *F28D7/00*(2006.01)i, *F28D7/02*(2006.01)i, *F28F21/06*(2006.01)i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F28F13/12, F24H9/00, F28D7/00, F28D7/02, F28F21/06

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2016
Kokai Jitsuyo Shinan Koho 1971-2016 Toroku Jitsuyo Shinan Koho 1994-2016

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y A	JP 2010-127610 A (Atago Manufacturing Co., Ltd.), 10 June 2010 (10.06.2010), claims; paragraphs [0001] to [0015]; fig. 1 to 2 (Family: none)	1,3,6 2,4,7 5
Y	JP 2008-190777 A (Corona Corp., Hitachi Appliances, Inc.), 21 August 2008 (21.08.2008), paragraphs [0013] to [0018]; fig. 1 to 2 (Family: none)	2
Y	JP 2013-160479 A (Hitachi Appliances, Inc.), 19 August 2013 (19.08.2013), paragraph [0012]; fig. 1 to 3 (Family: none)	2

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

* Special categories of cited documents:
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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
17 June 2016 (17.06.16)	28 June 2016 (28.06.16)
Name and mailing address of the ISA/ Japan Patent Office 3-4-3,Kasumigaseki,Chiyoda-ku, Tokyo 100-8915,Japan	Authorized officer Telephone No.

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

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/001909

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2003-329376 A (Atago Manufacturing Co., Ltd.), 19 November 2003 (19.11.2003), paragraphs [0008] to [0011]; fig. 1 (Family: none)	4
Y	JP 2002-162175 A (Sunpot Co., Ltd.), 07 June 2002 (07.06.2002), paragraph [0009]; fig. 1 (Family: none)	4
Y	JP 2015-034664 A (Dainippon Printing Co., Ltd.), 19 February 2015 (19.02.2015), claim 4; paragraphs [0033] to [0035]; fig. 1 (Family: none)	7
Y	JP 2008-292107 A (The Furukawa Electric Co., Ltd.), 04 December 2008 (04.12.2008), claims; paragraph [0036]; fig. 1 (Family: none)	7
Y	JP 2010-091128 A (Daikin Industries, Ltd.), 22 April 2010 (22.04.2010), paragraphs [0016] to [0017], [0093]; fig. 1 & WO 2010/038568 A1	7
A	JP 2000-297991 A (HDE Metallwerk GmbH), 24 October 2000 (24.10.2000), entire text; all drawings & US 6220344 B1 & EP 1033547 A2	1-7
A	JP 2013-029303 A (Choi Gun Shik), 07 February 2013 (07.02.2013), entire text; all drawings & US 2013/0025834 A1 & EP 2551622 A2 & CN 102901382 A & RU 2012129344 A	1-7
A	WO 2006/103788 A1 (Tsinghua University, Daikin Industries, Ltd.), 05 October 2006 (05.10.2006), entire text; all drawings & US 2008/0149309 A1 & EP 1873471 A1 & CN 1740729 A & AU 2005329849 A & KR 10-2008-0004516 A	1-7

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 4805179 B **[0019]**
- JP 4501446 B **[0019]**

Non-patent literature cited in the description

- heat-transfer engineering document. Japan Society of Mechanical Engineers **[0054]**