(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 802 383 A2

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
22.10.1997 Bulletin 1997/43

(51) Int Cl.6: **F28D 1/053**, F25B 39/02

(21) Application number: **97302586.9**

(22) Date of filing: **16.04.1997**

(84) Designated Contracting States:
DE FR GB IT SE

(30) Priority: **18.04.1996 JP 96394/96**

(71) Applicant: **SANDEN CORPORATION**
Isesaki-shi Gunma, 372 (JP)

(72) Inventors:
- **Yamaguchi, Tomohiro**
  Isesaki-shi, Gunma 372 (JP)
- **Morita, Tomonari**
  Isesaki-shi, Gunma 372 (JP)
- **Sasaki, Kenichi**
  Isesaki-shi, Gunma 372 (JP)
- **Tsunoda, Masataka**
  Isesaki-shi, Gunma 372 (JP)

(74) Representative: **Jackson, Peter Arthur et al**
GILL JENNINGS & EVERY
Broadgate House
7 Eldon Street
London EC2M 7LH (GB)

(54) Multitubular heat exchanger having an appropriate tube arrangement pattern

(57) A combination of tank members (2, 3) forms a fluid path for guiding a first heat exchange medium substantially in a first direction. A plurality of microtubes (1) each having an outer diameter D are interposed between the tank members (2, 3). A second heat exchange medium is guided by these microtubes (1) in a second direction intersecting the first direction for heat exchange with the first heat exchange medium. The microtubes (1) are staggered to have a pitch X in the first direction and are aligned to have a pitch Y in a third direction perpendicular to the first and the second directions. In particular, the pitch Y is selected within a range specified by $1.8 \leqq Y/D \leqq 2.4$. It is preferable that the pitch X is selected within a range specified by $1.12 \leqq X/D \leqq 1.8$.

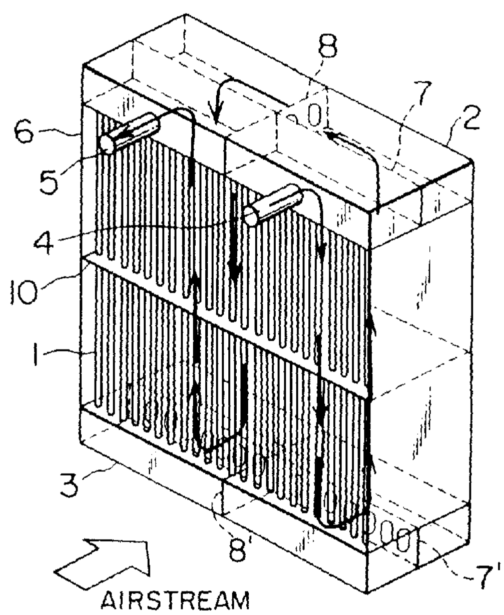

FIG. 1

EP 0 802 383 A2

Description

This invention relates to a heat exchanger for use in an air-conditioner and, in particular, to a multitubular or shell-and-tube heat exchanger comprising a tube bundle array.

A general heat exchanger for use in an air conditioner is required to have a low draft resistance (pressure loss) and a high heat transfer power (cooling capacity or heat radiation capacity). The heat transfer capacity (Q) is given by:

$$Q = K \times A \times dT,$$

where K represents an overall heat transfer coefficient, A, a heating surface area, and dT, a logarithmic mean temperature distance. If the heating surface area is constant, the heat transfer capacity is proportional to the overall heat transfer coefficient. It is noted here that the overall heat transfer coefficient is dependent upon the inside and the outside heat transfer coefficients inside and outside of the tubes. The outside heat transfer coefficient and the ventilating resistance are related to a tube arrangement pattern and an air velocity.

A multitubular heat exchanger of the type is disclosed in Japanese Unexamined Patent Publication (JP-A) No. 190287/1986 corresponding to United States Patent No. 4,676,305. The heat exchanger comprises a plurality of heat exchange modules each of which contains an array of a plurality of microtubes (may simply be referred to as tubes) arranged in parallel to one another and a shell surrounding the tubes. Each tube has an outside diameter not greater than 3mm. The heat exchange module is of a so-called counterflow type. Specifically, an outer fluid outside of the tubes flows within the shell in one direction parallel to an axial direction of each tube while an inner fluid inside of the tubes flows in a reverse direction reverse to the one direction. In a plane perpendicular to the axial direction, the tubes are arranged in a staggered pattern. Specifically, the tubes have a hexagonal-close-pack pattern with an angle of 60°. The row distance between rows of the tubes is equal to 0.866 times the tube center distance (TC) between tube centers. The tube center distance (TC) is generally equal to 1.3 to 2.8 times the outside diameter of the tubes. In such a multitubular heat exchanger also, it is important to increase a ratio of the overall heat transfer coefficient to the ventilating resistance.

In the multitubular heat exchanger, during passage of the air stream as the outer fluid, moisture contained in air is condensed at an outer surface of each tube. Such condensation of moisture has an influence upon heat transfer of the heat exchanger and can not be neglected.

However, the prior art shows no empirical formula to obtain the heat transfer coefficient taking the condensation of moisture into consideration, although proposal in the heat transfer engineering is made of a dimensionless formula capable of obtaining an outside heat transfer coefficient for an air stream flowing through a space in a tube bundle array containing a number of tubes arranged in a staggered pattern.

It is therefore an object of this invention to provide a multitubular heat exchanger which is capable of achieving an effective ratio of an outside heat transfer coefficient to a ventilating resistance by selecting an appropriate tube arrangement pattern taking condensation of moisture contained in air into consideration.

Other objects of this invention will become clear as the description proceeds.

A multitubular heat exchanger to which this invention is applicable is for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting the first direction.

According to an aspect of this invention, the heat exchanger comprises a plurality of microtubes each guiding the second heat exchange medium in the second direction. The microtubes are staggered to have a pitch X in the first direction and aligned to have a pitch Y in a third direction perpendicular to the first and the second directions. Each of the microtubes has an outer diameter not greater than 3mm. Each of the microtubes containes, as the second heat exchange medium, a refrigerant which is evaporated within each of the microtubes to make water be condensed as condensed water on an outer peripheral surface of each of the microtubes. The second direction is the vertical direction. The condensed water flows downward along the outer peripheral surface of each of the microtubes.

According to another aspect of this invention, the heat exchanger comprises a plurality of microtubes each having an outer diameter D and guiding the second heat exchange medium in the second direction. The microtubes are staggered to have a pitch X in the first direction and aligned to have a pitch Y in a third direction perpendicular to the first and the second directions. In the heat exchanger, the pitch X satisfies the relationship expressed by:

$$1.12 \leqq X/D \leqq 1.8.$$

According to still another aspect of this invention, the heat exchanger comprises a plurality of microtubes each having an outer diameter D and guiding the second heat exchange medium in the second direction. The microtubes are staggered to have a pitch X in the first direction and aligned to have a pitch Y in a third direction perpendicular to the first and the second directions. In the heat exchanger, the pitch Y satisfies the relationship expressed by:

$$1.8 \leqq Y/D \leqq 2.4.$$

In the accompanying drawings:-

Fig. 1 is a perspective view of a multitubular heat exchanger according to a first embodiment of this invention;
Fig. 2 is a perspective view of a multitubular heat exchanger according to a second embodiment of this invention;
Fig. 3 is a view for describing a staggered pattern of a tube bundle array of each of the multitubular heat exchangers illustrated in Figs. 1 and 2;
Fig. 4 is a view showing the state that sealing members are arranged upon measurement of characteristics of the tube bundle array in the multitubular heat exchanger illustrated in Fig. 1;
Fig. 5 shows a heat exchanger efficiency with respect to an X/D ratio;
Fig. 6 shows a heat exchanger efficiency with respect to a Y/D ratio;
Fig. 7 shows a heat exchanger efficiency with respect to the number of rows of tubes;
Fig. 8 is a perspective view of a multitubular heat exchanger according to a third embodiment of this invention;
Fig. 9 is a perspective view of a multitubular heat exchanger according to a fourth embodiment of this invention; and
Fig. 10 is a perspective view of a multitubular heat exchanger according to a fifth embodiment of this invention.

Now, description will be made about this invention with reference to the drawing.

Referring to Fig. 1, a multitubular heat exchanger according to a first embodiment of this invention is a multitubular evaporator. The evaporator comprises a large number of microtubes 1 each having an outer diameter not greater than 3.0mm. Each of the microtubes 1 has opposite ends bonded by brazing to upper and lower tanks 2 and 3 so as to allow fluid communication between the microtubes 1 and the upper and the lower tanks 2 and 3. In the following description, the microtubes 1 may collectively be called a tube bundle.

The upper tank 2 is connected to an inlet tube 4 and an outlet tube 5. The upper tank 2 is provided with partition plates 7 and 8 partitioning an internal space of the upper tank 2, as depicted by dotted lines in the figure. Likewise, the lower tank 3 is provided with similar partition plates 7' and 8'. Each of the upper and the lower tanks 2 and 3 is thus divided into four chambers. As seen from the figure, the partition plate 8 in the upper tank 2 is partially provided with communication holes to allow fluid communication between the rear-side two chambers. On the other hand, the partition plate 7' in the lower tank 3 is provided with communication holes to allow fluid communication between the right-side two chambers and between the left-side two chambers.

In the example being illustrated, a pair of side plates 6 are attached to both lateral sides of the evaporator. Furthermore, a horizontal center plate 10 is fixed to the side plates 6 to support the tube bundle.

As an inner fluid, a refrigerant flows through the inlet tube 4 into the upper tank 2, travels through a refrigerant path formed by the microtubes 1 and the upper and the lower tanks 2 and 3, and flows out from the upper tank 2 through the outlet tube 5, as depicted at thick solid lines with arrowheads. Specifically, the refrigerant path comprises four parts, namely, a front-side downward path, a rear-side upward path, a rear-side downward path, and a front-side upward path. Herein, the partition plates in the upper and the lower tanks 2 and 3 serve as an inner fluid guiding arrangement to define the above-mentioned four-part refrigerant path.

As an outer fluid or a first heat exchange medium, air flows into the evaporator from a front side thereof in a first direction depicted by a white arrow in the figure, namely, in the horizontal direction. The first heat exchange medium flows through a space in the tube bundle in the first direction which may be referred to as the airstream direction. Herein, a combination of the upper and the lower tanks 2 and 3, the side plates 6, and the center plate 10 serves as a guiding arrangement for guiding the first heat exchange medium in the first direction.

Between the upper and the lower tanks 2 and 3, each microtube extends in a second direction intersecting or perpendicular to the first direction, namely, in the vertical direction. The refrigerant flows upward or downward as a second heat exchange medium within the microtubes 1 for heat exchange with the first heat exchange medium. Thus, the illustrated multitubular evaporator is a multitubular heat exchanger of an orthogonal flow type.

Referring to Fig. 2, a multitubular heat exchanger according to a second embodiment of this invention is also a multitubular evaporator. The evaporator comprises a large number of microtubes 1 each having an outer diameter not greater than 3.0mm. Each of the microtubes 1 has opposite ends bonded by brazing to upper and lower tanks 11 and 12 so as to allow fluid communication between the microtubes 1 and the upper and the lower tanks 11 and 12.

The upper tank 11 is connected to an inlet tube 16 and an outlet tube 17. The upper tank 11 is provided with partition plates 13, 14, and 15 partitioning an internal space of the upper tank 11, as depicted by dotted lines in the figure. Likewise, the lower tank 12 is provided with similar partition plates 13', 14', and 15'. Each of the upper and the lower tanks 11 and 12 is thus divided into six chambers. As seen from the figure, the partition plate 15 in the upper tank 11 is provided with communication holes to allow fluid communication between the rear-side center and left chambers and between the front-side left and center chambers. Likewise, the partition plates 13' and 14' in the lower tank 12 are provided with communication holes to allow fluid communication between the rear-side right and center chambers, between the left-side two chambers, between the front-side center and right chambers, and between the rear-side center and right chambers.

Like in the first embodiment, a pair of side plates 6 are attached to both lateral sides of the evaporator. Furthermore, a horizontal center plate 10 is fixed to the side plates 6 to support the tube bundle.

As an inner fluid or a second heat exchange medium, a refrigerant flows through the inlet tube 16 into the upper tank 11, travels through a refrigerant path formed by the microtubes 1 and the upper and the lower tanks 11 and 12, and flows out from the upper tank 11 through the outlet tube 17, as depicted at thick solid lines with arrowheads. Specifically, the refrigerant path comprises six parts, namely, a rear-side right downward path, a rear-side center upward path, a rear-side left downward path, a front-side left upward path, a front-side center downward path, and a front-side right upward path. Herein, the partition plates in the upper and the lower tanks 11 and 12 serve as an inner fluid guiding arrangement to define the above-mentioned six-part refrigerant path.

As an outer fluid or a first heat exchange medium, air flows in a first direction depicted by a white arrow in the figure in the manner similar to that described in conjunction with the first embodiment. In the second embodiment also, a combination of the upper and the lower tanks 11 and 12, the side plates 6, and the center plate 10 serves as a guiding arrangement for guiding the first heat exchange medium in the first direction.

In each of the multitubular evaporators shown in Figs. 1 and 2, the microtubes 1 of the tube bundle are arranged in a staggered pattern, which will hereafter be described, so as to obtain an optimum ratio of an overall heat transfer coefficient to a ventilating resistance. In order to obtain the pattern exhibiting the above-mentioned optimum ratio, measurement has been made of the overall heat transfer coefficient and the ventilating resistance of the tube bundle for various arrangement pitches.

Referring to Fig. 3, description will be made about an arrangement of the microtubes 1 in each of the multitubular evaporators shown in Figs. 1 and 2. In Fig. 3, the first direction along which the outer fluid or the first heat exchange medium flows is depicted by a white arrow. The second direction along which the inner fluid or the second heat exchange medium flows is perpendicular to a drawing sheet. A third direction is perpendicular to the first and the second directions. Each of the microtubes 1 has an outer diameter D. The microtubes 1 are staggered or offset in each of the first and the second directions. In other words, the microtubes 1 are staggered to have a pitch X in the first direction while they are aligned to have a pitch Y in the third direction. For convenience of description, the microtubes 1 are classified into three groups, namely, first microtubes la, second microtubes 1b adjacent to the first microtubes la in the third direction, and third microtubes 1c adjacent to the first and the second microtubes 1a and 1b in a fourth direction obliquely intersecting the first and the third directions.

Experimentally, a plurality of samples of heat exchangers were manufactured with the structure illustrated in Fig. 1 or 2. These samples had different arrangement patterns with different X/D and Y/D ratios of the pitches X and Y to the outer diameter D of the microtubes. As a test apparatus, use was made of a cyclometric calorimeter including a refrigerant circuit using fluorocarbon as a refrigerant. Each sample was used as an evaporator in the refrigerant circuit.

The test conditions were as follows:

Air Temperature at Evaporator Inlet: 27°C
Relative Humidity: 50%
Refrigerant Pressure
before Expansion Valve: 1.74 MPa
Refrigerant Pressure
at Evaporator Outlet: 0.28 MPa
Subcooling and Superheating: 5 deg (Celsius)

An air volume (outlet air volume) passing through the evaporator was controlled to several predetermined air volumes between 300 and 450 m^3/h. Then, measurement was made of a cooling capacity and a ventilating resistance of the evaporator. An air velocity in each sample for these air volumes is about 6 to 7m/s or less in terms of a peak velocity, namely, a velocity during passage through a minimum spacing or gap between adjacent tubes. A Reynolds number Re_{max} is not greater than 1200 for the peak velocity and the tube diameter of a characteristic size.

For convenience of the test apparatus, the cooling capacity Q is calculated as a product of the weight of an outlet air passing through the evaporator and an enthalpy difference between the inlet air and the outlet air of the evaporator, as expressed by the following equation:

$$Q = Ga \times (Ia1 - Ia2),$$

where Ga represents the weight of flow of the outlet air, Ia1, the enthalpy of the inlet air, and Ia2, the enthalpy of the outlet air.

At an outer surface of each tube, heat transfer is accompanied by condensation of moisture contained in air passing through the evaporator. Therefore, a heat transfer characteristic of the tube bundle is represented as the overall heat transfer coefficient based on enthalpy difference.

The overall heat transfer coefficient based on enthalpy difference is represented by the following equation:

$$Q = K_i \times A \times dI.$$

The above equation is rewritten into:

$$K_i = Q/(A \times dI),$$

where K_i represents the overall heat transfer coefficient based on enthalpy difference, A, the outside total heating surface area, and dI, the logarithmic mean enthalpy difference. The logarithmic mean enthalpy difference dI is given by:

$$dI = [(Ia1 - Ia2)]$$

$$/\ln[Ia1 - Ir) - (Ia2 - Ir)],$$

where Ir represents the enthalpy of saturated air corresponding to the refrigerant saturation temperature at the mean evaporator pressure.

The pressure loss f is defined by the use of the ventilating resistance dP of the evaporator and the number N of rows of tubes in the first direction as follows:

$$f = dP/N.$$

For evaluation, the heat exchanger efficiency of the evaporator is defined as a ratio (K_i/f) of the overall heat transfer coefficient based on enthalpy difference to the pressure loss.

Fig. 4 schematically shows the arrangement of the tube bundle in the multitubular evaporators shown in Fig. 1. Gaps are present between lateral sides of the tube bundle and the side plates 6. In addition, another gap is present at the center of the tube bundle in correspondence to the partition plate within the tank described above. These gaps may possibly form bypassing paths of the airstream. In order to more accurately measure the ventilating resistance and the overall heat transfer coefficient of the tube bundle, such unexpected bypassing of the airstream in the tube bundle must be inhibited. For this purpose, sealing members 21, 22, and 23 are fitted in these gaps, as illustrated in Fig. 4.

Fig. 5 shows the K_i/f ratio as the heat exchanger efficiency with respect to the X/D ratio. In the figure, seven kinds of 3mm-diameter samples have the constant Y/D ratio of 2 and the different X/D ratios. The minimum X/D ratio is equal to 1.04, in which case the portion b in Fig. 3 gives a minimum path sectional area corresponding to the above-mentioned minimum spacing or gap. The second minimum X/D ratio is equal to 1.12, in which case the portions a and b have the same size. In the other 3-mm diameter samples, the minimum path sectional area is given by the portion a. For convenience of data evaluation, the relationship between the K_i/f ratio and the air velocity is shown in Fig. 5 for all of the samples. Herein, the air velocity is calculated from a predetermined air volume and the path sectional area based on the portion a. The number of rows of tubes in the first direction (airstream direction) is equal to 12 in the 3-mm diameter samples.

The four kinds of 2.2mm-diameter samples have the constant Y/D ratio of 1.82 and the different X/D ratios. The number of rows of tubes is equal to 11.

In the 3-mm diameter samples for the velocity of each of 6.0m/s, 5.0m/s, and 4.0m/s, the K_i/f ratio slightly increases with an increase of the X/D ratio until it is saturated at the X/D ratio of about 1.6. On the other hand, with a decrease of the X/D ratio, the K_i/f value at first decreases at the X/D ratio of 1.12 in which case the portions a and b have the same size, and further decreases at the smaller X/D ratio of 1.04. This is particularly because the ventilating resistance by the portion b giving the minimum path sectional area is predominant and great. On the other hand, the condensed water forms a water film on the surface of each tube. If the portion b is narrow, the water film further decreases the path sectional area. With an increase of the ventilating resistance, the problem of frost arises also. Therefore, although the heat exchanger is made compact by reducing the X/D ratio, the X/D ratio must be selected to be an optimum value such that the above-mentioned disadvantages are avoided.

In the 2.2mm-diameter samples, the K_i/f ratio shows the gradient substantially equal to that of the 3mm-diameter samples. It is noted here that, with the decrease in air velocity, the saturation tendency is slightly suppressed in the 2.2mm-diameter samples as compared with the 3mm-diameter samples.

Taking the above into consideration together, the X/D ratio must be equal to 1.12 or more, preferably, 1.2 or more in order to obtain the effective value of the heat exchange efficiency, namely, the ratio (K_i/f) of the overall heat transfer coefficient based on enthalpy difference to the pressure loss. On the other hand, the increase in X/D ratio results in a disadvantage that the tube bundle array is greater in size in the first or the airstream direction. Accordingly, taking the saturation of the K_i/f ratio into consideration, the X/D ratio must have an upper limit value on the order of 1.8, preferably, 1.7.

Fig. 6 shows the K_i/f ratio as the heat exchanger efficiency with respect to the Y/D ratio. In the figure, the six kinds of 3mm-diameter samples have the constant X/D ratio of 1.33 and the different Y/D ratios. The three kinds of 2.2mm-diameter samples have the constant X/D value of 1.27 and the different Y/D ratios.

In the 3mm-diameter samples, the K_i/f ratio tends to increase for each air velocity until the Y/D ratio reaches 2.2. Beyond the value, the K_i/f ratio is saturated.

On the other hand, in the 2.2mm-diameter samples, the K_i/f ratio exhibits the gradient substantially equal to that of the 3mm-diameter samples. It is noted here that the K_i/f ratios in the 2.2mm-diameter samples are slightly lower than those in the 3mm-diameter samples. The reason will presently be described in conjunction with a specific example. As described in the foregoing, the 3mm-diameter samples and the 2.2mm-diameter samples have the X/D ratios of 1.33 and 1.27, respectively, in Fig. 6. The difference in K_i/f ratio resulting from the above-mentioned difference in X/D ratio is about 3% as seen from the gradient at the air velocity of 4m/s in Fig. 6 for example. It will be understood that the K_i/f ratios of the 2.2mm-diameter samples in Fig. 6 are lower than those of the 3.0mm-diameter samples in correspondence to the above-mentioned difference in X/D ratio. In Fig. 6, the 2.2mm-diameter sample having the Y/D value of 2.0 shows the K_i/f ratio of about 88. Taking the above-mentioned 3% difference into consideration, the K_i/f ratio of about 88 is modified into about 90.6 which is nearer to the K_i/f value of the corresponding 3.0mm-diameter sample.

It will be understood from Fig. 6 that, in order to obtain the effective value of the heat exchanger efficiency, namely, the ratio (K_i/f) of the overall heat transfer coefficient based on enthalpy difference to the pressure loss, the Y/D ratio must be equal to about 2.2. On the other hand, the increase in Y/D ratio requires a reduced number of tubes to be arranged in the third or widthwise direction if the width of the heat exchanger is unchanged. This results in decrease in cooling capacity. In order to assure a sufficient cooling capacity, the number of the tubes must not be reduced. In this event, the increase in Y/D ratio results in an increase in size of the heat exchanger in the third direction and is therefore unfavorable. Taking the above and the practical aspect into consideration together, the Y/D ratio within a range between about 1.8 and 2.4 is preferable to provide an effective K_i/f ratio.

Fig. 7 shows the K_i/f ratio as the heat exchanger efficiency with respect to the number N of rows of tubes. In the figure, all samples have the tube diameter of 3mm and the X/D ratio of 1.33. The numbers of rows of tubes are equal to 10, 12, and 14. For each air velocity, the K_i/f ratio slightly increases in a linear fashion with respect to the increase of the number of rows of tubes.

In Figs. 5 and 6, description has been made about the heat exchanger efficiency for the tube bundles comprising 11 and 12 rows of the microtubes 1. It will be understood that the similar characteristic is obtained in case where the number of rows is between 10 and 14 and is even smaller or greater than the above-mentioned range.

In each of the foregoing embodiments, the evaporator is divided into four or six sections. It will be understood that the number of sections may be any appropriate number as far as the inner circuit pressure loss caused by the refrigerant flowing inside of the tubes can be suppressed within an allowable range and the refrigerant is substantially uniformly distributed in the respective tubes. At any rate, the refrigerant or the inner fluid flows into one of the tanks and flows out from the other tank.

Referring to Fig. 8, a multitubular heat exchanger according to a third embodiment of this invention has upper and lower tanks 11 and 12 each of which has no partition plate. The inner fluid travels through a single path from the lower tank 12 through the microtubes 1 to the upper tank 11.

Referring to Fig. 9, a multitubular heat exchanger according to a fourth embodiment of this invention has upper and lower tanks 11 and 12 each of which has two chambers divided by a single partition plate. The inner fluid travels through a two-part fluid path from the upper tank 11 through the microtubes 1, the lower tank 12, and the microtubes 1 to the upper tank 11.

In the foregoing, the heat exchanger has a pair of tanks arranged in parallel to each other in a vertical direction. Alternatively, the tanks may be arranged in parallel to each other in the horizontal direction with the axial direction of the tubes in parallel to the horizontal direction. In this case, however, additional means is required to assure substantially uniform flow of the inner heat exchange medium within the respective tubes.

Referring to Fig. 10, a multitubular heat exchanger according to a fifth embodiment of this invention has a single-sided tank structure. In each of the foregoing embodiment, the upper and the lower tanks (2, 3 or 11, 12) are arranged opposite to each other with the tube bundle interposed therebetween. In this embodiment, the heat exchanger comprises upper and lower tanks 11 and 12 arranged at one side (at the top in Fig. 10) of a tube bundle array. The upper and the lower tanks 11 and 12 are formed by partitioning a single tank at a center plane in the vertical direction to obtain a two-layer structure. Each microtube 1 is bent at the bottom of the heat exchanger to form a U shape and has one tube end communicating with the lower tank 12 and the other tube end communicating with the upper tank 11 after penetrating through the lower tank 12. With this structure, an inner heat exchange medium flows from the lower tank 12 through the tubes 1 into the upper tank 11.

This invention is preferably applicable to a tube bundle array including ten or more rows of tubes with the tube diameter of 3mm or more in case where an air stream has the Raynolds number Re_{max} far smaller than 2000 for the peak velocity and the tube diameter of a characteristic size.

As described above, the multitubular heat exchanger according to this invention is capable of achieving the effective value of the heat exchanger efficiency as the ratio of the outside heat transfer coefficient to the ventilating resistance by selecting an appropriate tube arrangement pattern taking condensation of moisture contained in air into consideration.

While the present invention has thus far been described in connection with a few embodiments thereof, it will readily be possible for those skilled in the art to put this invention into practice in various other manner. For example, the present invention may be embodied to have the following aspects.

1. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting the first direction, the heat exchanger comprising:

a first microtube;
a second microtube located adjacent to the first microtube in a third direction perpendicular to the first and the second directions; and
a third microtube located adjacent to the first and the second microtubes in a fourth direction obliquely intersecting the first and the third directions, a sum of a spacing between the first and the third microtubes and a spacing between the second and the third microtubes is approximated to but is not smaller than a spacing between the first and the second microtubes so that a heat exchanger efficiency is improved without increasing a pressure loss of the first heat exchange medium.

2. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting the first direction, the heat exchanger comprising:

a first microtubes;
a second microtubes located adjacent to the first microtube in a third direction perpendicular to the first and the second directions; and
a third microtube located adjacent to the first and the second microtubes in a fourth direction obliquely intersecting the first and the third directions;
the first, the second, and the third microtubes bing arranged so that a resistance against the first heat exchange medium during passage through a spacing between the first and the second microtubes is approximate to a sum of a resistance during passage through a spacing between the first and the third microtubes and a resistance during passage through a spacing between the second and the third microtubes.

3. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting the first direction, the heat exchanger comprising:

a first microtube;
a second microtube located adjacent to the first microtubes in a third direction perpendicular to the first and the second directions; and
a third microtubes located adjacent to the first and the second microtubes in a fourth direction obliquely intersecting the first and the third directions, characterized in that:
each of mutual spacings between outer surfaces of the first and the third microtubes and between outer surfaces of the second and the third microtubes is not smaller than a half of a mutual spacing between outer surfaces of the first and the second microtubes.

Claims

1. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting said first direction, said heat exchanger comprising a plurality of microtubes each guiding said second heat exchange medium in said second direction, said microtubes being staggered to have a pitch X in said first direction and aligned to have a pitch Y in a third direction perpendicular to said first and said second directions, each of said microtubes having an outer diameter not greater than 3mm, each of said microtubes containing, as said second heat exchange medium, a refrigerant which is evaporated within each of said microtubes to make water be condensed as condensed water on an outer peripheral surface of each of said microtubes, said second direction being the vertical direction, said condensed water flowing downward along said outer peripheral surface of each of the microtubes.

2. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting said first direction, said heat exchanger comprising a plurality of microtubes each having an outer diameter D and guiding said second heat exchange medium in said second direction, said microtubes being staggered to have a pitch X in said first direction and aligned to have a pitch Y in a third direction perpendicular to said first and said second directions, said pitch X satisfying the relationship expressed by:

$$1.12 \leqq X/D \leqq 1.8.$$

3. A multitubular heat exchanger as claimed in claim

2, wherein said outer diameter D is not greater than 3mm.

4. A multitubular heat exchanger as claimed in claim 2, wherein each of said microtubes contains, as said second heat exchange medium, a refrigerant which is evaporated within each of said microtubes to makes water be condensed as condensed water on an outer peripheral surface of each of said microtubes.

5. A multitubular heat exchanger as claimed in claim 4, wherein said second direction is the vertical direction, said condensed water flowing downward along said outer peripheral surface of each of the microtubes.

6. A multitubular heat exchanger for carrying out heat exchange between a first heat exchange medium flowing substantially in a first direction and a second heat exchange medium flowing in a second direction intersecting said first direction, said heat exchanger comprising a plurality of microtubes each having an outer diameter D and guiding said second heat exchange medium in said second direction, said microtubes being staggered to have a pitch X in said first direction and aligned to have a pitch Y in a third direction perpendicular to said first and said second directions, the pitch Y satisfying the relationship expressed by:

$$1.8 \leqq Y/D \leqq 2.4.$$

7. A multitubular heat exchanger as claimed in claim 6, wherein said outer diameter D is not greater than 3mm.

8. A multitubular heat exchanger as claimed in claim 6, wherein each of said microtubes contains, as said second heat exchange medium, a refrigerant which is evaporated within each of said microtubes to makes water be condensed as condensed water on an outer peripheral surface of each of said microtubes.

9. A multitubular heat exchanger as claimed in claim 8, wherein said second direction is the vertical direction, said condensed water flowing downward along said outer peripheral surface of each of the microtubes.

10. A multitubular heat exchanger as claimed in claim 6, whrerein said pitch X satisfying the relationship expressed by:

$$1.12 \leqq X/D \leqq 1.8.$$

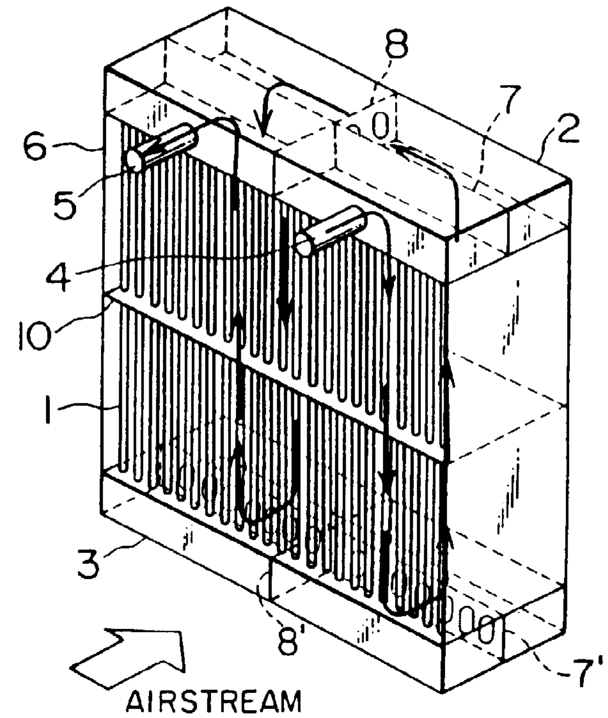
8
7
2
6
5
4
10
1
3
8'
7'
AIRSTREAM

FIG. 1

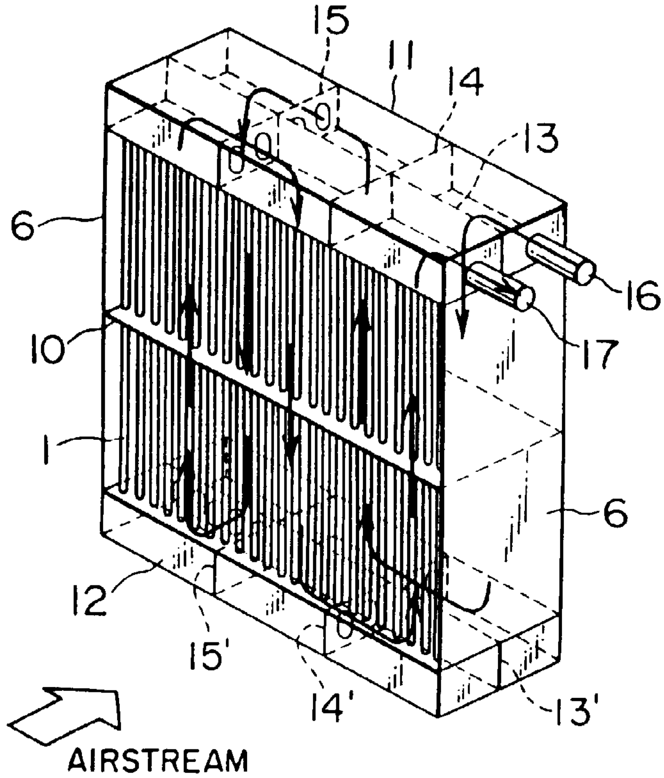
15
11
14
13
6
16
17
10
1
6
12
15'
14'
13'
AIRSTREAM

FIG. 2

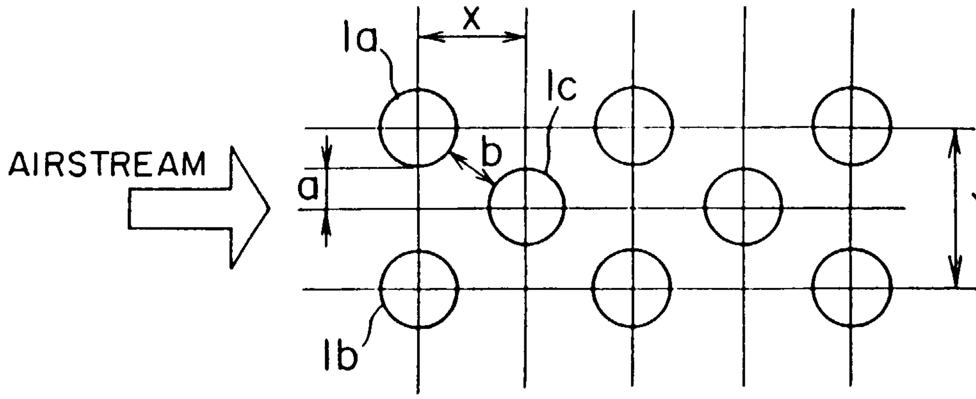

FIG. 3

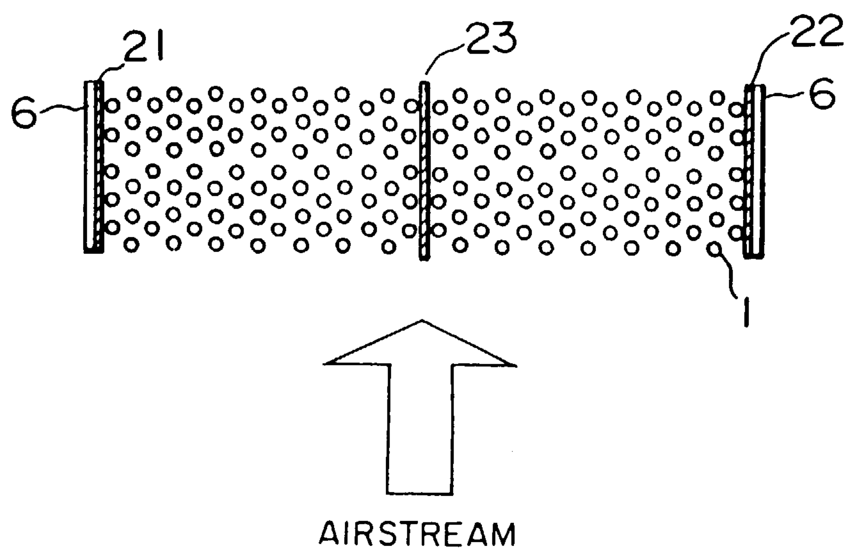

FIG. 4

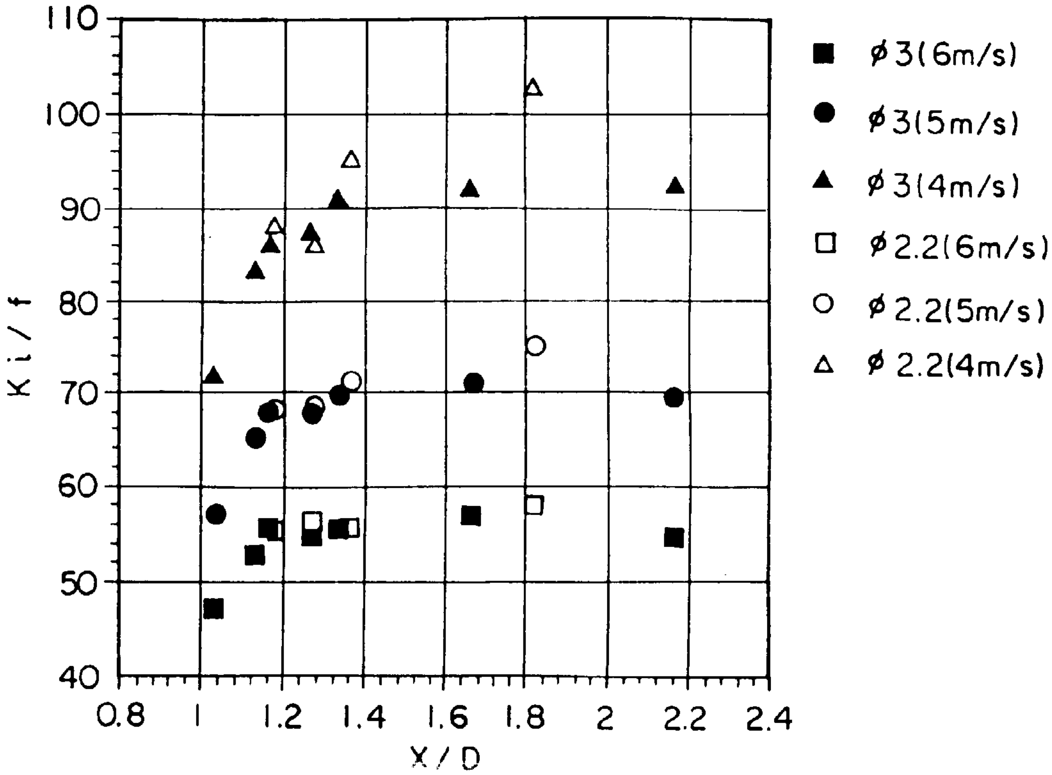
110
100
90
80
70
60
50
40
Ki/f
0.8
1
1.2
1.4
1.6
1.8
2
2.2
2.4
X/D
■ φ3(6m/s)
● φ3(5m/s)
▲ φ3(4m/s)
□ φ2.2(6m/s)
○ φ2.2(5m/s)
△ φ2.2(4m/s)

FIG. 5

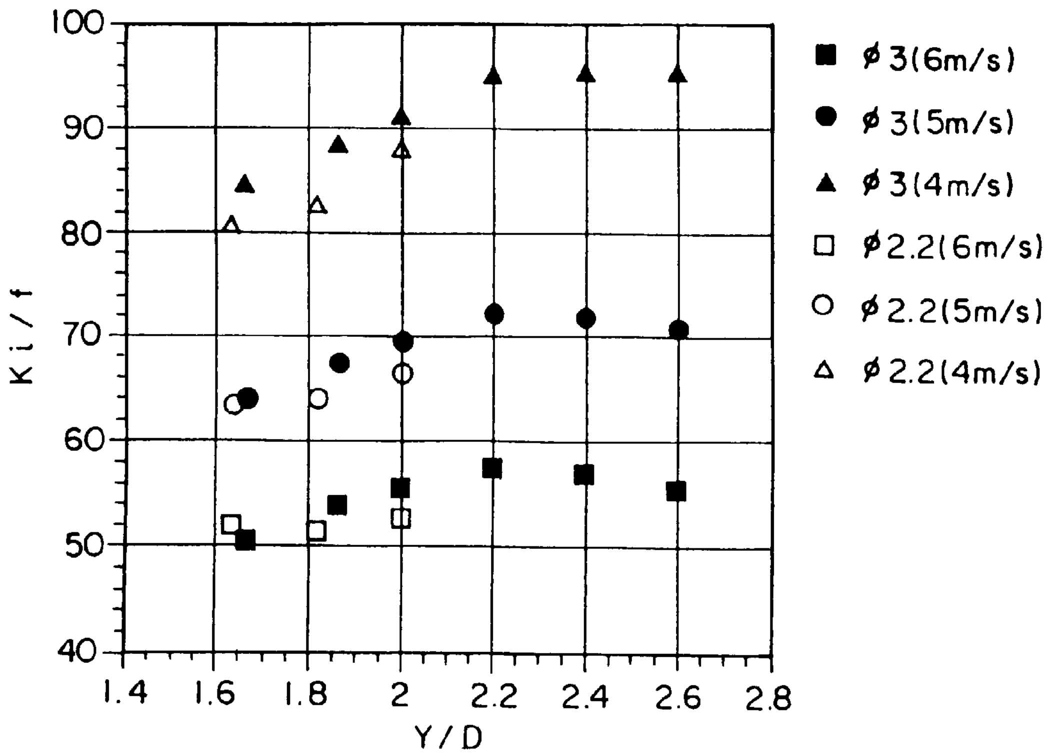
100
90
80
70
60
50
40
Ki/f
1.4
1.6
1.8
2
2.2
2.4
2.6
2.8
Y/D
■ φ3(6m/s)
● φ3(5m/s)
▲ φ3(4m/s)
□ φ2.2(6m/s)
○ φ2.2(5m/s)
△ φ2.2(4m/s)

FIG. 6

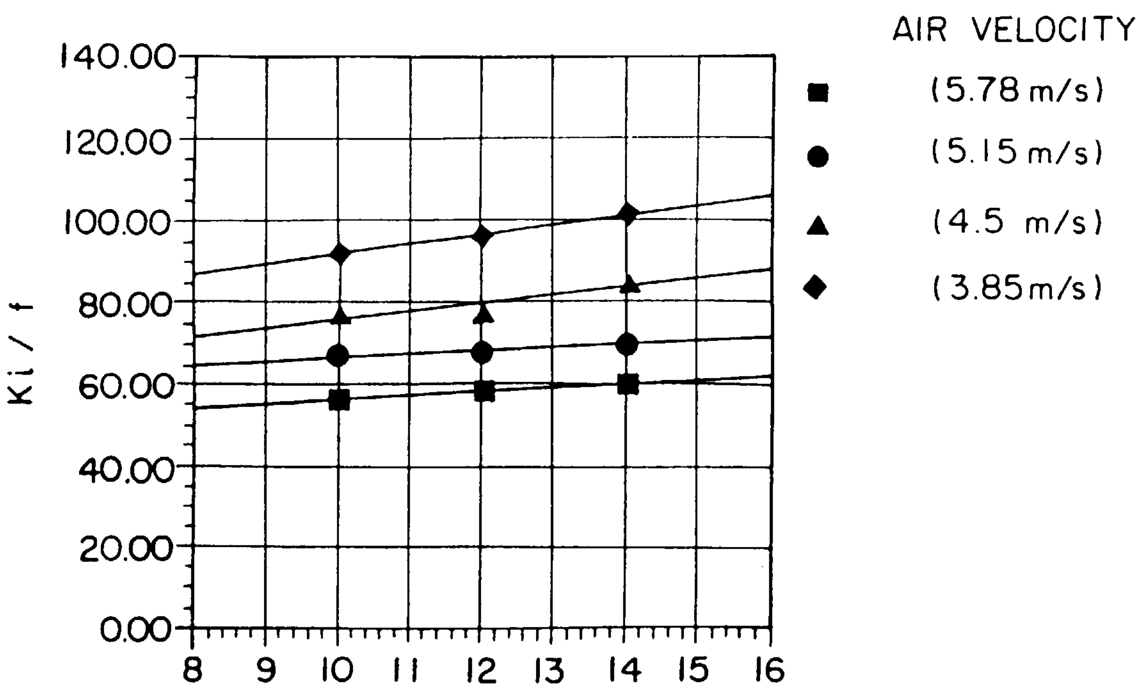
AIR VELOCITY
(5.78 m/s)
(5.15 m/s)
(4.5 m/s)
(3.85 m/s)
140.00
120.00
100.00
80.00
60.00
40.00
20.00
0.00
Ki / f
8
9
10
11
12
13
14
15
16

FIG. 7

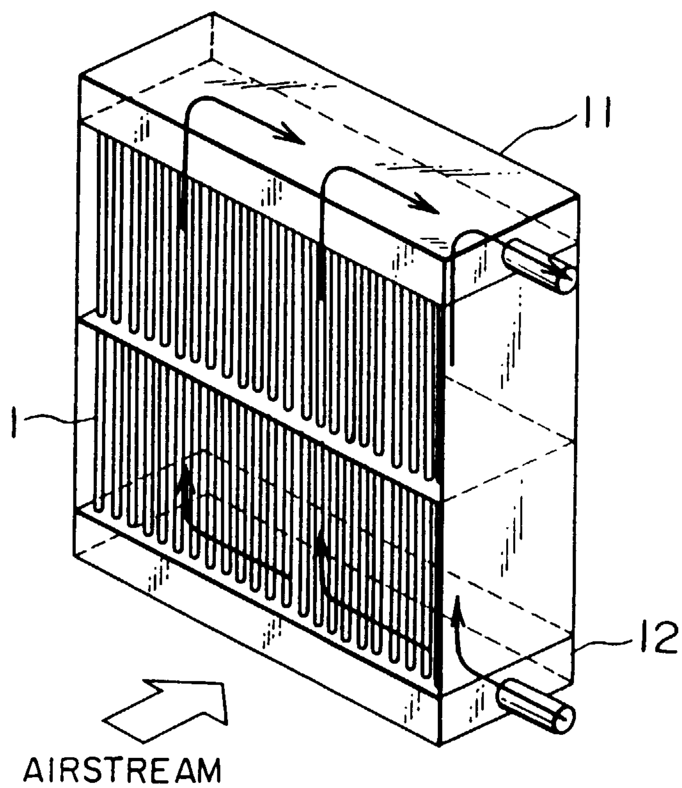
11
1
12
AIRSTREAM

FIG. 8

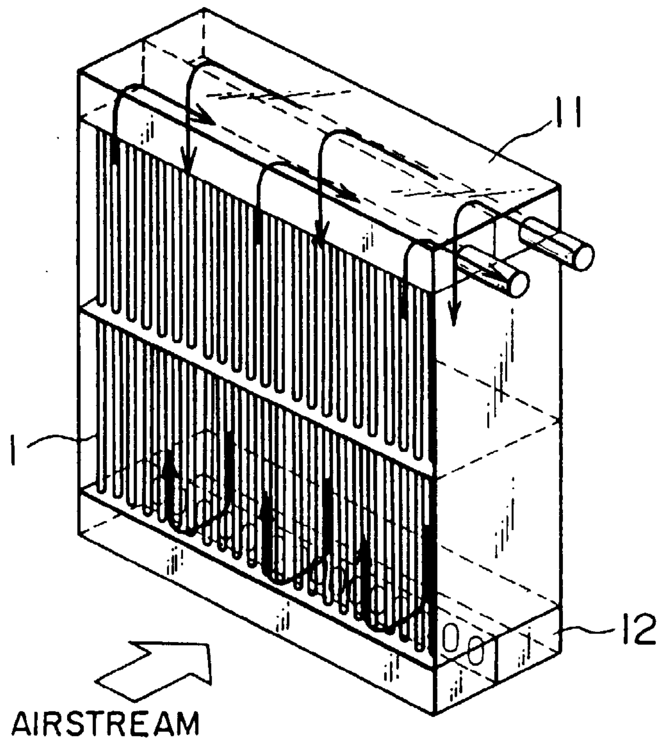

FIG. 9

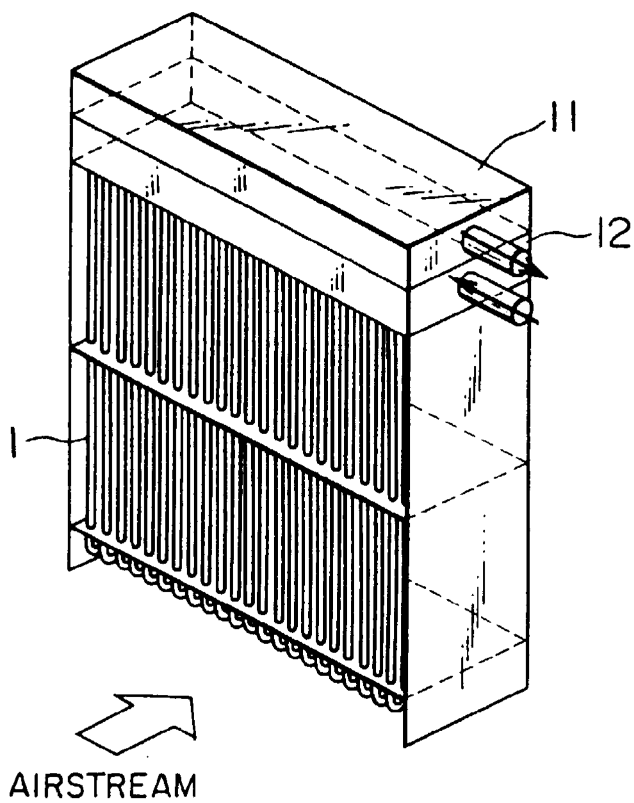

FIG. 10