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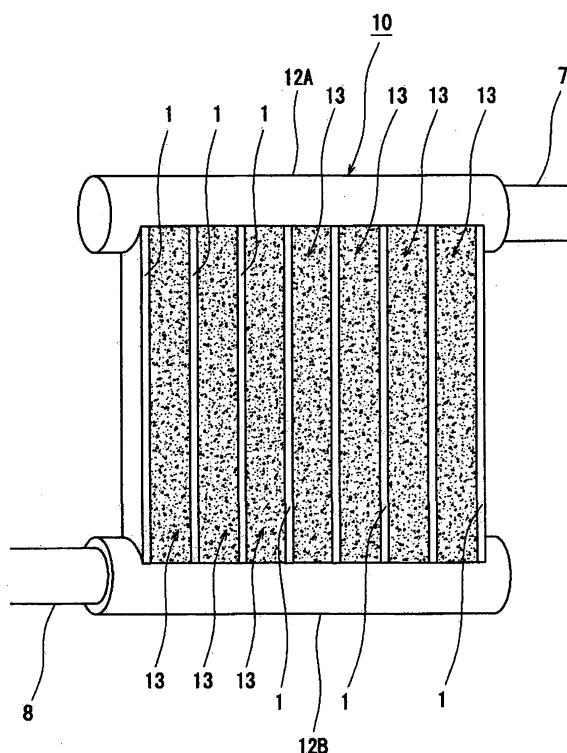
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(54) **HEAT TRANSFER FIN FOR HEAT EXCHANGER**

(57) A heat transfer fin (13) is provided on a heat transfer tube (1) wherein a fluid for exchanging heat with

air flows. The heat transfer fin which transfers heat by having contact with air is composed of a metal foam having a pore density of 20 PIP or more.

Fig.1



Description

TECHNICAL FIELD

5 **[0001]** The present invention relates to a heat transfer fin appropriate for use in a heat exchanger for air conditioners and other types of heat exchangers.

BACKGROUND ART

10 **[0002]** Improvement in the heat transfer performance of an air side heat transfer fin for a heat exchanger or the like used in an air conditioner is an essential factor when miniaturizing the heat exchanger for saving energy in the system itself.

[0003] To improve the heat transfer performance, a cross fin type heat exchanger including slit fins or louver fins has been proposed. Japanese Laid-Open Patent Publication No. 4-93595 and Japanese Laid-Open Patent Publication No. 9-26279 disclose such heat exchangers. When slits or louvers are arranged for heat transfer fins made of thin plates
15 having satisfactory heat transmittance such as aluminum plates, their front edges function to improve the heat transfer performance (heat transfer coefficient) with air.

[0004] Japanese Laid-Open Patent Publication No. 2002-195774 proposes the use of a stacked type heat exchanger including flat heat transfer tubes and corrugated fins in an air conditioner. The entire heat exchanger including the flat heat transfer tubes and the corrugated fins as well as each part of the heat exchanger is shown in Figs. 31 and 32.

20 **[0005]** A heat exchanger 10 includes pipe-shaped upper and lower headers 12A and 12B through which a refrigerant is flows in and out. A plurality of parallel flat heat transfer tubes 1, which extend between the headers 12A and 12B in a direction orthogonal to the headers 12A and 12A, are formed in communication with the headers 12A and 12B. Corrugated fins 11, which are formed by continuously bending a flat aluminum plate or the like, is arranged between the heat transfer tubes 1 to join the adjacent heat transfer tubes 1.

25 **[0006]** As shown in Fig. 32, the heat transfer tube 1 includes a plurality of refrigerant passages 2 having a square cross-section partitioned by partitions. The refrigerant flowing through the headers 12A and 12B from external refrigerant pipes 7 and 8 flows uniformly into each refrigerant passage 2 so that efficient heat exchange is immediately performed between the inside refrigerant and the outside air by the wide heat transfer area of the flat surface of the heat transfer tubes 1 and the corrugated fin 11.

30 **[0007]** A plurality of louvers 11a and 11b for increasing the heat transfer efficiency with air are formed in the corrugated fins 11 along a plane from the upstream side to the downstream side and configured so that the front edges function to immediately improve the heat exchange performance between the refrigerant and the air.

DISCLOSURE OF THE INVENTION

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PROBLEMS THAT THE INVENTION IS TO SOLVE

[0008] However, improvement in the heat transfer performance has limits when just forming slits and louvers in heat transfer fins made of thin aluminum plates regardless of the form of the heat transfer fin used. Selecting the material for
40 the heat transfer fin and improving the heat transfer performance of the heat exchanger are thus being considered.

[0009] It is an object of the present invention to provide heat transfer fins for a heat exchanger that significantly improves the heat transfer performance by forming air side heat transfer fins for a heat exchanger from a foam metal manufactured by foaming copper, aluminum, or the like, which have high heat transmittance.

45 MEANS FOR SOLVING THE PROBLEM

[0010] The present invention is configured as below to achieve the above object.

(1) First Solution

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[0011] A first solution according to the present invention is a heat transfer fin arranged on a heat transfer tube through which fluid flows for exchanging heat with air. The heat transfer fin contacts air to exchange heat. The heat transfer fin is made of foam metal having a pore density of 20 PIP or greater.

[0012] The foam metal has an open cell type porous structure with fine linear grooves continuously connected to one
55 another enabling fluid to flow therethrough. Thus, the surface area per unit volume is large. Therefore, the heat transfer area of the foam metal is large. The heat transfer is enhanced by disturbance in the fluid since the foam metal has a complex passage.

[0013] Furthermore, since the foam metal includes linear grooves, a temperature boundary layer can be easily renewed,

and a high heat transfer coefficient can be obtained. The heat transfer performance of the heat transfer fin thus becomes extremely high.

[0014] Therefore, the heat exchange performance of a heat exchanger is greatly improved when employing the heat transfer fin made of foam metal.

[0015] However, the passage configuration is complex in the foam metal and the pressure loss is large. Thus, an optimal pore density must be determined when employing foam metal as the material for the heat transfer fin. According to the results of various analyses and experiments, the more preferable pore density for maximizing the heat transfer property of the foam metal is 20 PIP or greater.

(2) Second Solution

[0016] In a second solution according to the present invention, the heat transfer tube through which fluid flows for exchanging heat with air is provided as a plurality, and the plurality of heat transfer tubes are set at an interval of 12 mm or less.

[0017] As described above, the heat transfer fin made of foam metal has a superior heat transfer performance since the surface area per unit volume is large and the heat transfer area is large. However, the fin efficiency is low compared to a louver fin or the like because the foam metal has fine linear grooves. Therefore, the interval of the plurality of heat transfer tubes must be optimized. According to the results of various analyses, the interval of the plurality of heat transfer tubes is effective when it is 12 mm or less, and the heat transfer performance is sufficiently improved especially when the pore density is 20 PIP or greater.

(3) Third Solution

[0018] In a third solution according to the present invention, a heat exchanger is a stacked type heat exchanger.

[0019] The stacked type heat exchanger is configured so that the heat transfer tube is flat and extends in the air flowing direction, and so that the heat transfer fin arranged in between is sufficiently long in the air flowing direction. Therefore, the stacked type heat exchanger itself has high heat transfer performance.

[0020] The heat transfer performance further improves when employing the heat transfer fin made of foam metal in the stacked air-heat exchanger.

[0021] According to the above structure, a heat exchanger having a high heat exchange capability and appropriate for use in an air conditioner is formed with low cost, and miniaturization of the heat exchanger is achieved.

(4) Fourth Solution

[0022] In a fourth solution according to the present invention, the pore density is 20 PIP or greater and 60 PIP or less.

[0023] In addition to the first solution, the optimal upper limit value of the bore density is obtained through various analyses and experiments.

(5) Fifth Solution

[0024] In a fifth solution according to the present invention, an interval H between the plurality of heat transfer tubes is set at 4 mm or greater and 12 mm or less.

[0025] As described above, the foam metal has superior heat transfer performance since the surface area per unit volume is large and the heat transfer area is large. However, the fin efficiency is low. Therefore, the interval for the plurality of heat transfer tubes must be optimized. From the results of various experiments, the heat transfer performance improved when the interval of the heat transfer tubes was 4 mm or greater and 12 mm or less.

BRIEF DESCRIPTION OF THE DRAWINGS

[0026]

Fig. 1 is a perspective view showing the configuration of a heat exchanger according to a first embodiment of the present invention;

Fig. 2 is an enlarged view showing the configuration of the material structure of a heat transfer fin made of foam metal;

Fig. 3 is an enlarged view showing samples having different structure densities for the material structure of the heat transfer fin made of foam metal in which (A) shows a first (No. 1) sample, (B) shows a second (No. 2) sample, and (C) shows a third (No. 3) sample (pore ratios of 10 PPI, 20 PPI, an 40 PPI);

Fig. 4 is a graph showing the relationship between the front surface side wind velocity V_f and the heat transfer

performance QN per unit volume in the heat transfer fin;

Fig. 5 is a graph showing the relationship between the heat transfer performance per unit volume and the pump power in the heat transfer fin;

Fig. 6 is a graph showing the relationship of the pressure loss ΔP with respect to the front surface side wind velocity V_f in the heat transfer fin;

Fig. 7 is a graph showing the pressure loss property when warming the foam metal in the heat transfer fin;

Fig. 8 is a graph showing changes in a friction loss coefficient f when warming the heat transfer fin;

Fig. 9 is a graph showing the relationship between the heat transfer coefficient and the surface area β per unit volume when warming the heat transfer fin;

Fig. 10 is a graph showing the relationship between h_o and the front surface side wind velocity V_f when warming the heat transfer fin;

Fig. 11 is a graph showing the relationship between the calculation result obtained from equation (10) in the description of the operation of the heat transfer fin and the experiment result;

Fig. 12 is a graph showing the relationship between the pressure loss ($\Delta P/D$) with respect to the front surface side wind velocity (V_f) when dry cooling the heat transfer fin;

Fig. 13 is a graph showing the relationship between the pressure loss ($\Delta P/D$) with respect to the front surface side wind velocity (V_f) when wet cooling (water temperature 5°C) the heat transfer fin;

Fig. 14 is a graph showing the relationship between the pressure loss ($\Delta P/D$) with respect to the front surface side wind velocity (V_f) when wet cooling (water temperature 10°C) the heat transfer fin;

Fig. 15 is a graph showing the ratio $\Delta P_{wet}/\Delta P_{dry}$ of the pressure loss ΔP_{wet} during wet cooling (water temperature 5°C) with respect to the pressure loss ΔP_{dry} during dry cooling (5°C) in the heat transfer fin;

Fig. 16 is a graph showing the ratio $\Delta P_{wet}/\Delta P_{dry}$ of the pressure loss ΔP_{wet} during wet cooling (water temperature 10°C) with respect to the pressure loss ΔP_{dry} during dry cooling (15°C) in the heat transfer fin;

Fig. 17 is a graph showing the relationship between the heat transfer coefficient h_{dry} and the front surface side wind velocity V_f when dry cooling the heat transfer fin;

Fig. 18 is a graph showing the relationship between the heat transfer performance $h_{dry}\beta$ per unit volume and the front surface wind velocity (V_f) when dry cooling the heat transfer fin;

Fig. 19 is a graph showing the heat transfer performance $h_{dry}\beta$ per unit volume when dry cooling the heat transfer fin in comparison with the louver fin;

Fig. 20 is a graph showing the heat transfer coefficient h_{wet} when wet cooling (water temperature 5°C) the heat transfer fin;

Fig. 21 is a graph showing the heat transfer coefficient h_{wet} when wet cooling (water temperature 10°C) the heat transfer fin;

Fig. 22 is a graph showing the heat transfer performance h_{wet} per unit volume when wet cooling (water temperature 5°C) the heat transfer fin;

Fig. 23 is a graph showing the heat transfer performance h_{wet} per unit volume when wet cooling (water temperature 10°C) heat transfer fin;

Fig. 24 is a graph showing the relationship between the substance transfer coefficient h_{mass} and the front surface side wind velocity V_f when wet cooling (water temperature 5°C) the heat transfer fin;

Fig. 25 is a graph showing the relationship between the substance transfer coefficient h_{mass} and the front surface side wind velocity V_f when wet cooling (water temperature 10°C) the heat transfer fin;

Fig. 26 is a graph showing the relationship between the heat transfer performance $h_{dry}\beta$ per unit volume and the necessary power $E\beta$ per unit volume when dry cooling the heat transfer fin;

Fig. 27 is a graph showing the relationship between the heat transfer performance $h_{dry}\beta$ per unit volume and the necessary power $E\beta$ per unit volume when wet cooling the heat transfer fin;

Fig. 28 is a graph showing the relationship between the mass transfer performance $h_{mass}\beta$ per unit volume and the necessary power $E\beta$ per unit volume when cooling the heat transfer fin;

Fig. 29 is a perspective view showing the structure configuration of a heat exchanger according to a second embodiment of the present invention;

Fig. 30 is a perspective view showing a configuration of a heat exchanger according to a third embodiment of the present invention;

Fig. 31 is a perspective view showing a configuration of an air-heat exchanger in the prior art; and

Fig. 32 is a partially cut-away perspective view showing the configuration of the main part of the conventional air-heat exchanger.

BEST MODE FOR CARRYING OUT THE INVENTION

(First embodiment)

[0027] Figs. 1 and 2 show the configuration for an entire heat exchanger and the main part thereof according to preferred embodiment 1 of the present invention. As shown in Fig. 1, the heat exchanger 10 is configured so that a plurality of parallel flat heat transfer tubes 1 extend between pipe shaped upper and lower headers 12A and 12B through which refrigerant flows in and out. The heat transfer tubes 1 are connected to the headers 12A and 12B and extend in a direction orthogonal to the headers 12A and 12B. Corrugated fins 11, which are formed by continuously bending a flat aluminum plate or the like, is joined to the adjacent heat transfer tube 1 between the heat transfer tubes 1.

[0028] In this embodiment, the heat transfer fin 13 is not the corrugated fin of the prior art and is made of foam metal of open cell type having a porous structure, as shown in Fig. 2, that allows fluid to flow therethrough.

[0029] The heat transfer tube 1 has a plurality of refrigerant passage grooves with square cross-sections partitioned by partitions in the same manner as the prior art shown in Fig. 32. Refrigerant, which is drawn into and distributed by the external refrigerant piping 7 or 8, flows uniformly through each refrigerant passage groove from the upper side toward the lower side or from the lower side toward the upper side via the upper header 12A or the lower header 12B. Heat exchange is immediately and efficiently performed with the ambient air through the heat transfer surface of the refrigerant flowing through each refrigerant passage and the fin surface with the porous structure in the heat transfer fin 13 made of foam metal.

[0030] The foam metal forming the heat transfer fin 13 is a porous substance. Therefore, the foam metal has a large heat transfer area since the surface area per unit volume is large and complex passages are formed therein. Thus, effective heat transfer promotion may be expected due to the disturbance of the fluid. Since foam metal has many fine linear grooves connected to each other (see structure Fig. 2), a temperature boundary layer may easily be renewed, and an extremely high heat transfer coefficient can be obtained.

[0031] When the heat exchanger 10 having such a configuration is used, for example, as a condenser, the refrigerant introduced from the external refrigerant piping 7 via the upper header 12A is uniformly distributed to flow from the upper side to the lower side of the heat transfer tube 1, and discharged from the outer refrigerant pipe 8 through the lower header 12B. If the heat exchanger 10 is used as an evaporator, the refrigerant flows in the opposite direction.

[0032] In such a stacked type heat exchanger, the heat transfer tubes 1 are flat and elongated in the air flow direction, and the heat transfer fins 13 arranged in between also extend in the air flowing direction. Thus, the heat transfer performance is large. The heat transfer fins 13 are easily formed by foaming and molding a metal having a high heat transfer coefficient, such as aluminum or copper, into a shape that can be brazed.

[0033] Therefore, the heat exchanger suitable for use in an air conditioner may be formed with a reduced size at a lower cost and with a high heat exchange performance.

[0034] The foam metal forming the heat transfer fin 13 of the present embodiment is a porous substance, in which the surface area per unit area expands as the pore density PPI becomes higher as in (A) 10 PPI, (B) 20 PPI and (C) 40 PPI shown in Figs. 3(A), (B) and (C). However, the pressure loss increases. Therefore, the determination of an optimal range for the pore density PPI is important when using foam metal for the heat transfer fin 13 of the air-heat exchanger described above. The PPI (pore per inch) represents the density of gas bubbles per cubic inch.

[0035] Through the results of various analyses and experiments, the pore density PPI of the foam metal was found to be generally preferable at 20 PPI (Figs. 3(B), (C)) or greater and 60 PPI or less.

[0036] A problem in which the fin efficiency is low compared to the louver fin of the prior art arises for the heat transfer fin 13 made of foam metal since the diameter of the linear grooves connected to one another is small. Therefore, the interval (fin width) H of the heat transfer tube 1 must have an optimal value. According to the results of various analyses and experiments, the interval H of the heat transfer tube 1 was found to be optimal in the range of 4 mm or greater and 12 mm or less.

[0037] That is, according to the experiment results, the interval H of the heat transfer tube 1 is effective when it is 4 mm or greater and 12 mm or less, as described below. In particular, the heat transfer performance QN (W/m^3) per unit volume with respect to the front surface wind velocity V_f greatly improves compared to that for the louver fin, as shown in the graph of Fig. 4, if the interval H of the heat transfer tube 1 is 4 mm or greater and 12 mm or less when the pore density of the heat transfer fin 13 made of foam metal is greater than or equal to 20 PPI and less than or equal to 60 PPI. Furthermore, when the interval H of the heat transfer tube 1 and the pore density of the heat transfer fin are as described above, the heat transfer amount per unit volume at the same power increases by about 25% compared to that for the louver fin, as shown in the graph of Fig. 5. This obtains the improvement effect for the effective heat transfer performance.

[0038] As will be described below, the dimensions 5 mm, 8 mm, 12 mm in the examples of Figs. 4 and 5 are three sets of dimensional data used to analyze the interval (hereinafter considered as width of heat transfer fin 13) H of the heat transfer tube 1.

(Test Example)

[0039] Several experiments were performed to check the improvement effect for the effective heat transfer performance of the heat transfer fin 13 made of foam metal.

1. First Test Example (Check Heating Performance)

[0040] A foam metal made of aluminum (aluminum alloy 6010) was used for the heat transfer fin as the open cell type tested foam metal. Three types of the foam metal having a pore density PPI of, for example, No. 1, 10 PPI shown in Fig. 3(A); No. 2, 20 PPI shown in Fig. 3(B); and No. 3, 40 PPI shown in Fig. 3(C) were prepared. Furthermore, three heat transfer tubes having different width dimensions (i.e., interval of heat transfer tube 1) H of 5 mm, 8 mm, and 12 mm were prepared for each of the three types of bore density PPI. For the total of nine types of samples, heat exchange was performed between the refrigerant (warm water as one example) at the heat transfer tube side 1 in the configuration of Fig. 1 and the air flowing outside.

[0041] The pressure loss and the heat transfer coefficient of the heat transfer fin for each case were experimentally obtained, and analysis was performed to clarify the basic heat transfer property and the influence of the wall surface of the heat transfer tube 1 serving as the heat source.

[0042] The experiments were conducted under the conditions of 20°C for air temperature and 50% for relative humidity. The measurement of the pressure loss was performed in a non-load condition in which no warm water was supplied to the heat transfer tube 1, and the measurement of the heat transfer coefficient was performed by supplying warm water of 50°C as a warm heat source. The wind velocity range was 0.5 to 2.3 m/s in terms of the wind velocity V_f at the front surface side (upstream side) of the heat transfer fin 13.

[0043] The specific material of the aluminum foam metal used in this experiment was aluminum alloy 6101, as mentioned above. The detailed specification is shown in [Table 1]. In order to check the influence of the interval H between the wall surfaces for the foam metal No. 1 to No. 3 (10, 20, 40 PPI) of the three types of pore densities described above, fins of the three widths H of 5 mm, 8 mm, and 12 mm were prepared. Thus, a total of nine types of samples were prepared. The height L of the foam metal was 89 mm, the depth D was 13 mm, and the surface area per unit volume was β .

[Table 1]

Sample	Pore Ratio PPI	Porosity	Surface Area β (m ² /m ³)	Pore Size d_{pore} (mm)	Width H (mm)		
No. 1	10 PPI	0.914	791	2.5	5	8	12
No. 2	20 PPI	0.915	1744	1.25	5	8	12
No. 3	40 PPI	0.908	2740	0.65	5	8	12

1) Pressure Loss

[0044] The graph of Fig. 6 shows the relationship of the pressure loss P(Pa) with respect to the front surface wind velocity V_f (m/s). The pressure loss ΔP increases as the pore density PIP increases, or as the pore size d_{pore} decreases. Further, the pressure loss ΔP increases as the fin width H decreases. This is because the surface area (including wall surface) per unit volume increases as the pore size d_{pore} decreases and the fin width H decreases. The foam metal has a higher pressure loss ΔP than a louver fin (width H=7.9 mm, D=13.6 mm, fin pitch=1.5 mm) of a comparative example.

[0045] The pressure loss property of the foam metal is expressed as follows using the permeability (K) and the Ergun coefficient (C_E).

$$\frac{\Delta P}{D} = \frac{\mu}{K} V_f + \frac{C_E}{\sqrt{K}} \rho V_f^2 \quad \text{----- (1)}$$

[0046] The graph of Fig. 7 is obtained, for example, when K and C_E obtained through the least-square method using equation (1) are shown in relation with the pore size d_{pore} . K increases as the pore size d_{pore} increase, and the fin width H increases as the friction caused by the wall surface decreases. C_E slightly decreases as the pore size d_{pore} increases. However, a definite tendency cannot be found in the influence of the fin width H.

[0047] The friction loss coefficient f and Re_K are defined as below using the obtained K .

$$f = \left(-\frac{dP}{dx} \cdot \frac{\sqrt{K}}{\rho V_f^2} \right) \quad \text{--- (2)}$$

$$Re_K = \rho V_f \sqrt{K} / \mu \quad \text{--- (3)}$$

[0048] Fig. 8 shows change in the friction loss coefficient f with respect to Re_K .

2) Heat Transfer Coefficient

[0049] The graph of Fig. 9 shows the product of the measured heat transfer coefficient h_a and the surface area β per unit area. First, the heat transfer coefficient h_a is defined by the following equation.

$$h_a = Q_a \cdot A_t / \Delta T_{LMTD} \quad \text{--- (4)}$$

[0050] Q_a is the heat transfer amount, A_t is the total heat transfer area combining the surface area of the foam metal and the area of the wall surface, and ΔT_{LMTD} is the logarithmic mean temperature difference. Further, $h_a\beta$ represents the heat transfer performance per unit volume. It is apparent here that the heat transfer performance increases as the pore density PPI increases and the smaller the fin width H decreases. In particular, the heat transfer performance becomes higher than the conventional louver fin in the 20 PIP No. 2 fin sample and the 40 PIP No. 3 fin sample. This suggesting the possibility of sufficient miniaturization of the heat exchanger.

[0051] Although the above heat transfer coefficient h_a includes the fin efficiency, the heat transfer coefficient h_o should not include the fin efficiency for optimal designing. However, the heat transfer coefficient h_o cannot be easily obtained due to the complex passage shape resulting from foam metal. Accordingly, h_o is obtained through the following approximation method.

[0052] The fin efficiency of flat plates spaced from each other by dimension H is expressed with the following equation.

$$\eta = \frac{\tanh(mH/2)}{mH/2} \quad \text{--- (5)}$$

$$m = \sqrt{h_o P / kA} \quad \text{--- (6)}$$

[0053] P is the perimeter, A is the cross-sectional area, and k is the heat conductivity. Since P/A of equation (6) is unknown for the foam metal, the value of m is assumed as in equation (7), h_o is assumed as being constant due to dimension H , and the values of C are obtained for each pore density PPI (10 PPI, 20 PPI, 40 PPI) of No. 1 to No. 3 from experimental data in which the dimension H is changed. The results are shown in [Table 2].

$$m = \sqrt{C \cdot h_o / k} \quad \text{-----} \quad (7)$$

[Table 2]

	10 PPI	20 PPI	40 PPI
C	30900	68000	94800

[0054] From the above results, it is apparent that C increases as the pore density PPI increases. This is because the line diameter of line material increases as the pore density PPI increases and thereby increases P/A. The result indicating h_o with respect to the front surface wind velocity V_f is the graph of Fig. 10. As apparent from the graph, h_o increases as the pore density PPI decreases. However, this effect is barely seen when the pore density PIP 20 PIP or less. Further, d_{pore} and $\sqrt{K}$ are considered as the characteristic lengths of the porous material. However, since K changes in accordance with the value of the interval H between the wall surfaces when influenced by the wall surface, as shown in Fig. 7, the value of K at H=12 mm (K_{12}) in which the influence of the wall surface is minimum is used for the characteristic length, and the non-dimensional variable is defined as in the following equation.

$$Nu_{K_{12}} = h_o \sqrt{K_{12}} / k \quad \text{-----} \quad (8)$$

$$Re_{K_{12}} = \rho V_f \sqrt{K_{12}} / \mu \quad \text{-----} \quad (9)$$

[0055] By organizing data based on the equations, the following correlation equation is obtained.

$$Nu_{K_{12}} = 0.116 Re_{K_{12}}^{0.5423} \left(\sqrt{K_{12}} / d_{pore} \right)^{-0.45} Pr^{1/3} \quad \text{-----} \quad (10)$$

[0056] The contrast of the correlation equation of equation (10) and the actual experimental results are shown in Fig. 11. As a result, it is apparent that they frequently match, and 90% of the data is within an error of $\pm 6\%$.

2. Second Test Example (Check Cooling Performance)

[0057] In this case, in the same manner as in the first test example described above, aluminum alloy 6010 was used as the foam metal forming the heat transfer fin. Three types of the foam metal having a pore density PIP of No. 1, 10 PIP; No. 2, 20 PIP; and No. 3, 40 PIP were prepared. Furthermore, three heat transfer tubes having different width dimensions (i.e., interval of heat transfer tube 1) H of 5 mm, 8 mm, and 12 mm were prepared for each of the three types of bore density PIP. For the total of nine types of samples, heat exchange was performed between the refrigerant (cold water as one example) at the heat transfer tube side 1 in the configuration of Fig. 1 and the air flowing outside.

[0058] The pressure loss and the heat transfer coefficient in this case were experimentally obtained and analyzed to clarify the basic heat transfer property and the influence of the wall surface of the heat transfer tube 1 serving as the heat source.

[0059] The experiments were conducted under the conditions of 20°C for air temperature and 50% for relative humidity. The measurement of the pressure loss was performed in a non-load condition in which no cold water was supplied to the heat transfer tube 1, and the measurement of the heat transfer coefficient was performed by supplying cold water of 50°C as a cold heat source. The wind velocity range was 0.5 to 2.3 m/s in terms of the wind velocity V_f at the front

surface side (upstream side) of the heat transfer fin 13.

[0060] The specific material for the aluminum foam metal used in this experiment is aluminum alloy 6101, as mentioned above. In the same manner as in the first test example, in order to check the influence of the interval H between the wall surfaces for the foam metal No. 1 to No. 3 (10, 20, 40 PIP) of the three types of pore densities described above, fins of the three widths H of 5 mm, 8 mm, and 12 mm were prepared. Thus, a total of nine types of samples were prepared. The height L of the foam metal in the vertical direction was 89 mm, the depth D was 13 mm, and the surface area per unit volume was β .

1) Pressure Loss

[0061] In this case, the pressure loss ΔP under a non-load condition in which there is no flow of cold water is the same as in the first test example described above (refer to the graph of Fig. 6).

[0062] However, when cold water flows under a non-load condition, two states must be considered, one being when the fin surface is dry and the other being when the fin surface is wet.

1-1) Pressure Loss In Dry State (State In Which Fin Surface Is Dry)

[0064] First, the graph of Fig. 12 shows the pressure loss $\Delta P/D$ (Pa/m) with respect to the front surface wind velocity V_f (m/s) in the dry state. In the second test example, since the difference in both dry and wet states are considered, to increase the measurement accuracy of the pressure loss, the pressure loss is calculated by including the air flow direction length D (m) of at the fins ($\Delta P/D$). In the following description, this is simply referred to as ΔP .

[0065] In this case as well, the pressure loss ΔP increases as the pore density PIP increases, that is, as the pore size d_{pore} decreases, and the pressure loss ΔP increases as the fin width H decreases. This is because the surface area (including wall surface) per unit volume increases as the pore size d_{pore} decreases and the fin width H decreases. It is apparent that foam metal has a high pressure loss ΔP compared to the louver fin (width $H=7.9$ mm, $D=13.6$ mm, fin pitch=1.5 mm) of the comparative example.

1-2) Pressure Loss In Wet State (State In Which Fin Surface Is Wet)

[0067] Fig. 13 shows the pressure loss ΔP (Pa) with respect to the front surface wind velocity V_f (m/s) in the wet state in when the temperature of the water serving as the refrigerant is 5°C, and Fig. 14 shows the same state when the temperature of the water serving as the refrigerant is 10°C.

[0068] The influence of the pore density PPI and the fin width H is substantially the same as the tendency in the dry state. However, the value of the pressure loss ΔP greatly increases compared to that in the dry state (see Fig. 12). The pressure loss ΔP greatly increases because the condensed water accumulated on the fin surface becomes a ventilation resistance, and thus can be predicted that water drainage becomes an essential factor in comparison to the dry state. The ratio of the pressure loss ΔP between the wet state and the dry state is shown in Fig. 15 (for a water temperature of 5°C) and Fig. 16 (for a water temperature of 10°C). In the case of water temperature of 5°C in Fig. 15, the ratio of the pressure loss gradually increases as a whole when the air flow rate increases but the pressure loss decreases when the air flow rate increases in the cases of 8 mm, 12 mm at 10°C (Fig. 16).

[0069] This is because the fin efficiency decreases when the supply water temperature is high, the fin width is large, and the wind velocity is fast. Thus, the temperature of the fin surface distant from the wall surface becomes higher than the dew point temperature of the air, the moisture in air cannot condense, and the increasing rate of the pressure loss ΔP decreases. That is, moisture condensation occurs in only parts of the fin.

[0070] In the case of Fig. 15 (water temperature of 5°C), the ratio $\Delta P_{\text{wet}}/\Delta P_{\text{dry}}$ of the pressure loss ΔP in a dry state and in a wet state increases as the pore density PPI increases. The ratio is greater for the foam metal fin than for the louver fin. That is, the water drainage is poorer in the foam metal fin than the louver fin.

[0071] However, the foam metal fin used in the present test is in an experimental level in which the fin surface is not processed. Thus, the problem of water drainage is sufficiently improved, for example, by applying a hydrophilic agent.

1-2) Heat Transfer Coefficient

[0072] In this case as well, two states of the fin surface, a dry state and a wet state, must be considered.

1-2-1) Heat Transfer Coefficient In Dry State

[0073] Fig. 17 shows the relationship of the heat transfer coefficient h_{dry} with respect to the front surface wind velocity V_f (m/s) in a dry state. The heat transfer coefficient h_{dry} increases as the pore density PIP decreases, and the heat transfer coefficient dry decreases as the fin width H increases. The heat transfer coefficient h_{dry} for a foam metal fin of 10 PPI, $H=5$ mm is about the same as the louver fin but becomes inferior to the louver fin if the pore density PPI increases.

[0074] However, since the foam metal fin has a large surface area β per unit volume, to evaluate the heat transfer performance described above, evaluation is performed with the heat transfer performance $h_{\text{dry}}\beta$ per unit volume. Fig.

18 shows the relationship between $h_{dry}\beta$ and the front surface wind velocity V_f (m/s) in the dry state. It is apparent that the heat transfer performance increases as the pore density PIP increases and the fin width H decreases. Fig. 19 shows the ratio of the heat transfer performance of the foam metal fin with respect to the louver fin. In the foam metal fin of pore density 40 PIP and fin width H=5 mm, the heat transfer performance is greater than the louver fin by 1.5 times. This suggests the possibility of effective miniaturization of the heat exchanger.

1-2-2) Heat Transfer Coefficient In Wet State

[0075] Figs. 20 and 21 show the relationship of the heat transfer coefficient h_{wet} in the wet state with respect to the front surface wind velocity V_f (m/s) in the wet state when the temperature of cold water is 5°C and 10°C. When comparing Figs. 20 and Fig. 21, it can be considered that the influence of the change in temperature (5°C to 10°C) of water is not large. Compared with the heat transfer coefficient h_{dry} (see Fig. 17) in the dry state, h_{wet} is slightly smaller than h_{dry} . In addition to the heat transfer of sensible heat in the wet state, this is because the fin efficiency decreases in a dry state due to the heat transfer of latent heat resulting from condensation of the moisture in air.

[0076] Fig. 22 (water temperature 5°C) and Fig. 23 (water temperature 10°C) show the heat transfer performance $h_{wet}\beta$ per unit volume in the wet state. As can be seen, the increase in $h_{wet}\beta$ of the foam metal fin with respect to the louver fin is greater than in the dry state and is about 1.8 times greater for the pore density of 40 PPI and the fin width of H=5 mm. That is, the heat transfer promoting effect in the wet state is greater than in the dry state.

[0077] Fig. 24 (water temperature 5°C) and Fig. 25 (water temperature 10°C) show the relationship between substance transfer coefficient h_{mass} and air flow rate V_f (m/s). In the same manner as the heat transfer coefficient h_{wet} of Fig. 20 (water temperature 5°C), the substance transfer coefficient h_{mass} of Fig. 24 (water temperature 5°C) increases as the entire air flow rate and fin width H increase and as the pore density PIP decreases. However, when H=12 mm and PPI=20, 40, it does not increase even if the wind velocity rises from around the speed of 1.0m/s. Similar to the explanation for the pressure loss, this is because the temperature of part of the fin surface becomes higher than or equal to the dew point temperature and the moisture does not condense. This tendency is more significant for the water temperature of 10°C in Fig. 25. This is also seen in the substance transfer coefficient $h_{mass}\beta$ (kg/m³s)- V_f (m/s) per unit volume (not shown).

3) Comprehensive Analysis of Pressure Loss and Heat Transfer Coefficient

[0078] As described above, the foam metal fin of the present embodiment has higher pressure loss and higher heat transfer coefficient per volume compared to the existing louver fin. However, in order to be configured as an air conditioner heat exchanger, the pressure loss and the heat transfer coefficient must be comprehensively analyzed. The pump power necessary for unit volume is expressed by the following equation.

$$E\beta = \frac{V_f A_c \Delta P}{A_t} \frac{A_t}{V} \quad \text{--- (1)}$$

[0079] V is the volume, and A_c is the flow cross-sectional area.

[0080] Figs. 26 and 27 show the heat transfer coefficients $h_{dry}\beta$, $h_{wet}\beta$ per unit volume for the dry state and the wet state in relation to the pump power EBA necessary per unit volume. In the dry state of Fig. 26, the highest heat transfer coefficient is obtained with 40 PIP and H=5 mm, and the heat transfer performance improvement effect of about 24% with respect to the louver fin is expected. In the wet state of Fig. 27, for 40 PPI, H=5 mm, 8 mm, and 12 mm, the heat transfer performance improvement effect is higher by about 28%.

[0081] As shown in Fig. 28, this is also seen in the mass heat transfer coefficient $h_{mass}\beta$ (kg/m³s)- $E\beta$ (w/m³) per unit volume.

(Second embodiment)

[0082] Fig. 29 shows the structure for an air-heat exchanger according to a second embodiment of the present invention.

[0083] The present embodiment relates to a serpentine heat exchanger in which a flat heat transfer tube 21 is bent into a serpentine shape as a single continuous structure.

[0084] The high heat transfer performance is also achieved in the heat exchanger of such configuration in the same manner as the heat exchanger of the first embodiment described above.

(Third embodiment)

[0085] Fig. 30 shows the structure for a heat exchanger according to a third embodiment of the present invention.

[0086] In this embodiment, a plurality of plate shaped heat transfer tubes 31 extending in the horizontal direction are connected in a stack form by left and right connecting members 22, and the heat transfer fin 13 made of foam metal is arranged between the connecting members 22 of each layer. Each heat transfer tube 31 includes refrigerant inlet and outlet holes 23. The holes 23 connected through the connecting members 22 form a refrigerant passage. The connecting members 22 are used for a structure similar to that of the first embodiment in a stacked plate type air-heat exchanger.

[0087] In such a configuration, high heat transfer performance is realized in the same manner as in the first embodiment.

(Other Embodiments)

[0088] The heat transfer fin of the present invention is not limited to the structure of the heat exchanger in each embodiment and may obviously be applied to a heat transfer fin for performing heat exchange with air, such as a cross fin type or the like.

Claims

1. A heat transfer fin for a heat exchanger, wherein a heat transfer fin (13) is arranged on a heat transfer tube (1) through which fluid flows for exchanging heat with air, the heat transfer fin (13) contacts air to exchange heat, and the heat transfer fin (13) is made of foam metal having a pore density of 20 PIP or greater.
2. The heat transfer fin for a heat exchanger according to claim 1, wherein a heat transfer tube (1) through which fluid flows for exchanging heat with air is provided as in a plurality, and the plurality of heat transfer tubes (1) are set at an interval H of 12 mm or less.
3. The heat transfer fin for a heat exchanger according to claim 1 or 2, wherein the heat exchanger is a stacked type heat exchanger.
4. The heat transfer fin for a heat exchanger according to any one of claims 1 to 3, wherein the pore density is 20 PIP or greater and 60 PIP or less.
5. The heat transfer fin for a heat exchanger according to claim 2, wherein an interval H between the plurality of heat transfer tubes (1) is set at 4 mm or greater and 12 mm or less.

Fig.1

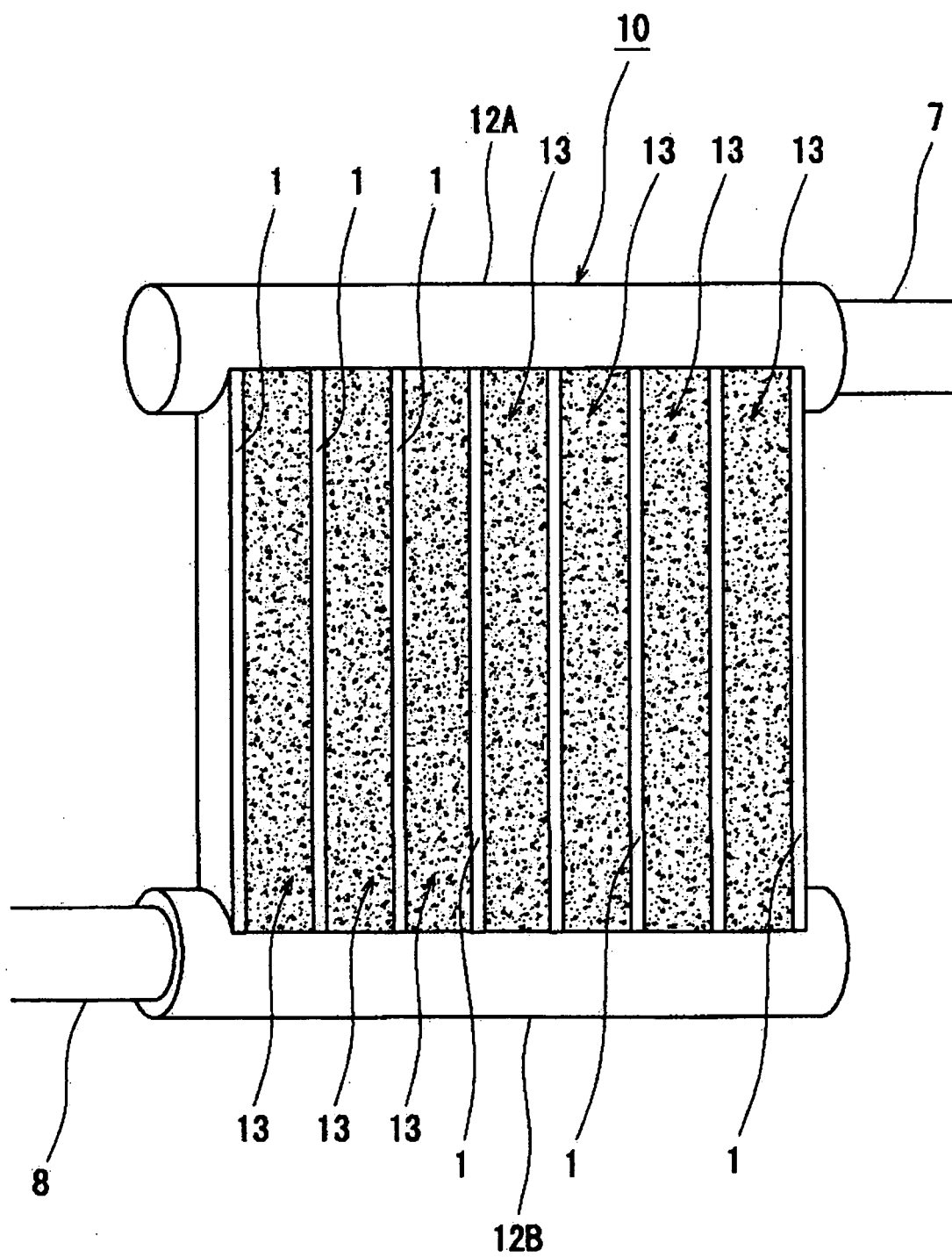


Fig.2

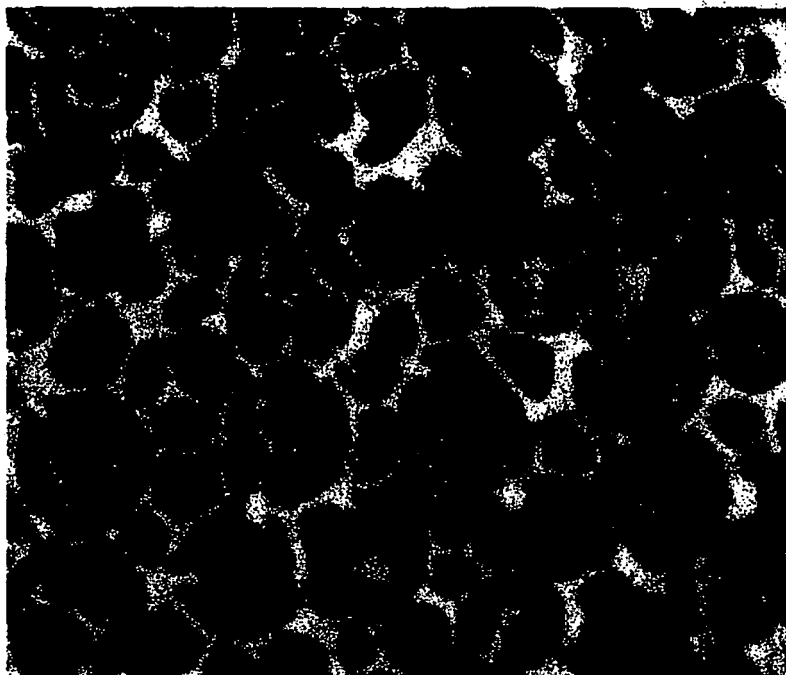
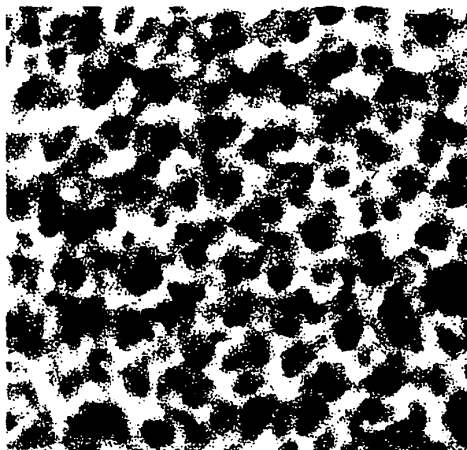
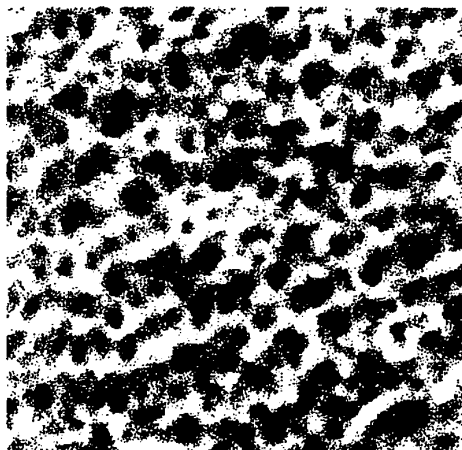


Fig.3

(A)



(B)



(C)

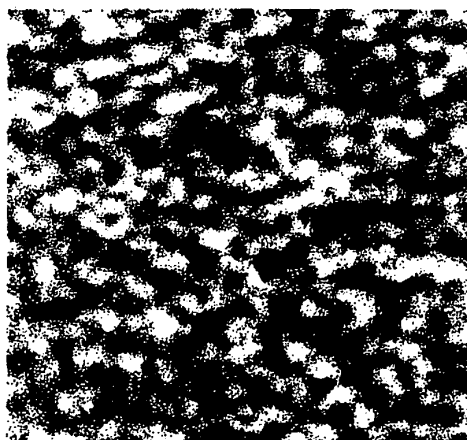


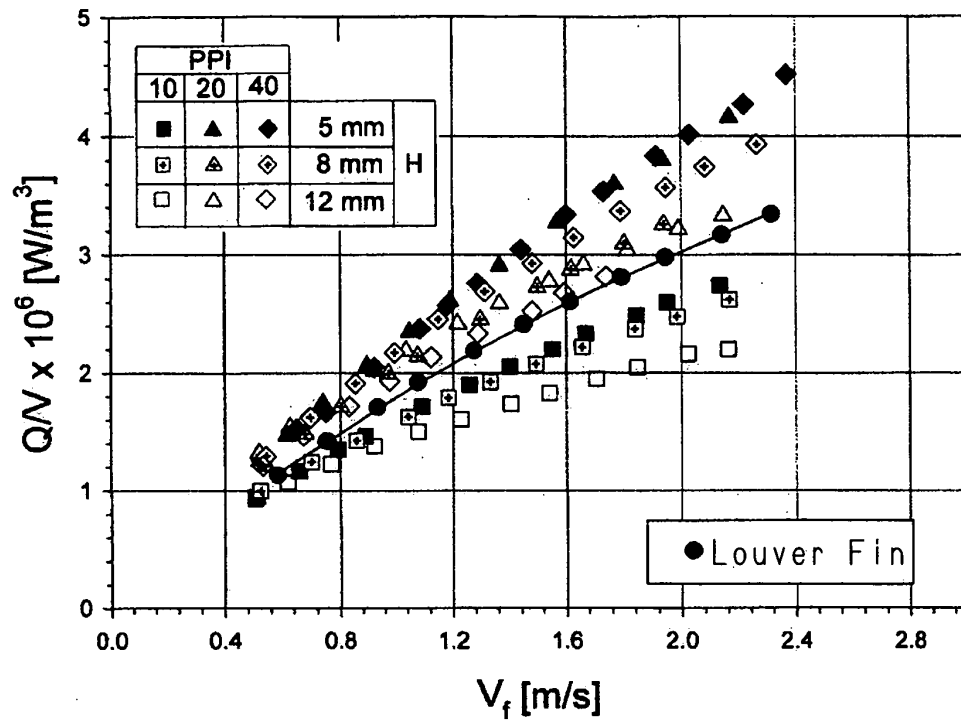
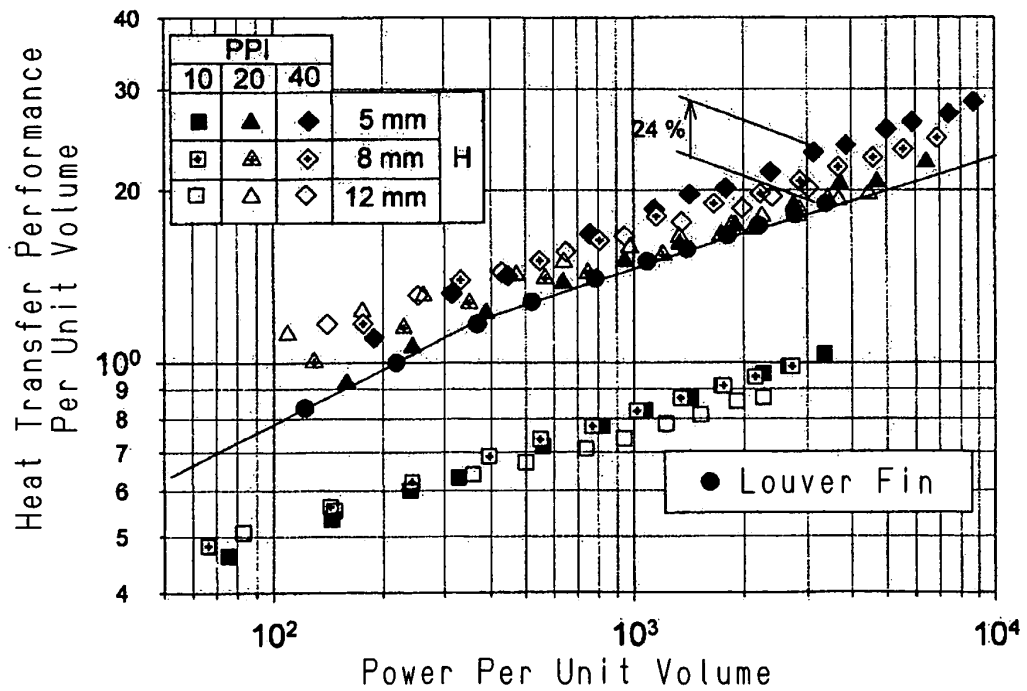
Fig. 4**Fig. 5**

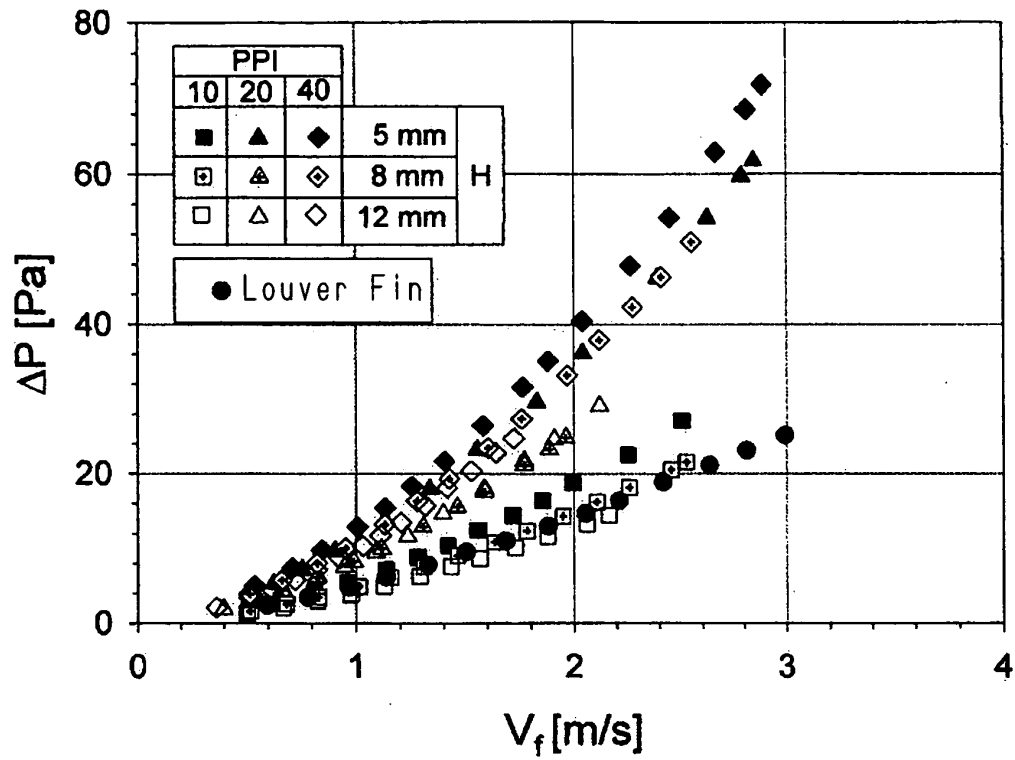
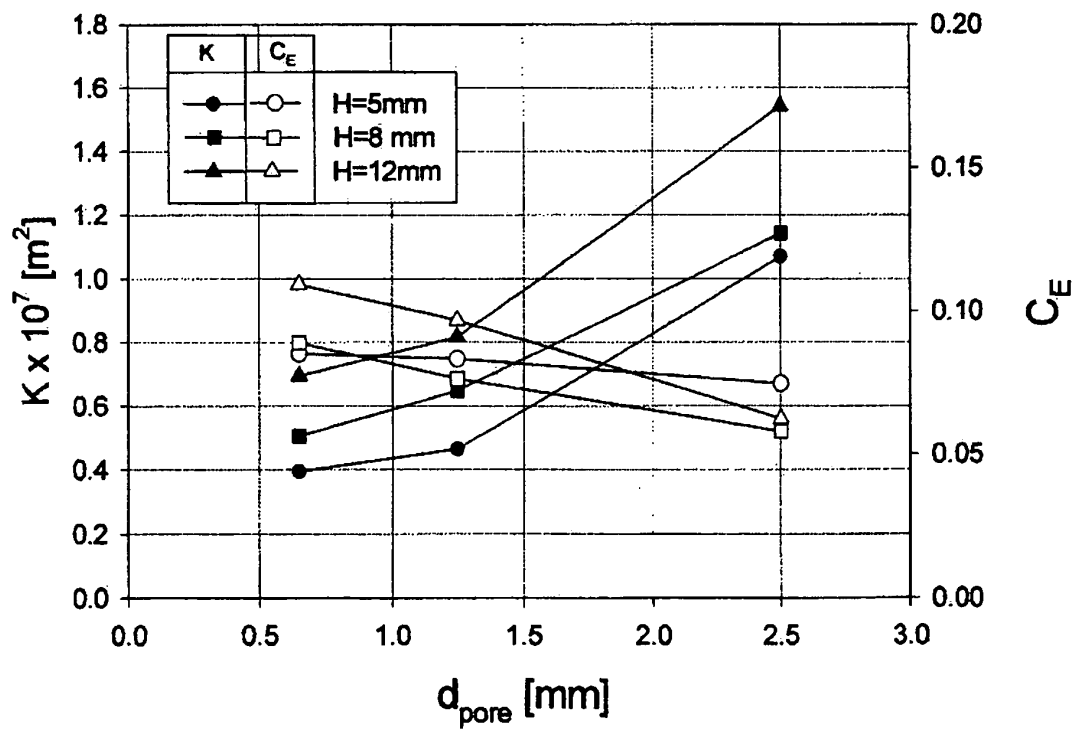
Fig. 6**Fig. 7**

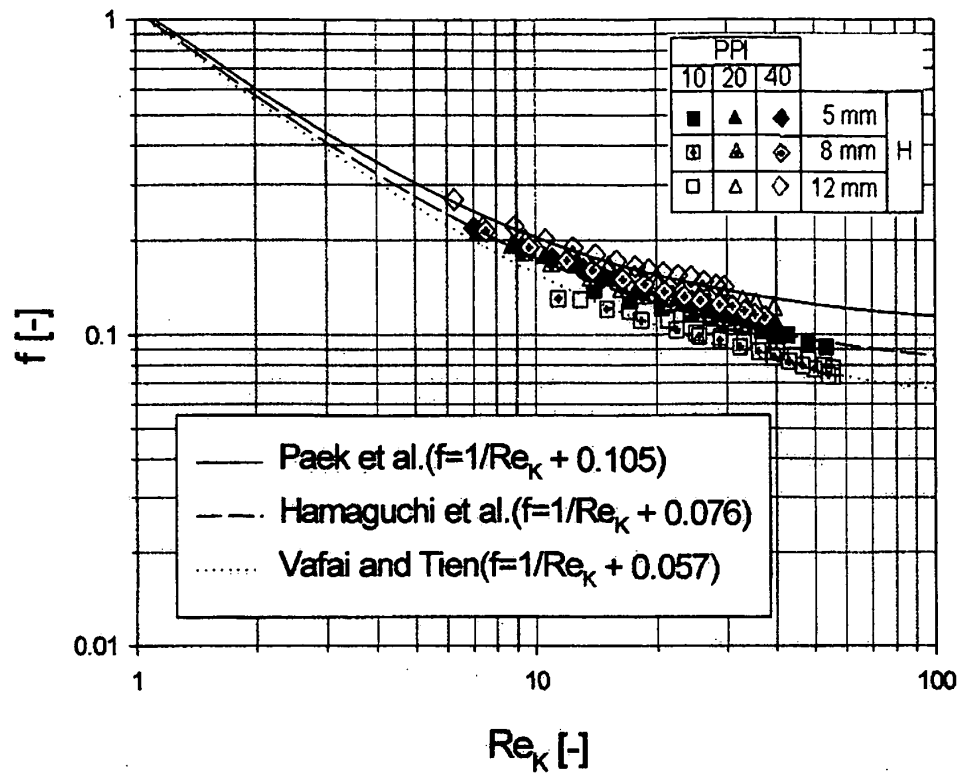
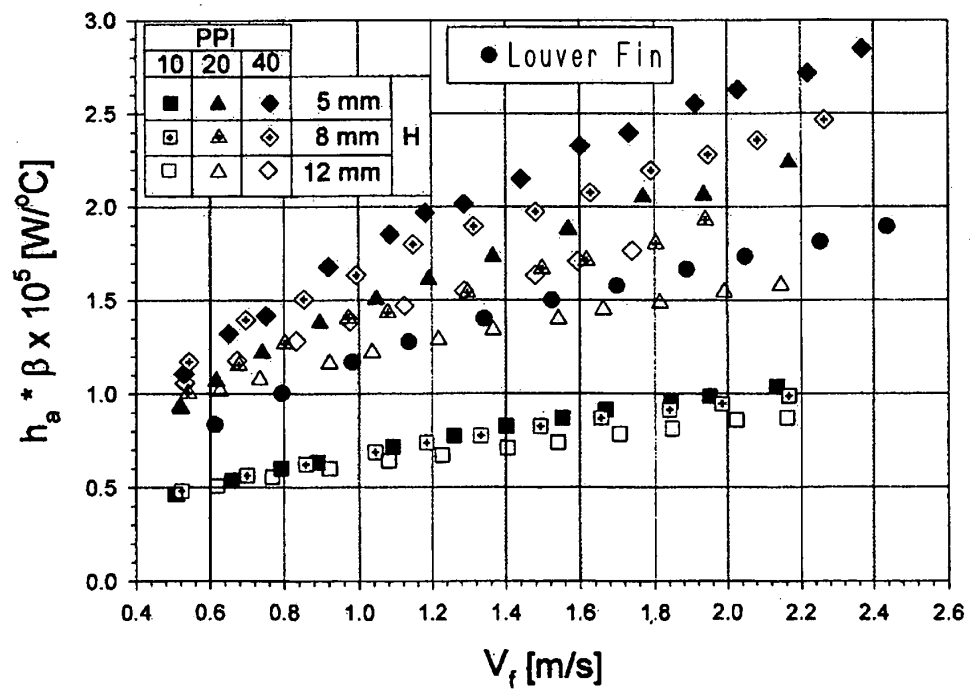
Fig. 8**Fig. 9**

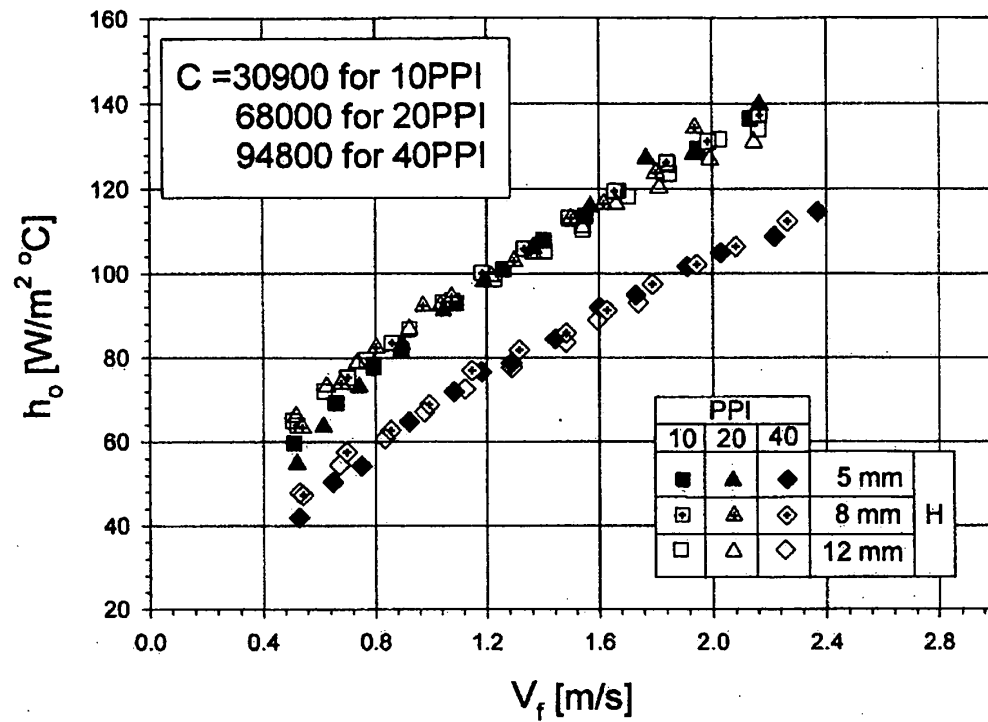
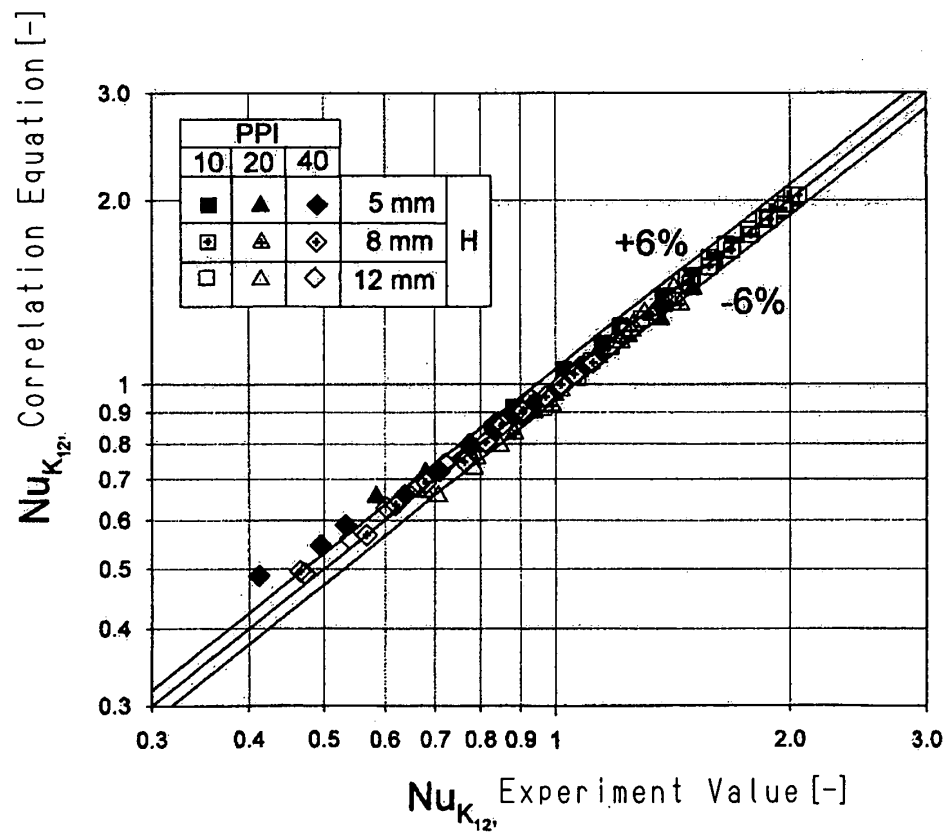
Fig.10**Fig.11**

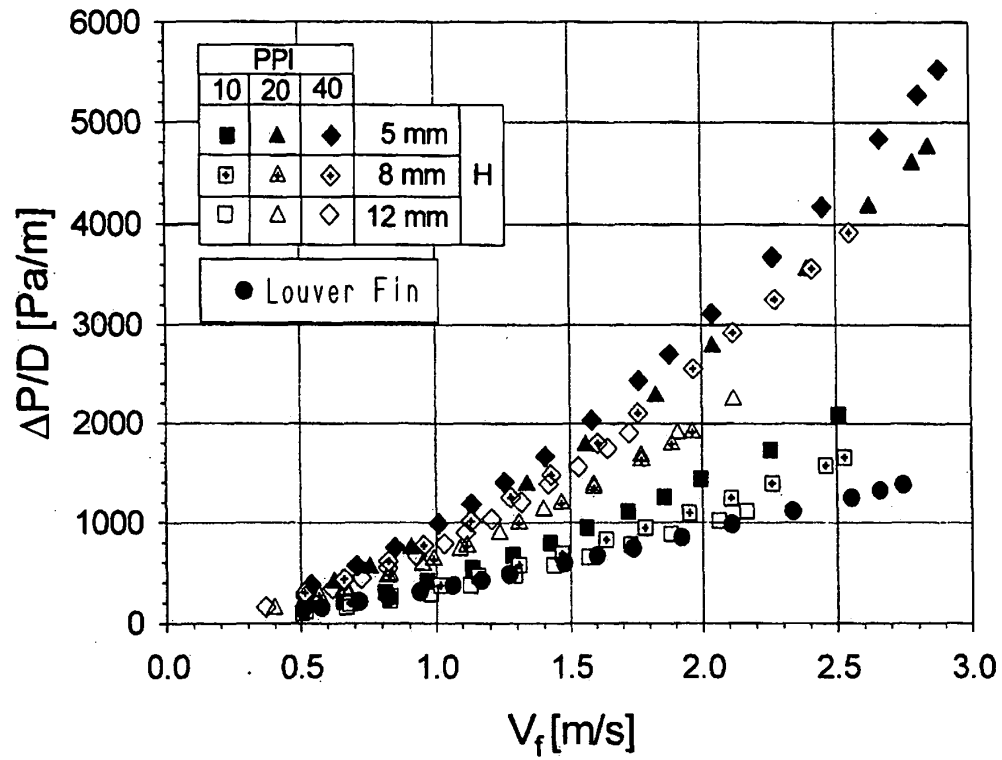
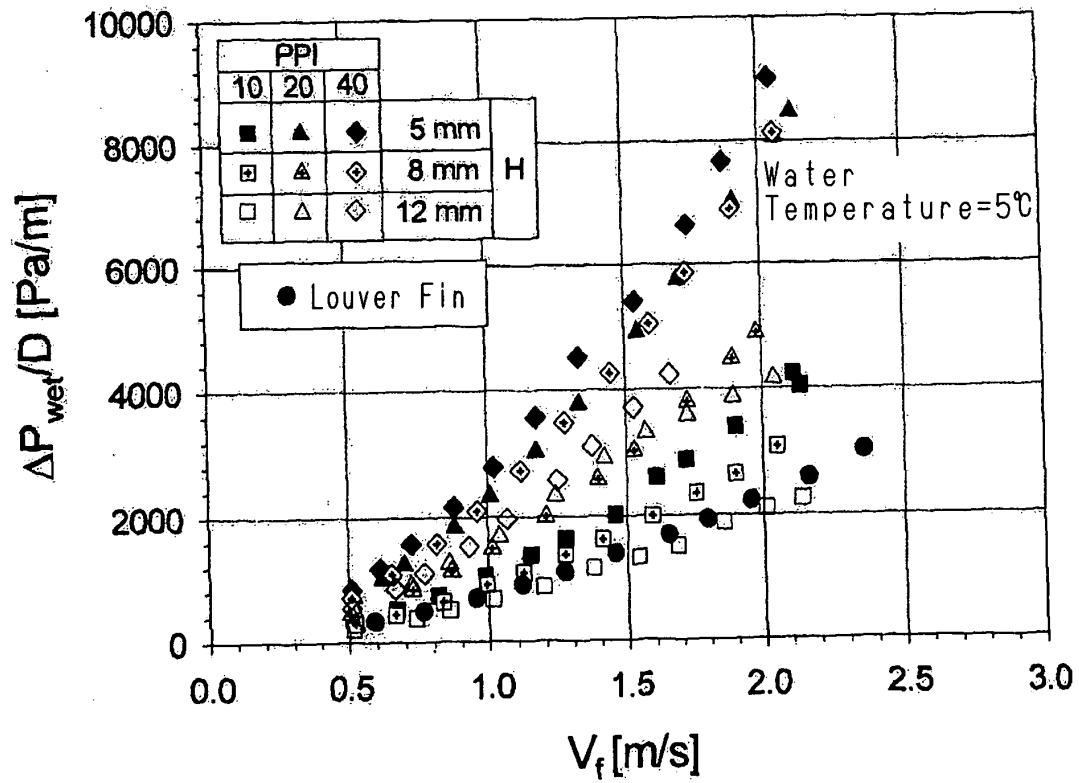
Fig.12**Fig.13**

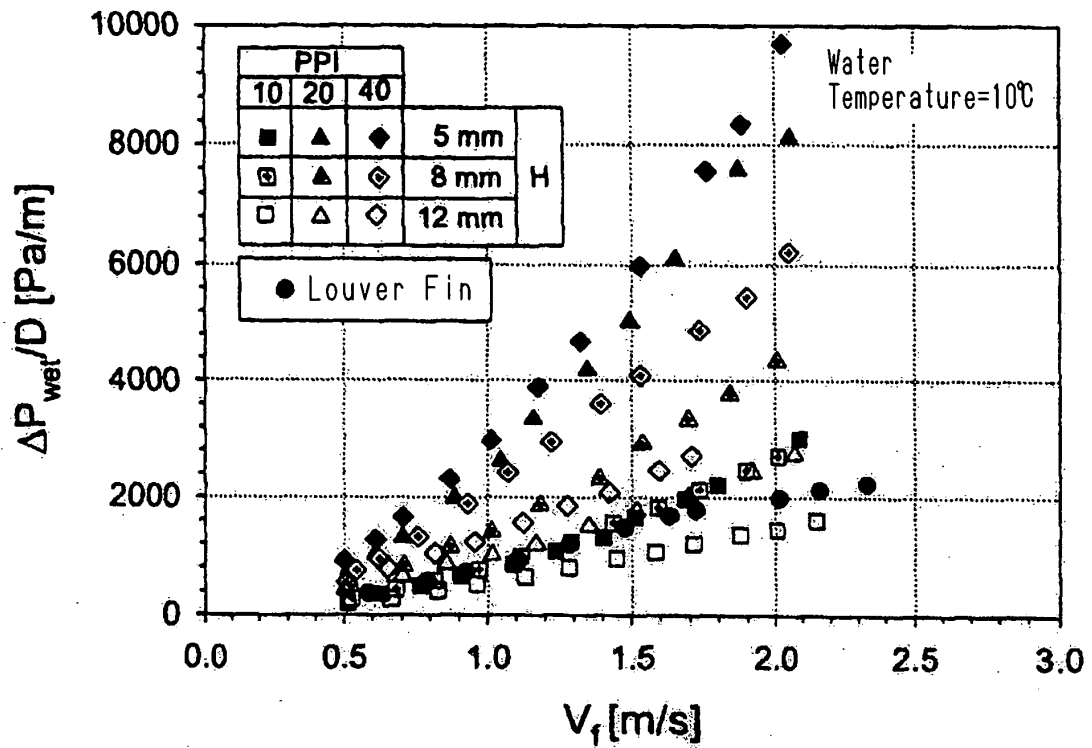
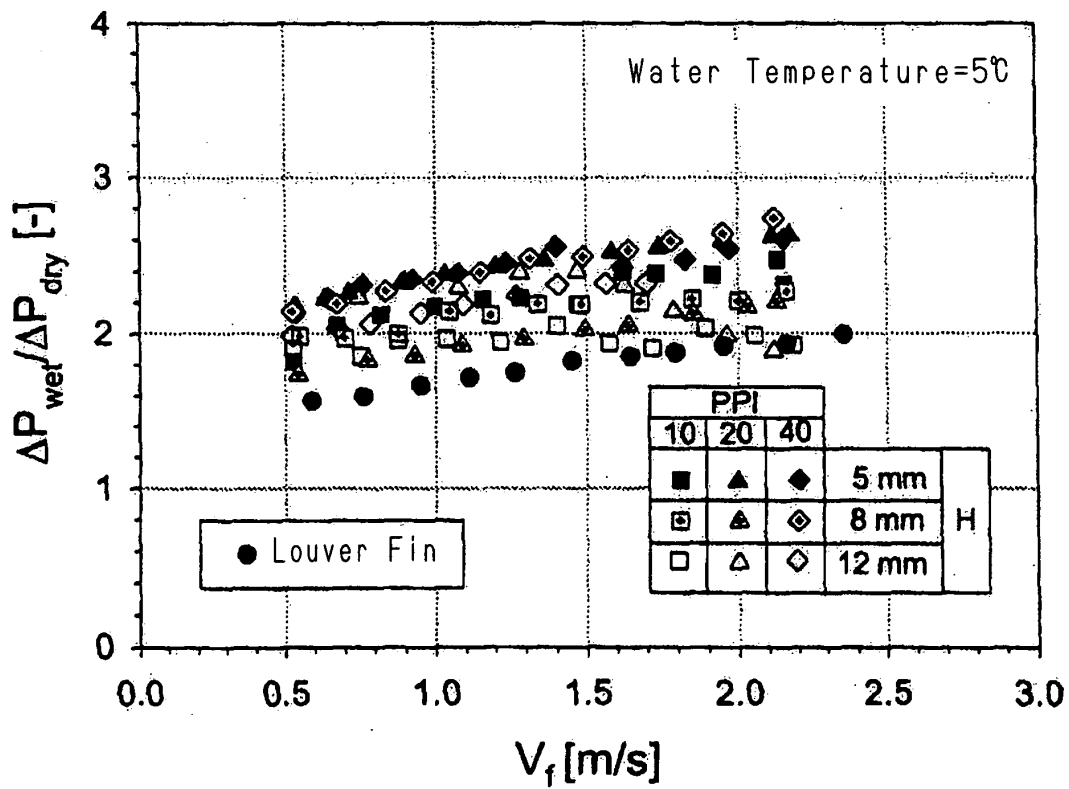
Fig.14**Fig.15**

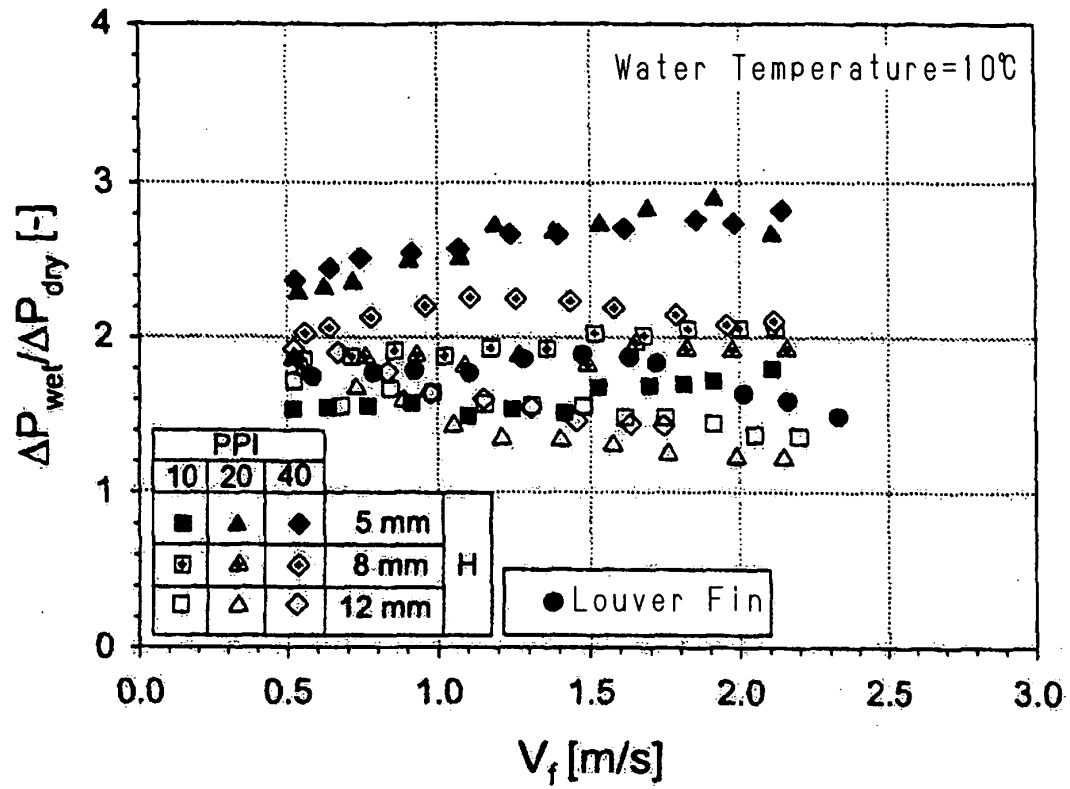
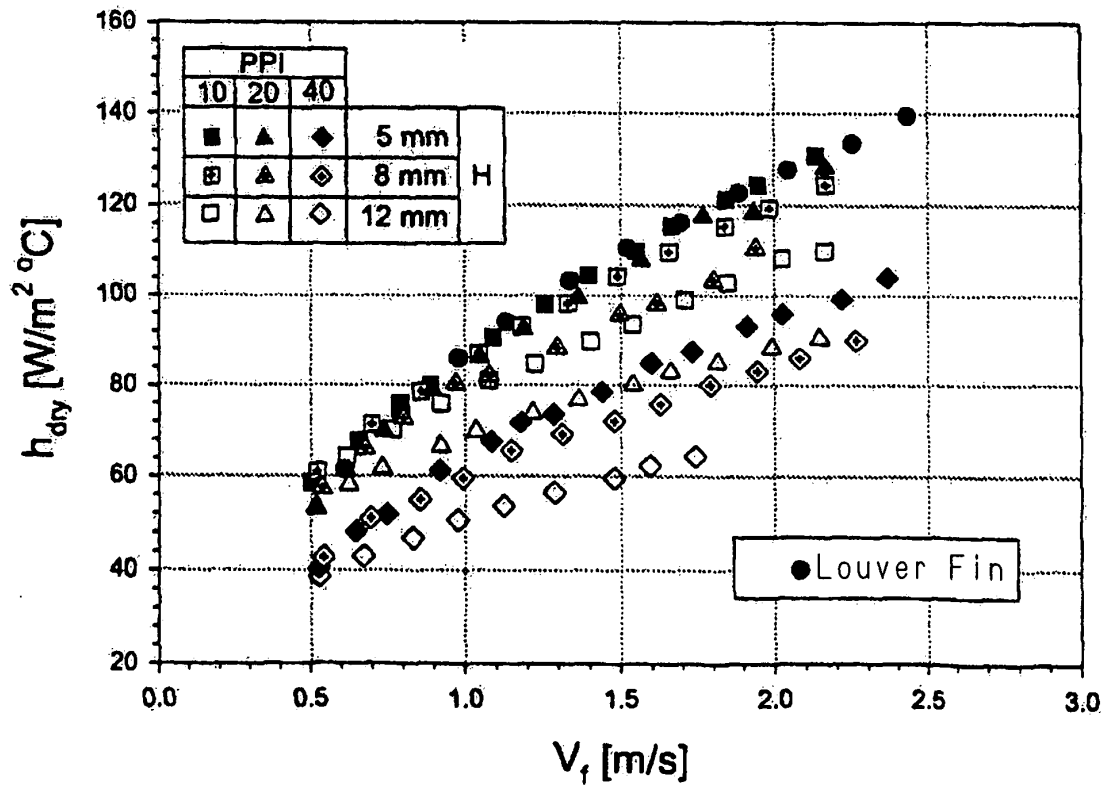
Fig.16**Fig.17**

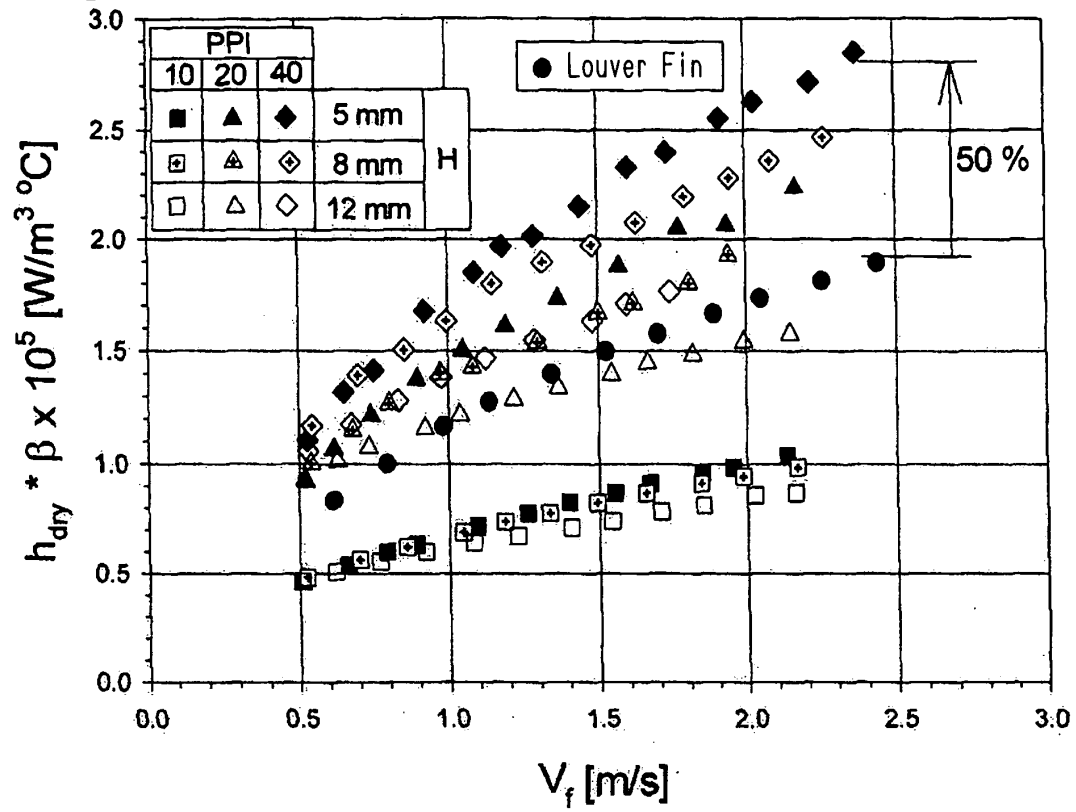
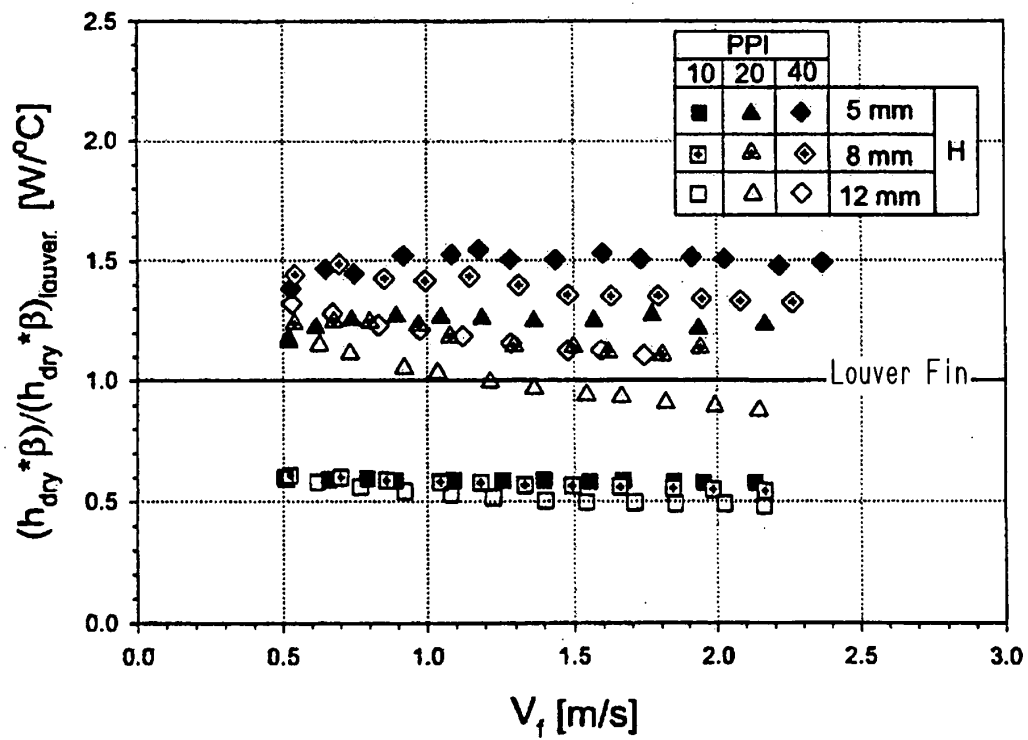
Fig.18**Fig.19**

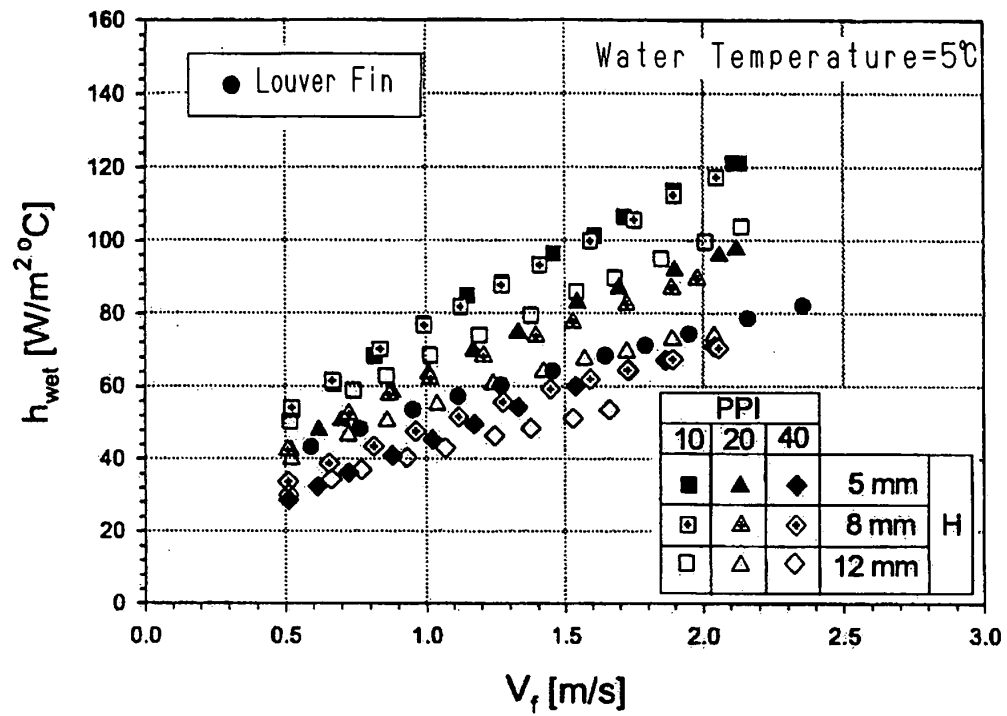
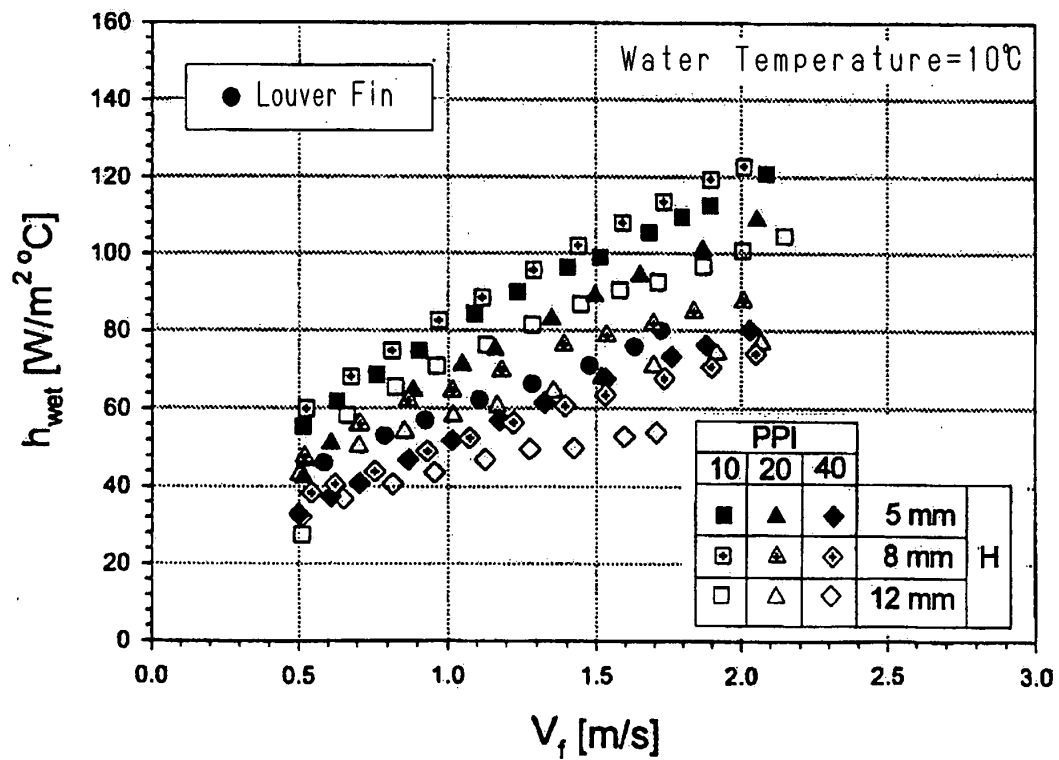
Fig.20**Fig.21**

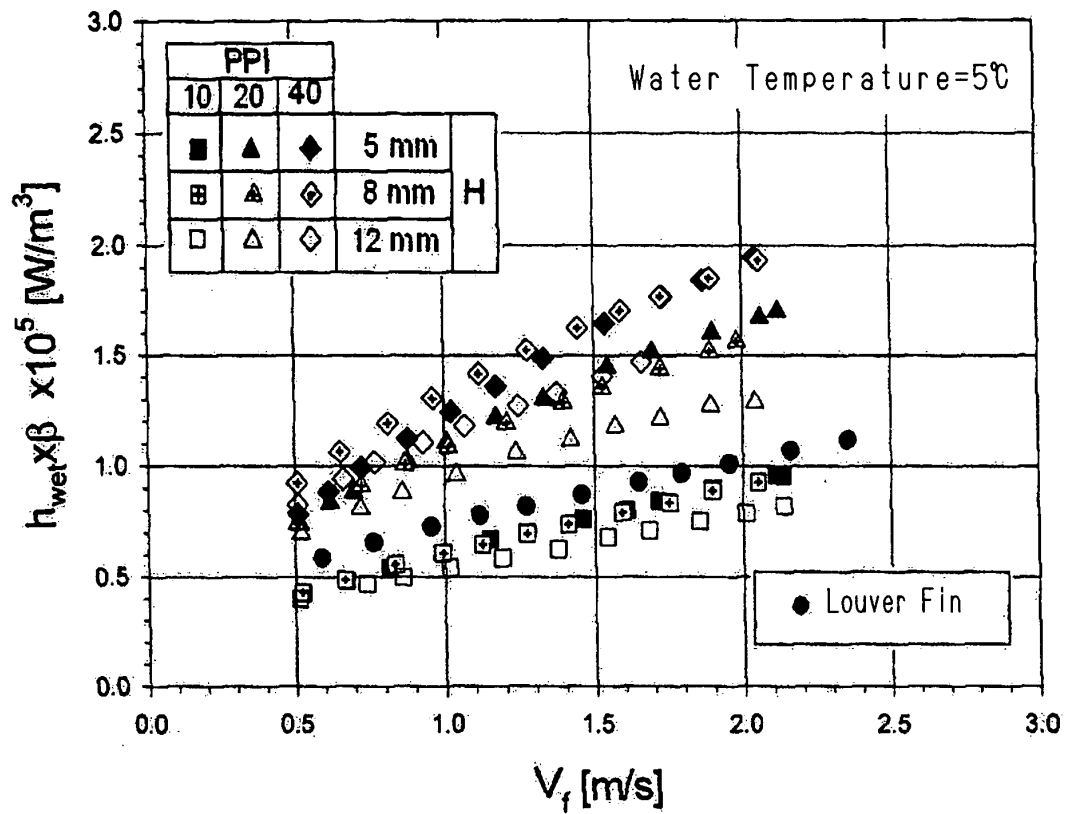
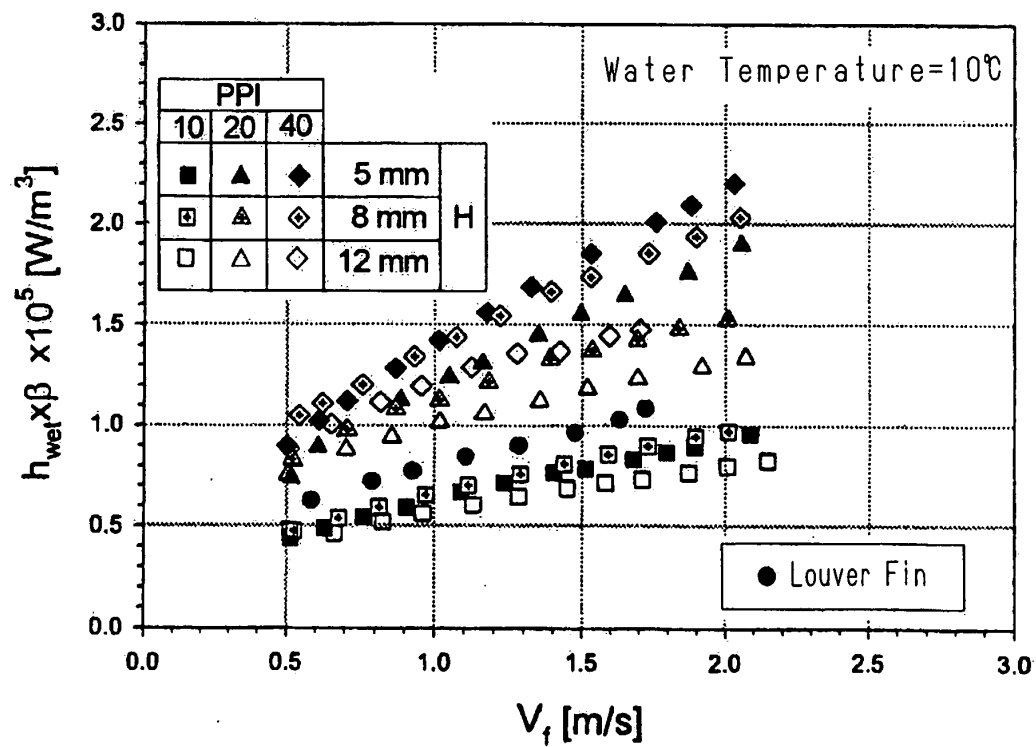
Fig. 22**Fig. 23**

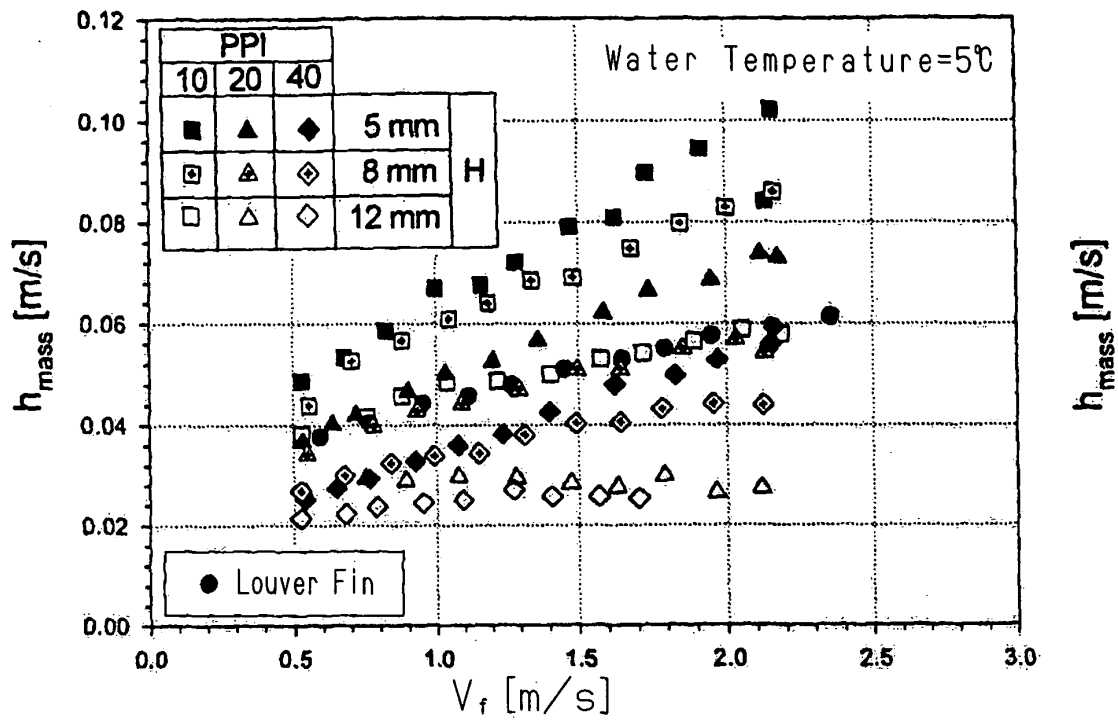
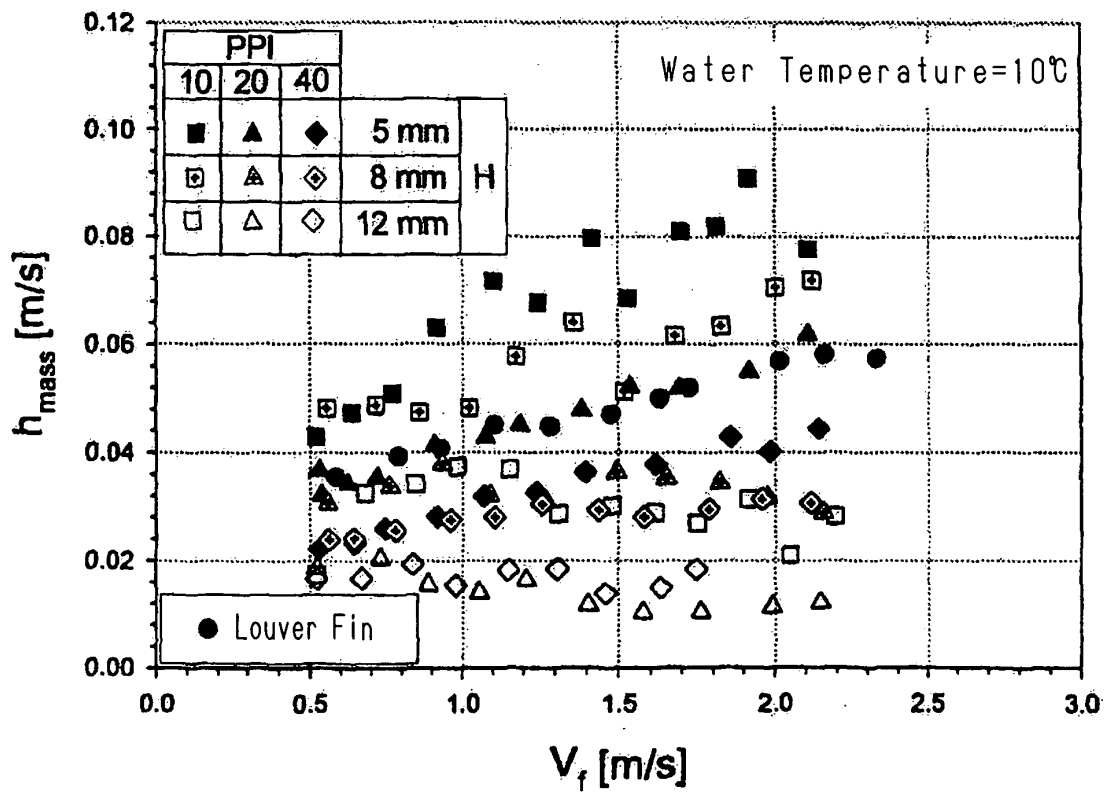
Fig.24**Fig.25**

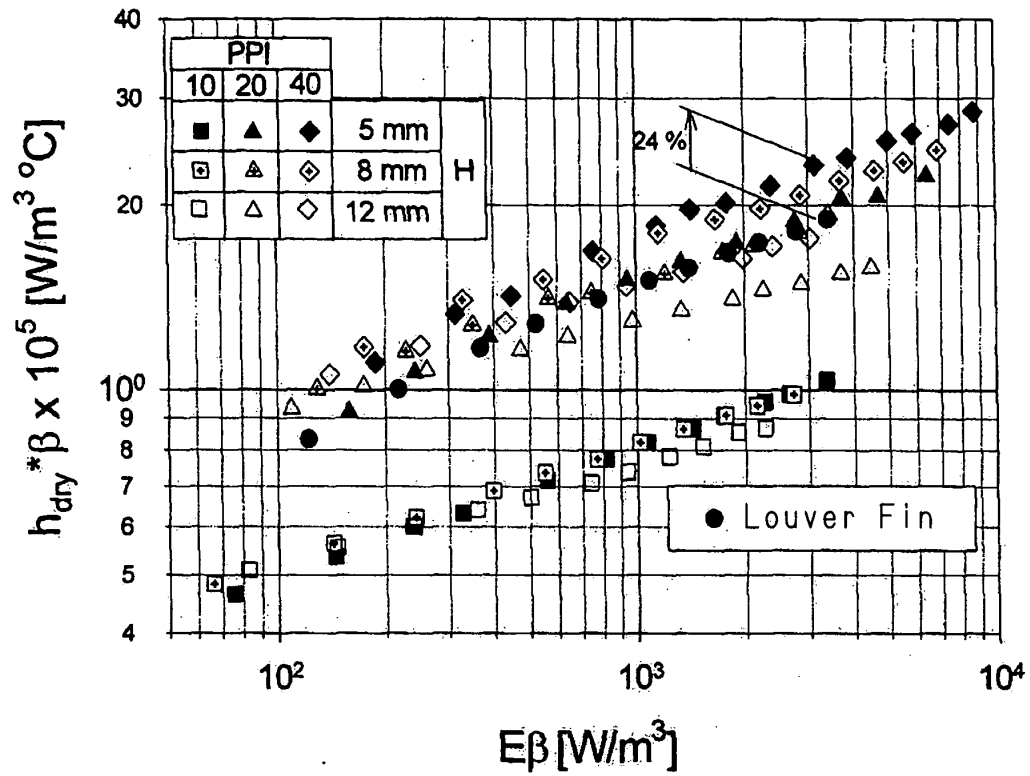
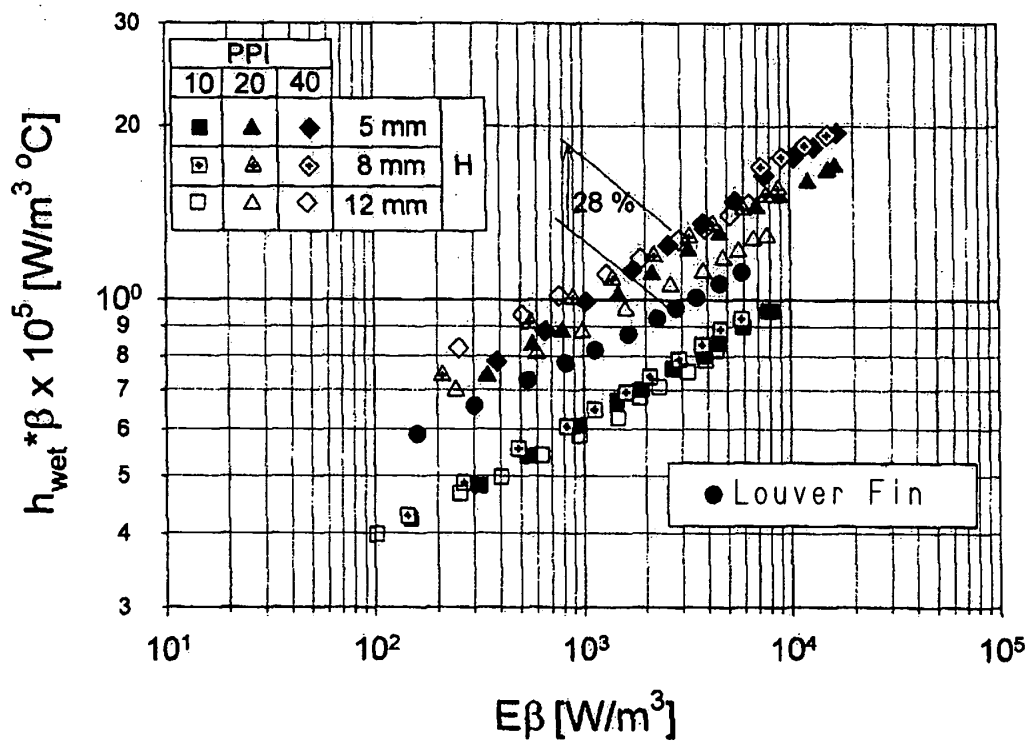
Fig.26**Fig.27**

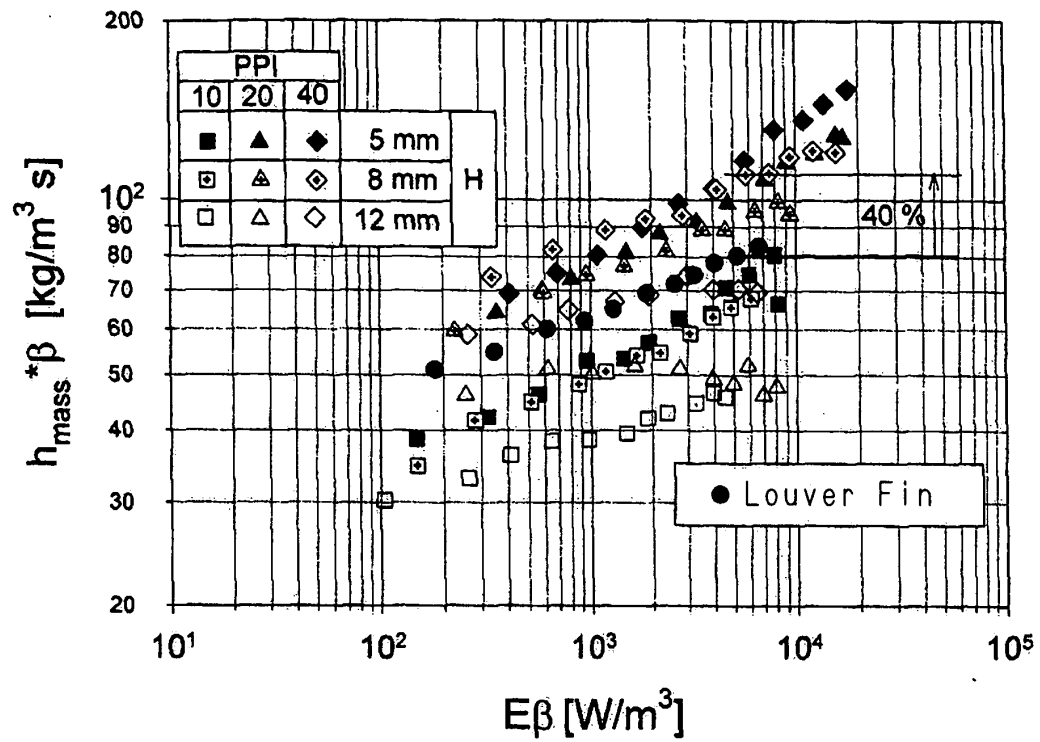
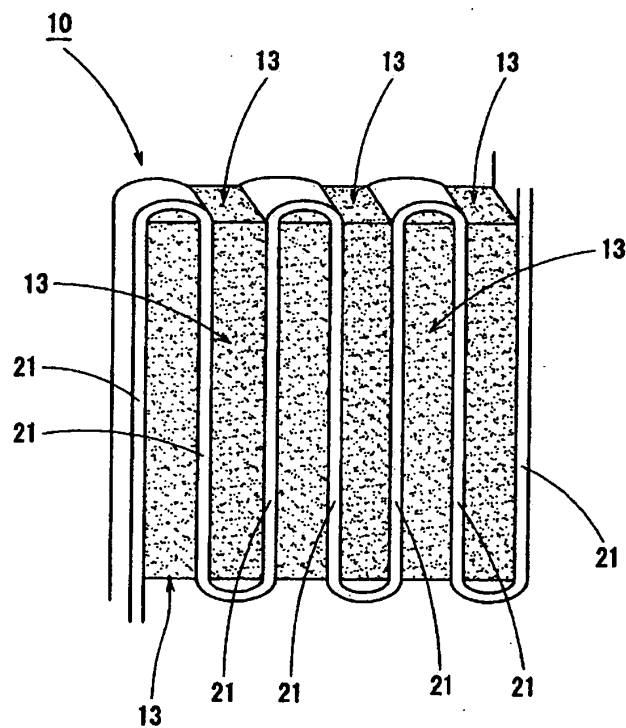
Fig.28**Fig.29**

Fig.30

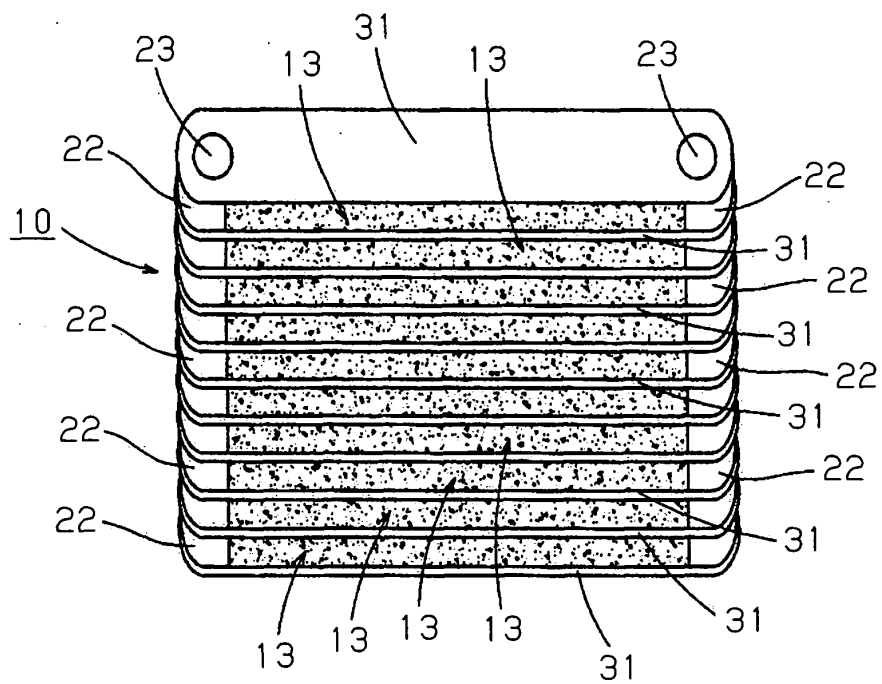


Fig.31

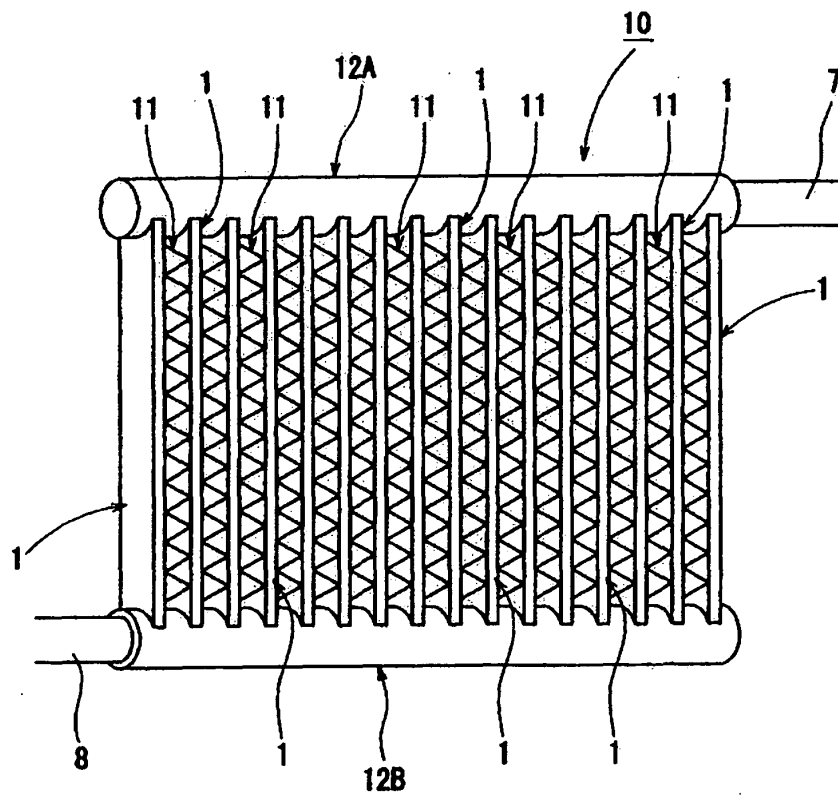
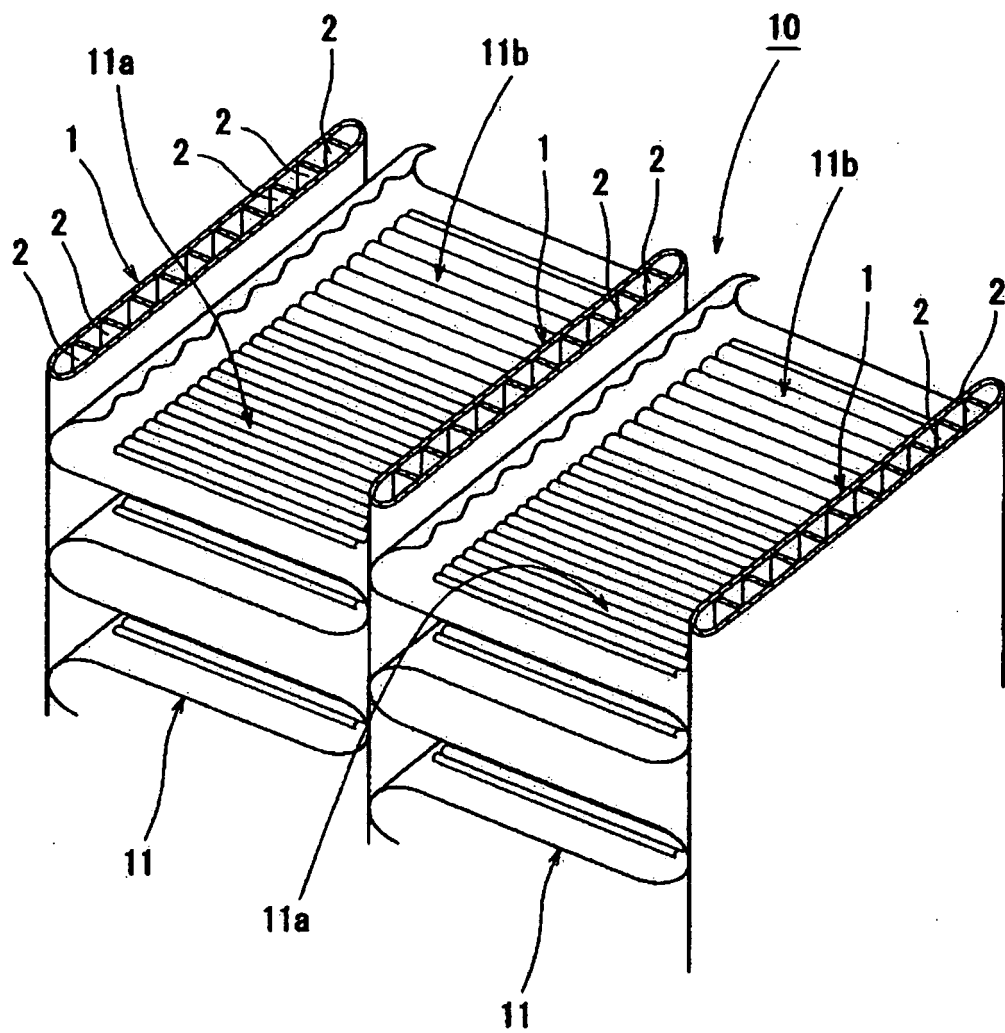


Fig.32



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2005/007383

A. CLASSIFICATION OF SUBJECT MATTER Int.Cl. ⁷ F28F1/10, F28D1/053		
According to International Patent Classification (IPC) or to both national classification and IPC		
B. FIELDS SEARCHED Minimum documentation searched (classification system followed by classification symbols) Int.Cl. ⁷ F28F1/10, F28D1/053		
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 1996-2005 Kokai Jitsuyo Shinan Koho 1971-2005 Toroku Jitsuyo Shinan Koho 1994-2005		
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	WO 2002/042707 A1 (STORK SCREENS B.V.), 30 May, 2002 (30.05.02), Page 9, line 36 to page 12, line 15 & JP 2004-522121 A page 7, line 14 to page 8, line 38 & US 2004/0226702 A1 & EP 001346184 A1 & NL 001016713 C	1, 2, 4, 5 3
Y	JP 52-134153 A (Nippondenso Co., Ltd.), 10 November, 1977 (10.11.77), Page 2, upper left column, line 9 to upper right column, line 7; Figs. 4, 5 (Family: none)	3
☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.		
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Date of the actual completion of the international search 11 July, 2005 (11.07.05)		Date of mailing of the international search report 02 August, 2005 (02.08.05)
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