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Fine-tube heat exchanger woven with threads†

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Abstract—A new type of compact heat exchanger is suggested, and the fundamental heat transfer performance is studied chiefly from experimental aspects. The design concept is based on the new principles for heat transfer enhancement, namely 'reducing the size of heat transfer surfaces' and 'arranging a couple of turbulence promoters so as to cause drastic change in the turbulence structure'. To satisfy such requirements, the heat exchanger consists of fine tubes (o.d. ~ 1 mm) and woven threads (diameter ~ 0.3 mm); the latter play a combined role of both turbulence promoter and fin. The maximum heat transfer coefficient obtained in the experiment is 14-fold larger than that around a cylinder without threads. Owing to the combination of the woven threads and the very small characteristic length, the heat transfer coefficient per unit projected area is about 5×10^3 W m^{-2} K^{-1}, and that per unit volume reaches 3×10^6 W m^{-3} K^{-1}.

1. INTRODUCTION

THE RESEARCH and development of compact heat exchangers have a very long history. Since the first publication of *Compact Heat Exchangers* [1], which is the most extensive reference source in this field, various techniques of heat transfer enhancement have been tested, and a number of heat exchangers have been developed. Among them, the louvered-fin heat exchanger developed for an automotive radiator [2–4] has the highest heat transfer performance; its heat transfer is enhanced by means of its interrupted boundary layers, i.e., the thin boundary layers formed near the leading edges of louvered fins. This heat exchanger so efficiently utilizes the leading-edge effect that one may imagine that no compact heat exchanger will be superior to it, and thus the possibility of new techniques has been exhausted.

Such a view is correct if based on the conventional principle for heat transfer enhancement, but may not be valid if based on other principles. As alternatives in looking for a breakthrough in heat transfer enhancement, the authors consider the following principles:

(1) reducing the characteristic size of heat transfer surfaces; and
(2) arranging a couple of turbulence promoters so as to work in conjunction with each other.

The first principle means that heat transfer elements must be as small as possible because the heat transfer coefficient increases with decreasing characteristic length. On the other hand, the second principle denotes that two turbulence promoters installed upstream and downstream of the heat transfer surface work organically so that large-scale eddies in a free stream are induced; that is, unlike conventional turbulence promoters, which simply produce turbulence near heat transfer surfaces, the present turbulence promoters play a role in controlling the flow, the scale of which is comparable to the heat transfer surfaces.

As an example of a heat exchanger derived from these principles, we suggest a 'fine-tube heat exchanger woven with threads', as shown in Fig. 1. In order to satisfy the aforementioned principles, the heat exchanger consists of fine tubes (o.d. ~ 1 mm) and threads (diameter ~ 0.3 mm). When the working gas flows around the threads installed upstream and downstream of the fine tubes, the flow inevitably separates; eddies resulting from the repeated separations are expected to interact.

In addition to these features, the following advantages are associated with the fine-tube heat exchanger.

(1) The element bears high pressure because of the relatively thick tube walls. As a result, it can be applied to a superheater for ultracritical pressure boilers.
(2) Since the element is just like a textile fabric, the structure can be held without any fixer such as solder. Hence, the fine-tube heat exchanger can be used under high-temperature as well as high-pressure conditions.

In the present work, mainly from experimental aspects, we investigate the basic heat transfer characteristics of the fine-tube heat exchanger, and explore the optimum geometry which produces the best performance. A tentative comparison with the louver-fin heat exchanger is made in the Appendix.

2. APPARATUS AND TECHNIQUES

The magnified photographs of a fine-tube heat exchanger are shown in Fig. 2. It is composed of fine tubes (outside diameter $d_o = 1$ mm, inside diameter $d_i = 0.7$ mm, and transverse tube pitch $p = 2$ mm)

† Dedicated to Professor Dr.-Ing. Dr.-Ing.e.h. Ulrich Grigull.

NOMENCLATURE

A_f surface area of the thread (fin) [m^2]
A_o surface area of a fine tube [m^2]
Bi Biot number $= \alpha_f d_f/\lambda_f$
d_f thread diameter [m]
d_i inside diameter of a fine tube [m]
d_o outside diameter of a fine tube [m]
E pumping power $= u_o \Delta P$ [W m^{-2}]
l_f length of a thread (fin) [m]
n number of fine tubes per unit width $= p^{-1}$ [m^{-1}]
Nu_o average Nusselt number around a fine tube without threads, $\alpha_o d_o/\lambda_a$
Nu^* average Nusselt number around a fine tube with threads $\alpha^* d_o/\lambda_a$
p transverse tube pitch [m]
ΔP pressure drop [Pa]
Q total heat transfer rate of heat exchanger [W]
r_f radius of a thread [m]
R, Φ, Z dimensionless coordinates
r, Φ, z coordinates [m, —, m]
Re_f Reynolds number around a thread $= u_f d_f/\nu$
Re_o Reynolds number around a fine tube $= u_o d_o/\nu$
t solder thickness [m]
u_f air velocity approaching threads [m s^{-1}]
u_o air velocity approaching fine tubes [m s^{-1}].

Greek symbols
α_f average heat transfer coefficient around a thread [W m^{-2} K^{-1}]
α_o average heat transfer coefficient around a fine tube without threads [W m^{-2} K^{-1}]
α^* average heat transfer coefficients around a fine tube with threads $= Q/(A_o \Delta\theta)$ [W m^{-2} K^{-1}]
η fin efficiency
Θ dimensionless temperature $= (\theta - \theta_a)/(\theta_w - \theta_a)$
θ temperature [K]
$\Delta\theta$ logarithmic mean temperature difference between air and fine-tube wall [K]
λ thermal conductivity [W m^{-1} K^{-1}]
ν kinematic viscosity [m^2 s^{-1}].

Subscripts
a air (main stream)
f threads (fin)
i inside a fine tube
o outside a fine tube
w fine-tube wall.

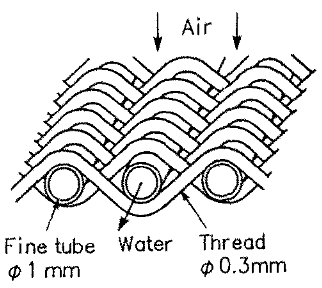

FIG. 1. Fine-tube heat exchanger woven with threads.

and woven threads (diameter $d_f = 0.3$ mm). When the elements are fabricated simply by the weaving process, the contact area between the fine tubes and woven threads is so small that the fin effect by the threads is hardly achieved. Hence, to bring about the fin effect as well as the effect of turbulence promoters, elements plated with solder of 35 μm thickness were also provided (the maximum contact width is about 300 μm). The influence of the fin efficiency was examined by trying different materials for the elements; elements made entirely of copper were provided in addition to those made of stainless steel. The characteristic dimensions of the typical test elements are summarized in Table 1, where we evaluate the surface area

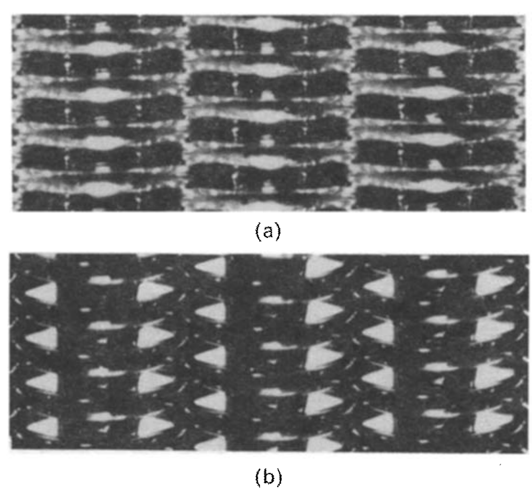

FIG. 2. Magnified photographs of fine-tube heat exchanger: (a) front view; (b) oblique view.

Table 1. Characteristic dimensions of the fine-tube heat exchanger ($d_o = 1.0$ mm, $d_f = 0.3$ mm)

Transverse tube pitch, p (mm)	Area ratio, A_f/A_o	Surface area density (m^2 m^{-3})
2	2.44	3380
4	4.21	2560

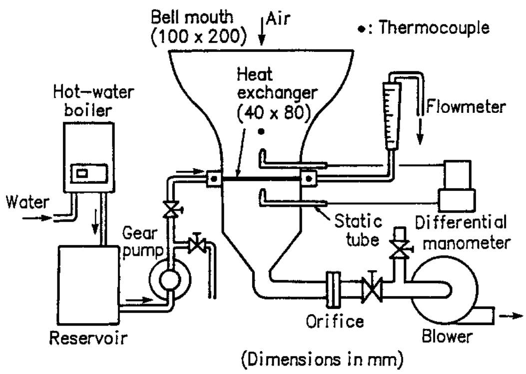

FIG. 3. Schematic of experimental facility.

density by dividing the total surface area per unit projected area by the element thickness ($d_o + 2d_f$).

A schematic diagram of the experimental facility is shown in Fig. 3. Air at room temperature was induced to flow through the wind tunnel by the blower. As the working fluid inside the heat exchanger, water heated by a hot-water boiler (about 70°C) was used, and supplied to the heat exchanger by the gear pump. (The mean water velocity inside the tube was about 1 m s^{-1}, and the pressure drop therein was about 1000 Pa.) Water inlet and outlet temperatures were measured at the entrance and exit pipes of the heat exchanger. At both locations the water is well mixed, so the bulk water temperature could be measured by means of a single thermocouple. Two static tubes installed upstream and downstream of the heat exchanger were used to obtain the pressure drop across the heat exchanger.

The overall heat transfer coefficient was determined by dividing the total heat flow (the product of the water temperature difference at the heat exchanger and the heat capacity rate) by logarithmic mean temperature difference and the total outside area. The inside heat transfer resistance, which is necessary for calculating the outside heat transfer coefficient from the overall heat transfer coefficient, was evaluated by means of the theoretical correlation obtained by Kays [5].

3. NUMERICAL CALCULATION OF FIN EFFICIENCY

3.1. *Analytical model*

Quantitative information on the fin efficiency of the threads is necessary in order to discuss experimental results. Hence, the numerical calculation on the heat conduction inside the thread was carried out prior to the experiment. If we regard the thread as a fin, its minimum unit is as shown in Fig. 4(a). Therefore, we analyze the three-dimensional heat conduction equation for a straight cylinder of length l_f

$$\frac{\partial^2\Theta}{\partial R^2}+\frac{1}{R}\frac{\partial\Theta}{\partial R}+\frac{1}{R^2}\frac{\partial^2\Theta}{\partial\phi^2}+\frac{\partial^2\Theta}{\partial Z^2}=0 \qquad (1)$$

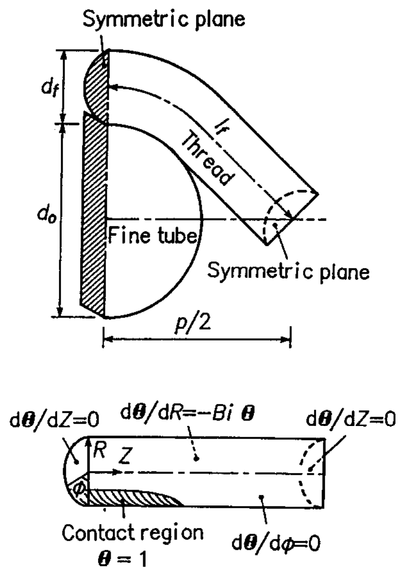

FIG. 4. Model for fin efficiency analysis: (a) minimum unit for thread and fine tube; (b) coordinate system and boundary conditions.

where the dimensionless variables are defined as follows:

$$\Theta=\frac{\theta-\theta_a}{\theta_w-\theta_a},\quad R=\frac{r}{r_f},\quad Z=\frac{z}{l_f}. \qquad (2)$$

The boundary conditions are shown in Fig. 4(b). In the analysis, the Biot number $Bi = \alpha_f d_f/\lambda_f$, which actually varies along the thread surface, was assumed to be constant, and was given by the average Nusselt number around a cylinder. The boundary condition ($\Theta = 1$) was applied in the region where the thread was in contact with the fine tube (the hatched area in Fig. 4(b)); it should be noted that the area changes depending on the solder thickness.

3.2. *Calculated results*

Figure 5 shows the fin efficiency for the transverse tube pitch $p = 2$ mm. In the figure, the Biot numbers for $Re_f = 300$ are indicated by arrows for the cases of stainless steel and copper. The fin efficiency is satis-

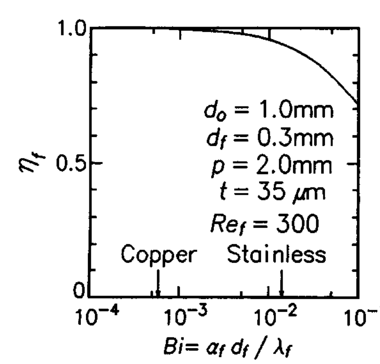

FIG. 5. Fin efficiency (effect of Biot number).

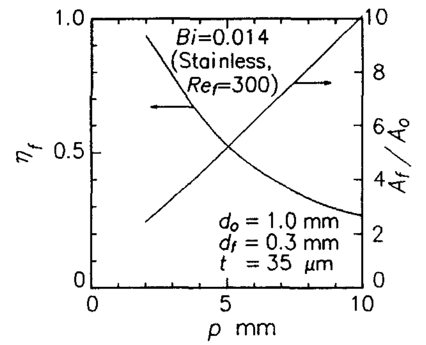

FIG. 6. Fin efficiency and area ratio vs transverse tube pitch.

factory for copper, while it shows a decreasing tendency for stainless steel.

To clarify this point, the fin efficiency for stainless steel is presented against the transverse tube pitch p in Fig. 6 (the area ratio A_f/A_o is also shown in Fig. 6). The fin efficiency decreases so markedly with increasing transverse tube pitch p that stainless steel is obviously not suitable for larger p.

The contribution of the threads to the total heat transfer can be evaluated if the heat transfer coefficient around the threads α_f is given, because the fin efficiency η_f and the area ratio A_f/A_o are already known. If we assume that α_o and α_f are given by the conventional correlation around a cylinder, the total heat transfer rate Q and thus the heat transfer coefficient α^* can be evaluated by the linear summation as follows:

$$\frac{Q}{A_o\Delta\theta} \equiv \alpha^* = \frac{\alpha_o A_o + \alpha_f \eta_f A_f}{A_o} = \alpha_o + \alpha_f \eta_f \frac{A_f}{A_o}. \quad (3)$$

Figure 7 shows the Nusselt number $Nu = \alpha^* d_o/\lambda_a$ calculated from equation (3) for $Re_o = 1000$, $\alpha_f = 2\alpha_o$. In the case of stainless steel, the product of η_f and A_f/A_o (both are shown in Fig. 6) becomes constant for $p > 4$ mm. As a result, Nu^* approaches an asymptotic value. Since the present analysis is based on several simplified assumptions, the results do not strictly correspond to the actual cases. It is important to note, however, that the possibility of an approximate sevenfold heat transfer enhancement is indicated, compared with the case without threads.

4. RESULTS AND DISCUSSION

4.1. *Basic heat transfer characteristics*

Prior to the discussion on the heat transfer performances for various geometric parameters such as d_f and p, we first show the basic heat transfer characteristics. Figure 8 compares the measured Nusselt number with that for a single circular cylinder, which is considered to be the most fundamental reference. The thread diameter is $d_f = 0.3$ mm, and the transverse tube pitch is $p = 4$ mm (the fine tube diameter d_o is 1 mm throughout the experiment). The Reynolds number Re_o is based on the approaching velocity of air u_o and the fine tube diameter d_o. The heat transfer coefficient α^* for the Nusselt number $Nu^* = \alpha^* d_o/\lambda_a$ was obtained using equation (3); thus the surface area of the threads is not taken into account. Compared with the solid line which denotes the empirical correlation for a single circular cylinder [6], the heat transfer coefficient increases approximately

six-fold for the case without solder;
nine-fold for the case with solder (stainless steel); and
14-fold for the case with solder (copper).

This heat transfer enhancement is ascribed to:

for the case without solder,
- (a) the effect of turbulence promotion by the threads and
- (b) the effect of air flow acceleration by blockage of threads;

for the case with solder,
- (a) the effect of turbulence promotion by the threads,
- (b) the effect of air flow acceleration by blockage of threads and
- (c) the fin effect.

From the experiment, the limiting value of the Nusselt

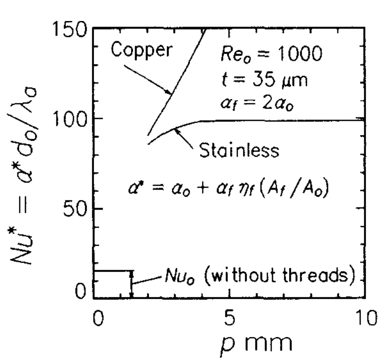

FIG. 7. Nusselt number due to fin effect vs pitch.

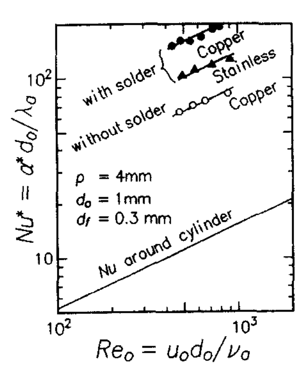

FIG. 8. Nusselt number (effects of solder plating and fin material).

number augmented only by the fin effect is not clear. Theoretically, however, it is approximately

six-fold for the case of stainless steel; and
nine-fold for the case of copper.

The measured increases in the Nusselt number for the case with solder (i.e., nine-fold for stainless steel and 14-fold for copper) are higher than these values. Thus, we confirm that the observed heat transfer enhancement is attributed to the nonlinear superposition of the effects mentioned above.

Here, we assess the experimental results in the light of the two principles for heat transfer enhancement described in the Introduction. First, judging from the six-fold increase in the Nusselt number for the case without solder, the woven threads have proved to be very effective as a turbulence promoter (the details of their function, especially relating to the eddy generating process, are now under study). Next, the use of the fine tubes has also been confirmed to be effective because the heat transfer coefficient around the fine tube reaches 3×10^3 W m^{-2} K^{-1} for the case of p = 2 mm, u_o = 13 m s^{-1} (Re_o = 850); furthermore, the heat transfer coefficient per unit projected area and the heat transfer coefficient per unit volume (i.e., total heat transfer rate divided by the projected area and the core thickness d_o+2d_f) reach the values of 5×10^3 W m^{-2} K^{-1} and 3×10^6 W m^{-3} K^{-1}, respectively.

4.2. *The effect of the thread diameter*

The following measurements were undertaken so as to obtain the optimum geometry of the fine-tube heat exchanger. Therefore, only the elements made of copper, which are always superior to those made of stainless steel, were tested.

Figures 9 and 10 show the results for three different thread diameters d_f. Although the fin surface area increases slightly with increasing d_f (the area for d_f = 0.8 mm is 18% larger than that for d_f = 0.3 mm), the Nusselt number Nu^* is largest for d_f = 0.3 mm. This is because the heat transfer coefficient around the threads increases with decreasing d_f. On the other hand, the pressure drop across the heat exchanger does not strongly depend on d_f. Thus, among the

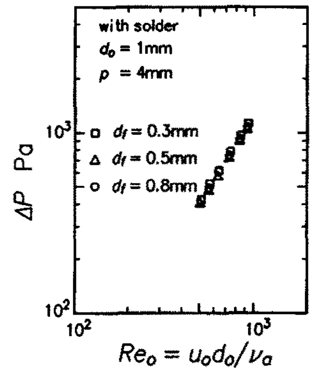

FIG. 10. Pressure drop (effect of thread diameter).

tested conditions, we find that the case for d_f = 0.3 mm is most desirable, and hence the thread diameter d_f is fixed at 0.3 mm in the following. Although logically the cases for d_f < 0.3 mm should also be tested, this was not done in the present study, because it was difficult to fabricate the element using such fine threads. Furthermore, we consider such fine threads useless as a turbulence promoter.

4.3. *The effect of the transverse tube pitch*

Figures 11 and 12 show the effect of the transverse tube pitch. In Fig. 11 it is interesting to note that, for the case without solder, the Nusselt number increases with decreasing p, whereas for the case with solder, the reverse tendency is observed. These tendencies are reasonably explained as follows. For the case without solder, the heat transfer for smaller p was effectively enhanced because of effects (a) and (b) described in 4.1. For the case with solder, however, the Nusselt number for larger p is increased mainly by the fin effect.

Unlike the heat transfer, the dependence of the pressure drop on the transverse tube pitch p is simple; the pressure drop increases with decreasing pitch irrespective of the cases with or without solder. This is because in the flow field there is no counterpart to the fin effect. Since the solder thickness t = 35 μm is not negligible compared with the radius of the thread, the

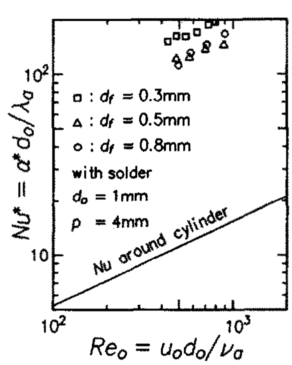

FIG. 9. Nusselt number (effect of thread diameter).

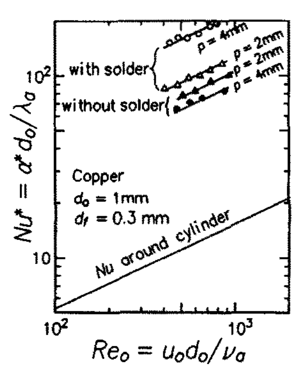

FIG. 11. Nusselt number (effect of transverse tube pitch).

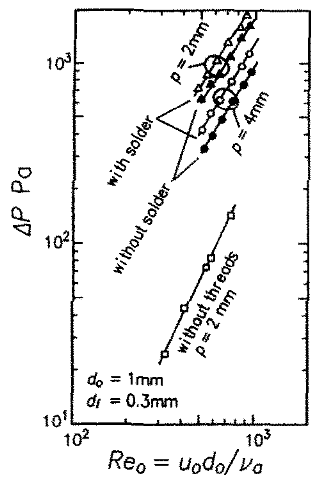

FIG. 12. Pressure drop (effect of transverse tube pitch).

pressure drop increases about 20% by using solder. The square symbol in Fig. 12 denotes the case without threads ($p = 2$ mm); owing to the threads, the pressure drop is one order of magnitude larger than that for bare tubes.

To get a complete picture of the heat exchanger performance, the pumping power must be taken into account. From Figs. 11 and 12, the heat exchanger with solder of $p = 2$ mm seems to be inadequate, because its Nusselt number is lower than that of $p = 4$ mm, but its pressure drop is relatively larger than that of the latter. In terms of the pumping power, however, it should be remembered that the heat transfer performance per unit projected area is important. Accordingly, the product of the heat transfer coefficient around the fine tube α^* and the number of fine tubes per unit width n $(=p^{-1})$ is displayed against the cubic root of the pumping power $E^{1/3}$ in Fig. 13. The solid line in Fig. 13 denotes the case for $p =$ 2 mm without threads (this is obtained on the basis

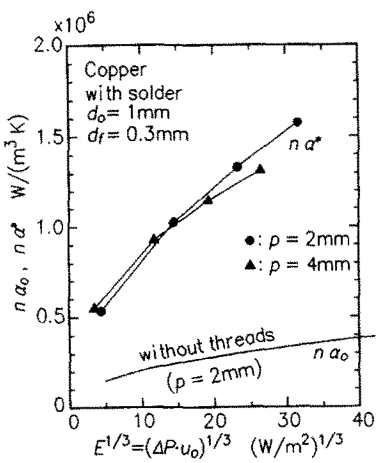

FIG. 13. Heat transfer performance vs pumping power.

of the pressure drop data shown in Fig. 12 and the correlation for a single cylinder with blockage correction). The values $n\alpha^*$ for both $p = 2$ mm and $p =$ 4 mm are much higher than $n\alpha_o$ for the case without threads. Consequently, it is demonstrated that the tested elements are effective as a new type of compact heat exchanger. Concerning the optimum geometry of the fine-tube heat exchangers, we cannot obtain a definite conclusion from Fig. 13, because the difference between $p = 2$ mm and 4 mm is within the measurement uncertainty.

5. CONCLUSIONS

A fine-tube heat exchanger based on new principles for heat transfer enhancement has been suggested and its basic performance has been investigated. As a result, a maximal 14-fold increase compared with the bare tube has been attained; the heat transfer coefficients, that per unit projected area and that per unit volume reach 3×10^3 W m^{-2} K^{-1}, 5×10^3 W m^{-2} K^{-1}, and 3×10^6 W m^{-3} K^{-1}, respectively. These extremely high values are due to the combination of the fine characteristic size and the coupled turbulence promoter with fin effect.

To obtain the geometry for optimum performance, experiments were conducted by systematically changing the geometric parameters. On the basis of the heat transfer and flow resistance data thus obtained, a total assessment has been made taking the pumping power into account.

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APPENDIX

Here, we discuss the heat transfer performance of the fine-tube heat exchanger in comparison with that of the louvered-fin heat exchanger. Since the heat exchangers differ greatly from each other in shape,† it is very difficult to compare their heat transfer performances fairly. Accordingly, it should be emphasized that the following comparison should only be regarded as a rough index by which the features of each heat exchanger are evaluated. As a representative example of the fine-tube heat exchanger we consider the case of $p =$ 2–4 mm, with solder, and copper construction, while for that of the louvered-fin heat exchanger, we choose the LIFT-3 as presented in ref. [4].

Provided that the approaching velocity u_o is assumed to be 13 m s^{-1} and the values for the fine-tube heat exchanger are normalized as unity, the values for the louvered-fin heat exchanger are as follows:

(1) the heat transfer coefficient per unit volume = 1/13–1/15;
(2) the pressure drop per unit core thickness = 1/50–1/90;
(3) the heat transfer performance for the same pumping power = 4–6.

Judging from this comparison, the merit of the fine-tube heat exchanger is its extremely high heat transfer density, while the louvered-fin heat exchanger has a better heat transfer performance in terms of pumping power.

† For example, the core thickness, which is used to calculate the quantities in (1) and (2), is 1.6 mm ($=d_o+2d_f$) for the fine-tube heat exchanger and 32 mm for the louvered-fin heat exchanger.

ECHANGEUR THERMIQUE A TUBES FINS ONDULES AVEC FILS TRESSES

Résumé—On suggère un nouveau type d'échangeur compact et les caractéristiques fondamentales du transfert thermique sont étudiées par les aspects expérimentaux. La conception est basée sur de nouveaux principes pour l'accroissement du transfert thermique, c'est-à-dire la réduction de la taille des surfaces de transfert et la disposition d'un couple de promoteurs de turbulence de façon à provoquer une forte modification de la structure de la turbulence. Pour satisfaire cela l'échangeur thermique est formé de tubes fins (diam. ext. ~ 1 mm) et de fils tressés ($d \sim 0,3$ mm); ces derniers jouent un double rôle de promoteur de turbulence et d'ailette. Le coefficient de transfert maximum obtenu dans les expériences est 4 fois plus grand qu'autour d'un cylindre sans fil. Grâce à la combinaison des fils tressés et de la très petite caractéristique de longueur, le coefficient de transfert par unité de surface projetée est environ 5×10^3 W m^{-2} K^{-1} et par unité de volume 3×10^6 W m^{-3} K^{-1}.

EIN WÄRMEAUSTAUSCHER AUS DÜNNEN DRAHTUMWOBENEN ROHREN

Zusammenfassung—In der vorliegenden Arbeit wird ein neuer Typ eines Kompaktwärmeaustauschers vorgeschlagen und sein grundlegendes Wärmeübergangsverhalten experimentell untersucht. Die Idee beruht auf neuen Prinzipien für eine Verbesserung des Wärmeübergangs: Reduzierung der Größe der wärmeübertragenden Oberflächen und drastische Änderung der Turbulenzstruktur durch Anordnung von Promotoren für die Turbulenz. Um derartigen Anforderungen entgegenzukommen, wird der Wärmeaustauscher aus dünnen Rohren (Außendurchmesser ungefähr 1 mm) aufgebaut, die von Draht (Durchmesser ungefähr 0,3 mm) umwoben sind. Letzterer wirkt gleichzeitig als Turbulenzerzeuger und als Rippe. Die maximalen Wärmeübergangskoeffizienten, welche bei den Versuchen ermittelt wurden, sind 14-fach größer als an einem Zylinder ohne das Drahtgespinst. Aufgrund der Kombination dieser Drähte mit einer sehr kleinen charakteristischen Abmessung erreicht der flächenbezogene Wärmeübergangskoeffizient ungefähr 5×10^3 W m^{-2} K^{-1} und der volumenbezogene Wärmeübergangskoeffizient 3×10^6 W m^{-3} K^{-1}.

ТЕПЛООБМЕННИК С ТРУБАМИ МАЛОГО ДИАМЕТРА ПРИ НАЛИЧИИ НАВИВКИ

Аннотация—Предложен новый тип компактного теплообменника. Проведено экспериментальное исследование его основных тепловых характеристик. Конструкция основана на новых принципах интенсификации теплообмена, а именно, на уменьшении площади поверхности теплопереноса и установке турбулизаторов с целью существенного изменения структуры турбулентности. Это достигается за счет того, что теплообменник состоит из труб малого диаметра (с внешним диаметром ~1 мм) и навивки из нитей (диаметром ~0,3 мм); при этом, нити играют роль турбулизаторов и ребер. Экспериментально полученный максимальный коэффициент теплопередачи в 4 раза больше, чем в случае цилиндра без навивки. Сочетание навивки и весьма малой характерной длины приводит к тому, что коэффициент теплопередачи на единицу проектируемой площади составляет около 5×10^3 Вт.м$^{-2}$ K^{-1}, а на единицу объема достигает 3×10^6 Вт$\cdot$м$^{-3}$ K^{-1}.