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Heat transfer enhancement accompanying pressure-loss reduction with winglet-type vortex generators for fin-tube heat exchangers

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Abstract

This paper proposes a novel technique that can augment heat transfer but nevertheless can reduce pressure-loss in a fin-tube heat exchanger with circular tubes in a relatively low Reynolds number flow, by deploying delta winglet-type vortex generators. The winglets are placed with a heretofore-unused orientation for the purpose of augmentation of heat transfer. This orientation is called as "common flow up" configuration. The proposed configuration causes significant separation delay, reduces form drag, and removes the zone of poor heat transfer from the near-wake of the tubes. This enhancement strategy has been successfully verified by experiments in the proposed configuration. In case of staggered tube banks, the heat transfer was augmented by 30% to 10%, and yet the pressure loss was reduced by 55% to 34% for the Reynolds number (based on two times channel height) ranging from 350 to 2100, when the present winglets were added. In case of in-line tube banks, these were found to be 20% to 10% augmentation, and 15% to 8% reduction, respectively. © 2002 Published by Elsevier Science Ltd.

1. Introduction

Fin-tube heat exchangers are employed in a wide variety of engineering applications, for example, in a geothermal, fossil, process plant and air-conditioning, etc. It is well known that most of the thermal resistance is on the fin-side in such heat exchangers. In order to enhance fin-side heat transfer between the fin and the gas flow, one way is to use vortex generators to produce longitudinal vortices inducing strong swirling motion that serves to bring about enhancement of heat transfer at a modest expense of the additional pressure-loss. Fiebig et al. [1] investigated experimentally the influence of delta winglet position on the fin surface having single tube, by punching a pair of vortex generators ahead and behind the tube, and found an increase of local heat transfer coefficient up to 100% and mean heat transfer coefficient up to 20% at the optimum winglet position, immediately behind tube, for the Reynolds number ranging from 2000 to 5000 based on fin pitch, H. They also observed a reduction of 10% in the pressure loss, and explained as the delayed separation on the tube due to longitudinal vortices generated by the vortex generator which introduces high momentum fluid into the wake region behind the tube. Fiebig and his group [2] extended experimentally the influence of the winglet vortex generators in fin-tube banks with three circular-tube rows in the streamwise direction. In their chosen geometry, the mean heat transfer coefficient increased by 55% to 65% for the inline tube arrangement, and by 9% for the staggered tube arrangement, and, the corresponding increases of pressure loss were 20% to 45%, and 3%, respectively. Biswas et al. [3] revealed numerically significant heat transfer enhancement immediately downstream of the tube in channel with built-in circular tube and a pair of delta winglet vortex generator. Their results show that longitudinal vortices generated by the winglets placed in the wake enhance the local heat transfer by 240%. A recent progress in vortex-induced air-side heat transfer enhancement for fin-tube heat exchangers was reviewed in detail [4–6].

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Nomenclature

A	area of heat transfer
B	channel width (fin width)
c_f	specific heat of fluid at constant pressure
D	diameter of the cylindrical tube
f	Fanning friction factor, $f = (2H/4X)\{(\Delta P/(\rho U_{in}^2/2)) - (K_c + K_e)\}$
H	channel height (fin pitch)
j	j-factor, $j = Nu/Re\,Pr^{1/3}$
h	heat transfer coefficient, height of the delta winglet
K_c	contraction coefficient
K_e	expansion coefficient
L	tube pitch, length of flow channel
l	base length of the delta winglet
m_f	mass of fluid contained by flow channel
$\dot{m}_f$	mass flow rate of fluid
NTU	number of transfer unit, $NTU = hA/\dot{m}_f c_f$
Nu	Nusselt number, $Nu = h_m \cdot 2H/\lambda$
ΔP	pressure loss of the test core
Pr	Prandtl number
Re	Reynolds number, $Re = U_{in} \cdot 2H/\nu$
T	temperature
U	mean flow velocity
X	axial dimension of coordinates
Y	spanwise dimension of coordinates

Greek symbols

β	attack angle of vortex generator
λ	thermal conductivity of air
ν	kinematic viscosity
θ	time
ρ	density of air
τ	time constant

Subscripts

c	minimum cross-section of flow channel
Go	fin-tube banks without vortex generator
in	inlet
m	average

In general, the delta winglet-type vortex generators used to be mounted on the flat surface with "common flow down" configuration. For such configuration, the transverse distance between the leading edges of the winglet pair is less than the transverse distance between the trailing edges of the winglet pair, as shown in Fig. 1(a). Even this configuration cannot enhance heat transfer without additional amount of pressure drop, though its amount is less than other conventional devices for enhancing heat transfer. Recently the design value of Reynolds number for a fin-tube heat exchanger decreases more and more to meet such demands as compactness, fan power saving and quietness. A new strategy for heat transfer enhancement with less pressure-loss penalty is eagerly anticipated in a relatively low Reynolds number flow.

The main aim of the present research is to propose a novel strategy that can augment heat transfer but nevertheless can reduce pressure-loss in a fin-tube heat exchanger with circular tubes in a relatively low Reynolds number flow, by deploying delta winglet-type vortex generators. The proposed strategy is to place the delta winglet pair with "common flow up" configuration on the fin surface, as shown in Fig. 1(b). With this configuration, the winglet pair can create constricted passages in aft region of the tube, which brings about separation delay. The fluid is accelerated in the constricted passages and as a consequence the point of separation travels downstream. Narrowing of the wake and suppression of vortex shedding are the obvious outcome of such a configuration which reduce form drag. Since the fluid is accelerated in this passage, the zone of poor heat transfer on the fin surface is also removed from the near-wake of the tube. In case of a low Reynolds number flow in absence of any vortex generators, the poor heat transfer zone is created widely on the fin surface in the

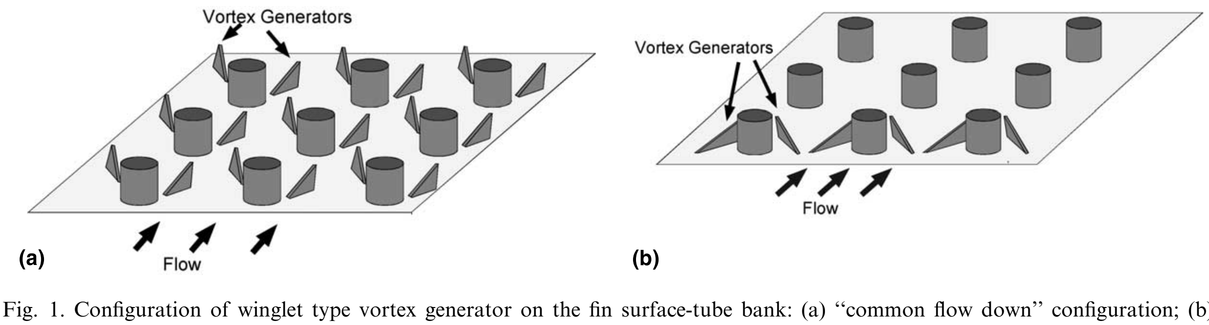

Fig. 1. Configuration of winglet type vortex generator on the fin surface-tube bank: (a) "common flow down" configuration; (b) "common flow up" configuration.

near-wake of the tube and may extend far downstream, even to the next row of the tube bank. Hence it is expected that the present strategy may be more effective for a lower Reynolds number flow.

2. Experimental method and procedure

2.1. Experimental apparatus

The present experimental apparatus for a modified single-blow method was designed and built, referring to Mochizuki et al. [7]. The experiments are performed in a small wind tunnel of open-circuit, with a vertical test section of dimensions of $150 \times 100 \times 300$ mm^3 (width $\times$ depth $\times$ length), as shown in Fig. 2. The blower fan is driven by a variable-speed, 1.5 kW electric motor to control the air velocity. The mean flow velocity in the test section can be varied from 0.5 to 3.5 m/s. A heating screen is made of stainless-steel ribbon heated directly by ohmic Joule heating, and is uniformly spread over an entire cross-section at the inlet of the test section so that it can heat the flow quickly and uniformly. The heating rate is to be controlled by a slide regulator. A bulk temperature of the flow at the inlet or outlet is to be measured directly by a special sensor without any cross-sectional integration of the flow temperature. The sensor with a quick response is made of a single Platinum wire of 0.03 mm in diameter that is woven diagonally and is spread over an entire cross-section at the inlet or outlet. Temperature is derived from its electric resistance.

2.2. Test-cores

The geometrical parameters of the test-core simulate fin-tube heat exchangers such as air-cooled condensers used in binary-cycle geothermal power plants. The test-cores of fin-tube banks consist of 16 parallel plates representing the fins and three rows of the circular tubes with inline or staggered arrangements. The geometric arrangements of the test-cores are shown in Fig. 3 and listed in Table 1. The fin pitch H is 5.6 mm. Both streamwise and spanwise pitches of tube banks are equally set to 75 mm. The vortex generators consist of delta winglet pairs made of 0.3 mm thick Bakelite. The present configuration of the delta winglet pair is called as common flow up configuration in which the spanwise distance between the leading edges of a winglet pair is wider than the one between the trailing edges, as shown in Fig. 3(a). The base length l and height h of the winglet are 30 mm and 5 mm, respectively. In order to illustrate the excellent performance of the configuration proposed in the present study, the configuration with all three rows of winglet pairs shown in Fig. 3(b) and Table 1 was also examined experimentally in the present study. This common flow down configuration of winglet pairs with height/base length aspect ratio, $h/l = 1/2$, was proposed by Fiebig et al. [2] that gave the best performance in their study.

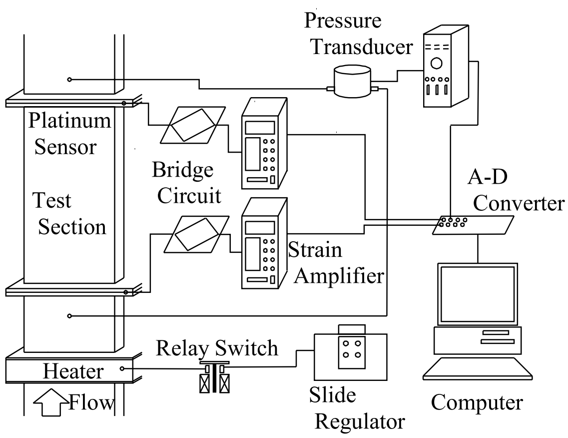

Fig. 2. Schematic diagram of wind tunnel for transient method.

2.3. Data reduction

The experimental data are represented in terms of the Colburn factor, j, and the Fanning friction factor, f, as function of the Reynolds number, Re, as follows:

$$j = \frac{Nu}{Re \cdot Pr^{1/3}}, \quad Nu = \frac{h_m \cdot 2H}{\lambda}, \quad Re = \frac{U_{in} \cdot 2H}{\nu},$$
$$f = \frac{2H}{4X}\left\{\frac{\Delta P}{\rho U_{in}^2/2} - (K_c + K_e)\right\}. \tag{1}$$

For determining heat transfer performance of heat exchanger units or surface, Liang and Yang [8] developed the energy equations between the heat transfer surface and the fluid that vary with time and position along with flow passage. They determined experimentally the boundary condition that describes the time-wise change of the inlet fluid temperature, as follows:

$$T_f^*(\theta^*, 0) = 1 - e^{-\theta^*/\tau^*}, \tag{2}$$

T_f^* is the dimensionless fluid temperature and the dimensionless time, θ^*, is defined as the ratio of the physical time, θ, to the time-constant, $\tau_{sys} = (m_s c_s/hA)$ of the solid–fluid system. Where m_s, c_s, h and A are mass of solid, specific heat of solid, average heat transfer coefficient and heat transfer area, respectively. τ^* is defined as τ_{in}/τ_{sys}, in which τ_{in} is the time-constant of the measured inlet fluid temperature to be determined experimentally. Employing Laplace transform method, they obtained the following analytical fluid outlet temperature expressions in two time-domains as

(i) when $\theta^* < t^*$, or equivalently $\theta < L/U_c$,

$$\bar{T}_f^*(\theta^*, NTU) = 0, \tag{3}$$

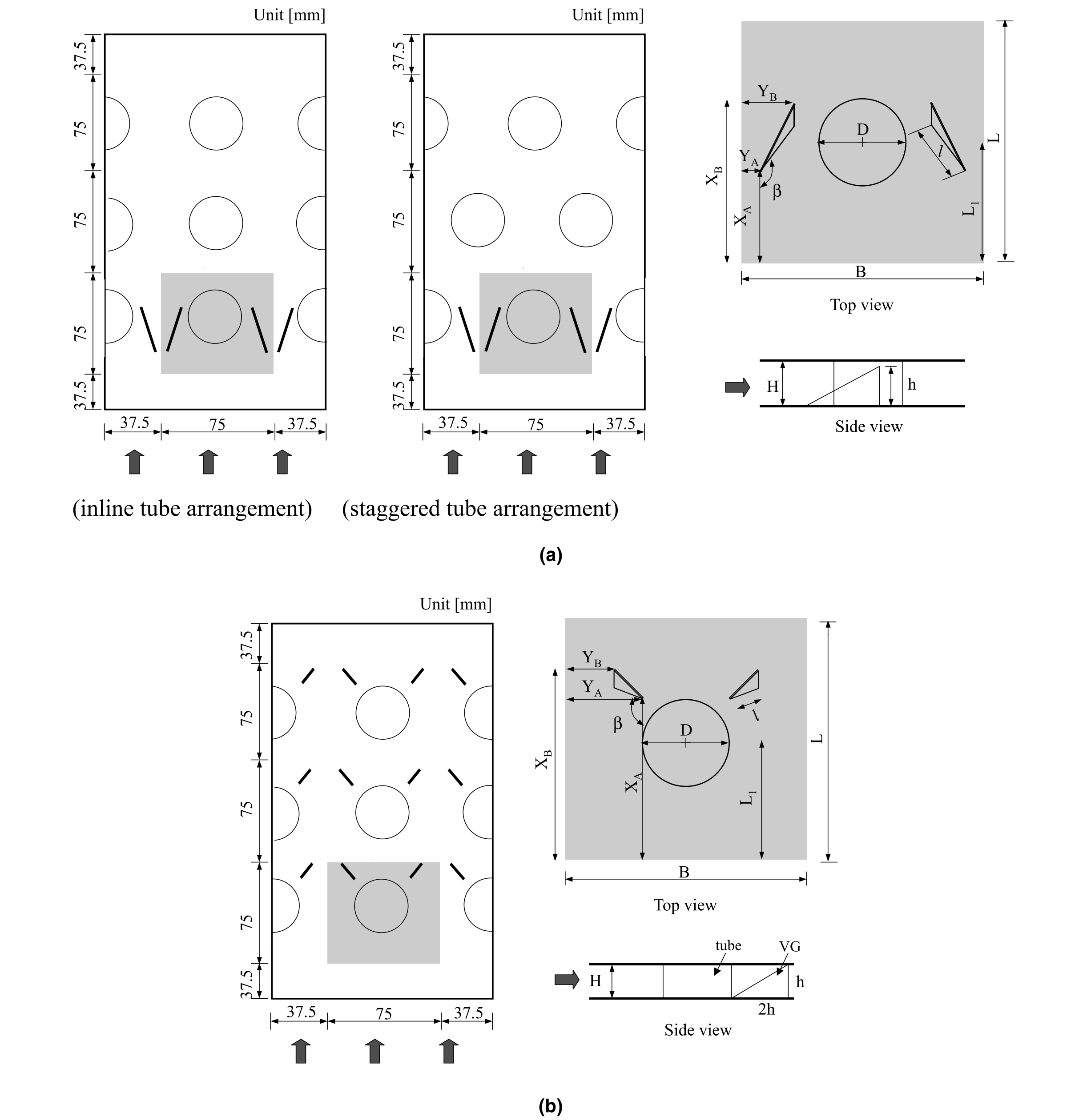

Fig. 3. Geometric arrangements of test core with three tube-rows and vortex generators: (a) the present proposed configuration of winglet; (b) the configuration of winglet proposed by a previous study [2].

(ii) when $\theta^* \geqslant t^*$, i.e., $\theta \geqslant L/U_c$,

$$\bar{T}_f^*(\theta^*, NTU) = \frac{1}{\tau^*}\int_{t^*}^{\theta^*} e^{-(\theta^*-\eta)/\tau^* - b_2 t^*}\left\{e^{-(\eta-t^*)}J_0\left[2\sqrt{b_2 t^*(\eta - t^*)}\right] + \psi_2(\eta, NTU)\right\} d\eta, \tag{4}$$

where J_0 is the zero-order Bessel function of the first kind,

$$t^* = NTU/b_1,$$
$$\psi_2(\eta, NTU) = \int_0^{\eta - t^*} e^{-\xi} J_0[2(b_2 t^* \xi)^{1/2}]\, d\xi, \tag{5}$$

t^* is dimensionless time defined as NTU/b_1, and b_1 is $U(m_s c_s)/(\dot{m}_f c_f L)$. U, $(m_s c_s)$, $\dot{m}_f$, c_f, and L are mean velocity of fluid in test core, heat capacity of solid, mass flow rate of fluid, specific heat of fluid at constant pressure, and length of flow channel (heat transfer sur-

Table 1
Summary of geometrical arrangements of test-cores and vortex generators

Variables	Arrangement proposed by the present study[a,b]	Arrangement proposed by Fiebig et al. [2][a]
B/H	13.39	13.39
L/H	13.39	13.39
D/H	5.36	5.36
L_1/D	1.25	1.25
D/B	0.40	0.40
X_A/H	2.44	9.38
X_B/H	7.61	10.79
Y_A/H	1.19	4.02
Y_B/H	2.57	2.60
h/H	0.9	1.0
l/D	1.0	0.37
β	165°	135°

[a] In-line tube arrangement.
[b] Staggered tube arrangement.

face), respectively. b_2 is heat capacity ratio between the solid and the fluid defined as $\{(m_s c_s)/(m_f c_f)\}$. U_c is mean velocity based on minimum cross-sectional area, A_c, ζ, and η are dummy variable.

Firstly, a measured inlet air temperature is fitted with Eq. (2) at three periods of time, θ_1, $\theta_2 = 2\theta_1$, $\theta_3 = 3\theta_1$, to determine the values of the final steady inlet temperature, T_{final}, and τ_{in}. The inlet temperature is recorded with a time step of 0.02 s for 1500 steps corresponding to 30 s after it starts rising. During the early period of time steps it cannot follow Eq. (2) but shows a response with a time lag, θ_0. In order to avoid this period and find the appropriate period automatically, the following procedure is taken.

The time, θ_1, is assumed to be 3 s to first approximation and is increased time-step by time-step until the measured inlet temperatures can be fitted well with Eq. (2) without introducing the time lag, θ_0, within one time step. Hence the measured inlet temperatures, T_{f1}, T_{f2}, and T_{f3}, at $\theta_1, \theta_2 = 2\theta_1$, and $\theta_3 = 3\theta_1$, respectively, satisfy Eq. (2) and give the final temperature T_{final}, and the time constant τ_{in} by the following equations as

$$\begin{aligned} T_{\text{final}} &= T_{f1} + (T_{\text{final}} - T_{f2})\left(\frac{T_{\text{final}} - T_{f2}}{T_{\text{final}} - T_{f3}}\right)^{(\theta_2-\theta_1)/(\theta_3-\theta_2)}, \\ \tau_{\text{in}} &= (\theta_2 - \theta_1)\Big/\left\{\ln\frac{T_{\text{final}} - T_{f1}}{T_{\text{final}} - T_{f2}}\right\}. \end{aligned} \tag{6}$$

The obtained value of τ_{in} ranges from 6.6 to 23 s, and the values of $\theta_1/\tau_{\text{in}}$ were found to be 0.16–0.53 for the present experiments.

By substituting the values of τ_{in}, T_{final}, geometrical and physical properties of flow channel and heat transfer surfaces, and an assumed value of heat transfer coefficient *h,* into the theoretical equation (4), the exit fluid temperature, $T_{f,\text{thnal-at exit}}$, is calculated and is then compared with the measured value of fluid temperature at the exit, $T_{f,\text{expt--at exit}}$. If the difference between $T_{f,\text{thit-at exit}}$ and $T_{f,\text{expt--at exit}}$ is within an acceptable degree of accuracy less than 0.1%, then the assumed value of h is considered to be correct. However, if the theory fails to agree with the measured response, a new value of h is picked and the procedure is repeated until the correct value of h is found. These calculations are performed to get the average value of the heat transfer coefficient, h, at 10 points of time in the neighborhood of τ_{in} for each experiment. The assumption that a temperature gradient is negligible inside the fin is acceptable since Biot number, $h_m\delta/2\lambda_f$, based on the measured, h_m, and fin thickness, δ, is order of 10^{-6}.

3. Experimental results and discussion

The Reynolds number, $Re = (U_{\text{in}}2H)/v$, based on the hydraulic diameter of the test-core inlet varies from 350 to 2400. The average heat transfer coefficient of the test-core is defined by using its heat transfer area excluding the surface area of tube and vortex generators. Fig. 4 shows the heat transfer enhancement rate of fin surface in in-line tube-banks with vortex generators with common flow down configuration proposed by Fiebig et al. [2] (see Fig. 3(b)), j/j_{Go}, where the subscript Go denotes inline tube-banks without vortex generator, as well as the pressure-loss penalty, f/f_{Go}, in a wide range of Reynolds number. The vortex generators bring 10% to 25% increase in the heat transfer enhancement and also 20% to 35% increase in the pressure-loss penalty, in

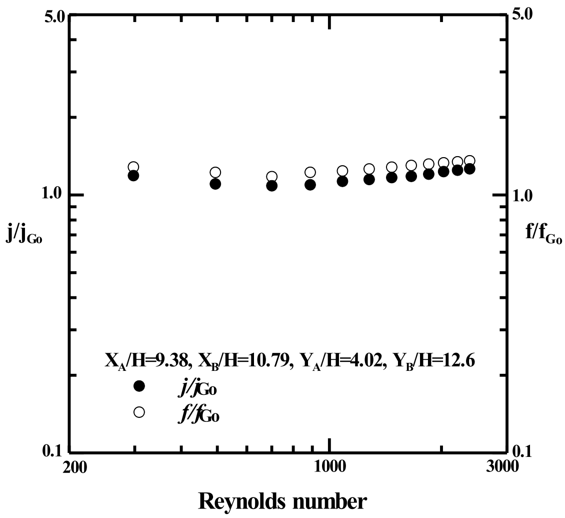

Fig. 4. The comparison of j/j_{Go} and f/f_{Go} with respect to Reynolds number (for the configuration of winglet illustrated in Fig. 3(b)).

comparison with tube-banks without vortex generators. Although it is attractive to the heat transfer enhancement, this configuration brings about the pressure loss penalty more than the heat transfer enhancement, as shown in Fig. 4. In addition, this configuration is found to be not so effective in heat transfer enhancement in a low Reynolds number as in a high Reynolds number flow studied by Fiebig et al. [2]. It should be noted that many heat exchangers including geothermal air-cooled condensers operate at such a low Reynolds number as below 1000. It suggests that heat transfer enhancement in a laminar flow of low Reynolds number necessitates an innovative concept being entirely different from the conventional concept effective to a high Reynolds number flow. In order to understand the physics, our collaborators, O'Brien and Sohal [9] investigated the local heat transfer in a single passage of narrow rectangular duct fitted with a circular tube and/or a delta winglet pair using imaging infrared camera. The location and aspect ratio of the winglets were the same as the present study shown in Fig. 3(b). As for the spanwise variation, they showed that the vortex generators reduced the size of wake region behind tube, but the heat transfer directly downstream of the tube were rather slightly reduced for the fin-tube with winglets, being compared to the one without winglets. According to their study, heat transfer is strongly enhanced downstream of winglet pair but indistinctly in the wake region behind the tube.

As mentioned previously, the present concept is to induce flow acceleration as well as vortex and turbulence, creating constricted passage between the tube and the winglets with common flow up configuration (see Fig. 3(a)). This flow acceleration brings about separation delay from the tube, reduces form drag across the tube and finally it removes the zone of poor heat transfer from the wake zone of the tube, without an excessive amount of pressure loss penalty. As shown in Fig. 5, the delta winglets with common flow up configuration in a fin-tube bank in an in-line tube arrangement successfully increase the average heat transfer by 10% to 20%, and simultaneously decrease the pressure drop by 8% to 15%. It is worth noting that such a remarkable performance can be brought about by a single front row of the winglet pairs in three rows of tube banks, comparing that the performance shown in Fig. 4 is achieved by means of all three rows of winglet pairs.

A much better performance has been achieved in a staggered tube arrangement with the same configuration of winglet pairs as in the in-line arrangement, as shown in Fig. 6. The pressure-loss reduction $(1 - f/f_{\mathrm{Go}})$ of 34% to 55% is achieved together with the heat transfer enhancement (j/j_{Go}) of 10% to 30%. It is a favorable performance that the pressure-loss reduction gets better with a lower Reynolds number. For a Reynolds number of 350, the pressure-loss reduction of 55% is achieved together with the heat transfer enhancement of 30%. This may be explained as follows. The nozzle-like flow passages created by the delta winglet pair and the aft region of the tube promote acceleration to bring about separation delay and thereby remove the zone of poor heat transfer from the near wake. This flow structure has been confirmed by visualizing the flow by means of "Particle Image Velocimetry" [10]. In the absence of any

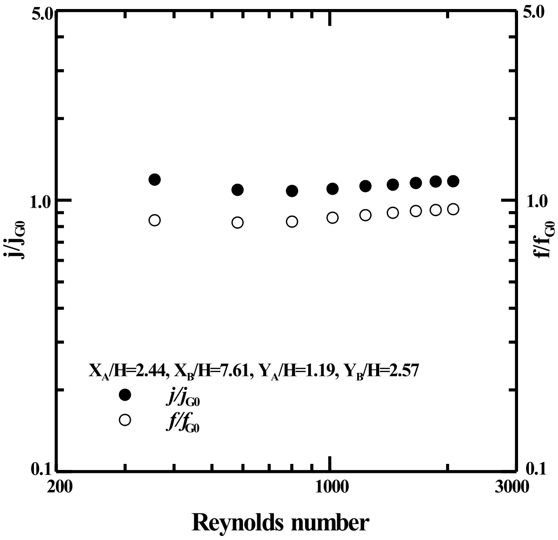

Fig. 5. The comparison of j/j_{Go} and f/f_{Go} with respect to Reynolds number (for the configuration of winglet in in-line tube arrangement illustrated in Fig. 3(a)).

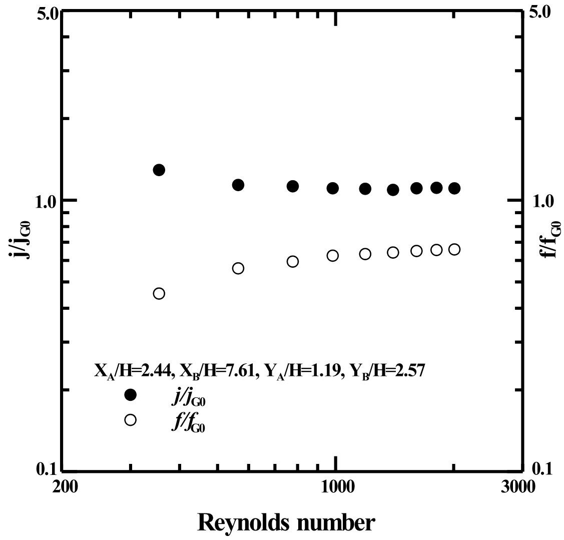

Fig. 6. The comparison of j/j_{Go} and f/f_{Go} with respect to Reynolds number (for the configuration of winglet in staggered arrangement illustrated in Fig. 3(a)).

winglet, a wake behind the tube gets weaker in vortex motion and develops a wider, longer zone of poor heat transfer as the Reynolds number decreases. When the wake can travel down to the next row, its length in the staggered arrangement gets twice as long as in the in-line one. Hence the delta winglet pair with the present configuration improves the performance for a low Reynolds number more for the staggered tube bank than for the in-line one.

4. Concluding remarks

An experimental study has been performed to obtain heat transfer and pressure loss in a test section, simulated to a fin-and-tube heat exchanger, with in-line or staggered tube banks with delta winglet vortex generators of various configurations. The present experiment verifies that the present novel technique combining a circular tube with the delta winglet pair of the "common flow up" configuration enables us to remarkably enhance the heat transfer together with a significant amount of pressure loss reduction. It is entirely different from a conventional heat transfer enhancement device. The nozzle-like flow passages created by the delta winglet pair and the aft region of the circular tube promote acceleration to bring about a separation delay and form drag reduction of the tube, and remove the zone of poor heat transfer from the wake.

1. In case of staggered tube banks, the heat transfer was augmented by 30% to 10%, with the winglet pairs of the present configuration, and yet the pressure loss was reduced by 55% to 34%, for the Reynolds number (based on two times the channel height) ranging from 350 to 2100.
2. In case of in-line tube banks, the heat transfer was augmented by 20% to 10% together with the pressure loss reduction of 15% to 8% in the same range of Reynolds number above.
3. Applying the configuration proposed by Fiebig et al. [2], the heat transfer was augmented by 25% to 10% but the pressure loss was also increased by 35% to 20% in the same Reynolds number range.

Acknowledgements

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