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Technical Note

Heat transfer and pressure loss penalty for the number of tube rows of staggered finned-tube bundles with a single transverse row of winglets

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Abstract

The objective of this research is to investigate the heat transfer and pressure loss penalty for various numbers of transverse rows in staggered finned-tube bundles with a single transverse row of the winglet pairs beside the front row of the tube bundles. Experiments were performed for two, three, four and five rows of staggered tube bundles. The pairs of winglets were placed with a heretofore-unused orientation for the purpose of augmentation of heat transfer and reduction of pressure loss penalty. This orientation is called as "common flow up" configuration. For three rows of tubes with a single transverse row of winglets beside the front row of the tubes, the heat transfer was augmented by 30–10%, and yet the pressure loss was reduced by 55–34% with the increase of the Reynolds number (based on two times channel height) from 350 to 2100. The reduction of the pressure loss penalty for three rows of tube bundles is the largest in comparison with the other numbers of rows.

1. Introduction

It is well known that the horseshoe vortices formed around tubes on fins enhance the heat transfer of fins, causing a large pressure loss due to form-drag in fin-tube heat exchangers [1,2]. The present authors proposed a novel strategy that can augment heat transfer but nevertheless can reduce pressure loss in a relatively low Reynolds number flow, by deploying delta winglet pair with "common flow up" configuration on the fin surface [3,4]. 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 in the downstream. Narrowing of the wake and suppression of vortex shedding are the obvious outcome of such a configuration to 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 generator, the poor heat transfer zone is created widely on the fin surface in the near-wake of the tube and may extend far downstream, even to the next row of the tube bundle. Hence it is expected that the present strategy may be more effective for a lower Reynolds number flow. According to their result, in case of staggered tube bundles with three rows of tubes, the heat transfer was augmented by 30–10%, and yet the pressure loss was reduced by 55–34% with the increase of Reynolds number (based on two times the channel height) from 350 to 2100, when the winglets proposed by the present authors were added only in the front first row of tube bundles.

The purpose of the present study is to extend the experiments on the heat transfer and pressure loss of

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Nomenclature

D	diameter of the cylindrical tube
f	Fanning friction factor, $f = \frac{2H}{4L}\left\{\frac{\Delta P}{\rho U_{in}^2/2} - (K_c + K_e)\right\}$
H	channel height (fin pitch)
h	heat transfer coefficient, height of the delta winglet
j	j-factor, $j = Nu/RePr^{1/3}$
K_c	contraction coefficient
K_e	expansion coefficient
L	length of flow channel
l	base length of the delta winglet
Nu	Nusselt number, $Nu = (h_m \times 2H)/\lambda$
ΔP	pressure loss of the test-core
Pr	Prandtl number
Re	Reynolds number, $Re = (U_{in} \times 2H)/\nu$
s	spanwise gap between the trailing edge of winglet and the surface of tube
U_{in}	mean flow velocity at inlet
Greek symbols	
α	attack angle of vortex generator
β	central angle from the front stagnation point of tube
λ	thermal conductivity of air
ν	kinematic viscosity
ρ	density of air
Subscripts	
G0	fin-tube bundle without vortex generator
m	average

staggered finned-tube bundles. In the present study, the effects of the number of transverse tube rows are investigated for two, four and five rows of staggered tube bundles with a single row of the winglet pairs beside the front row of the tube bundles, comparing with the result of three rows of staggered tube bundles acquired by the previous study [3].

2. Experimental method

2.1. Test-cores

The geometrical parameters for test-core are simulated to a fin-tube heat exchanger such as air-cooled condenser for binary-cycle geothermal power plant. The test-core of fin-tube bundles consists of 16 parallel plates simulating the fins and of the circular tubes with a staggered arrangement, as shown in Fig. 1(a). The test-core is tested at a vertical test section of dimensions of 150 mm × 100 mm × 525 mm (width × depth × length), to measure the overall heat transfer by means of a transient method, which is described in detail in Ref. [3]. The tube and fin are made of hollow acrylic tube of 30 mm outer diameter with 9.6 mm thickness and of aluminum flat plate of 0.3 mm thickness, respectively. The channel height (fin pitch), H, is 5.6 mm. Both streamwise and spanwise pitches of the tubes are equally set to 75 mm with a staggered arrangement, as shown in Fig. 1(a). The vortex generator consists of delta (triangular) winglet pair of 0.3 mm thick bakelite mounted on the aluminum channel wall (fin surface). As shown in Fig. 1(b), the base length of winglet, l, its height, h, a spanwise gap between the trailing edge of winglet and the surface of tube, s, an attack angle of winglet, α, the central angle from the front stagnation point of tube, β, are 30, 5, 9 mm, 15° and 110°, respectively. All geometries such as tube size, tube-pitches and fin height were fixed except the streamwise fin-length, L. The winglet stands upright on the fin plate.

2.2. Data reduction

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

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

where H, h_m, Nu, Pr, Re, λ, L and U_{in} are fin pitch, heat transfer coefficient, Nusselt number, Prandtl number, Reynolds number, thermal conductivity of air, test-core length, and mean flow velocity at inlet, respectively. The contraction and expansion coefficients for entrance and exit pressure loss, K_c and K_e, were determined from Ref. [5], as 0.42 and −0.35, respectively. The Reynolds number, $Re = (U_{in}2H)/\nu$, based on the hydraulic diameter of the test-core inlet varies from 300 to 2600. The average heat transfer coefficient of the test-core is defined by using, as heat transfer area, the surface area of aluminum fin plates excluding the base-areas of the tubes and vortex generators made of heat-resistant materials attached on the fin surface.

The uncertainty analysis was made by the method of Kline and McClintock [6]. Assuming a length-scale and physical properties (air) uncertainty of ±1%, the uncertainty of j factor was estimated to be 5.5% (bias error 5.5%, precision error 0.03%) at a lower Reynolds num-

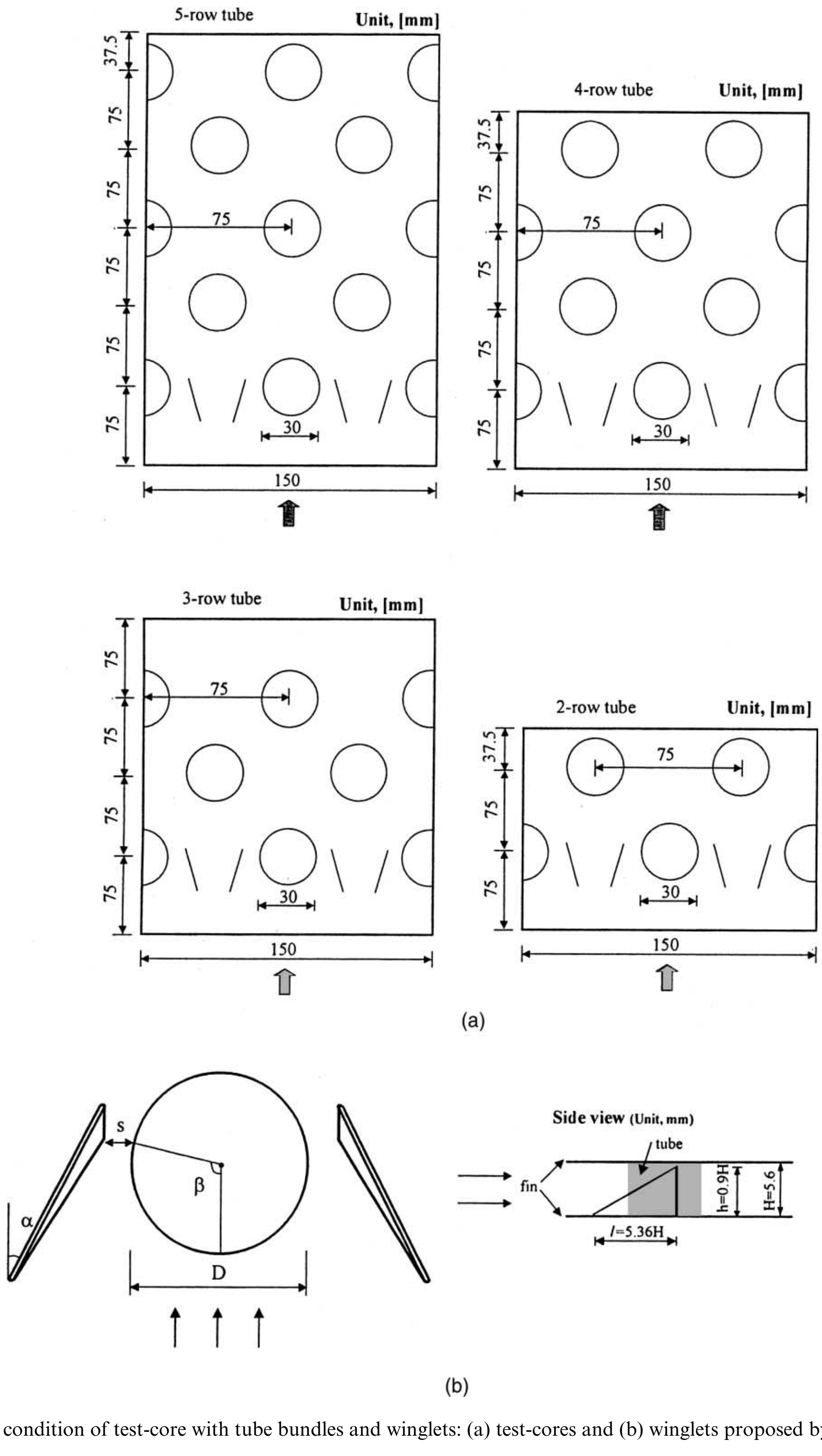

Fig. 1. Geometric condition of test-core with tube bundles and winglets: (a) test-cores and (b) winglets proposed by the present study.

ber flow and 4.0% (bias error 4.0%, precision error 0.03%) at a higher Reynolds number flow, and the uncertainty of f factor was 5.0% (bias error 5.0%, precision error 0.7%), respectively. The uncertainty of $(j - j_{G0})$ or

$(f - f_{G0})$ is equal to the precision error, since the bias error is cancelled out. Then the uncertainties of $(j - j_0)/(j_{G0} - j_0)$ and $(f - f_0)/(f_{G0} - f_0)$ are 0.04% and 1.0%, respectively, where the subscript of 0 denotes a plane channel without built-in tube.

3. Experimental results and discussion

Fig. 2 shows the heat transfer for various numbers of tube rows without winglet and for plane straight fin (channel), as *j*-factor with respect to Reynolds number. The *j*-factor of plane straight fin (channel) is compared with the empirical correlation of plane duct for laminar flow [7]. They show a good agreement with each other in a low Reynolds number flow, but deviate gradually from each other with the increase of Reynolds number. The heat transfer of fin having tubes becomes increasingly higher for a higher Reynolds number than the plane straight fin (channel) without tube, but becomes nearly the same for the Reynolds number below 300. It may be conjectured that the heat transfer enhancement caused by horseshoe vortex around the front root of tube cannot offset the heat transfer reduction caused by the poor heat transfer of the wake region behind the tube, in a low range of Reynolds number. For the higher range of Reynolds number, the heat transfer enhancement becomes increasingly higher than without tube. It may be explained as the results of both the stronger horseshoe vortex around the tube and the smaller size of wake region behind the tube due to the higher turbulence intensity, with the increase of Reynolds number. Heat transfer for various numbers of transverse tube rows is nearly constant up to five tube rows and is almost independent of the number of the tube rows in the entire range of Reynolds numbers. This means that the heat transfer rate per unit area for a various number of the tube rows is nearly the same, which may be explained as follows. In case of a staggered tube arrangement, the strength of horseshoe vortices generated at every tube row is nearly the same between the first and second rows, but weakens only slightly at the downstream rows such as the third, fourth and fifth rows. On the contrary, the flow turbulence increases at the downstream rows, which may compensate for weakening of horseshoe vortex. This may be the reason why the heat transfer rate per unit area can be nearly constant up to the fifth row of tubes, as shown in Fig. 2.

Fig. 3 shows pressure loss for the number of tube rows and for plane straight fin (channel) mentioned in Fig. 2, as *f*-factor with respect to Reynolds number. The *f*-factor of plane straight fin (channel) is compared with the correlation proposed by Shah and London [8]. They deviate slightly from each other with the increase of Reynolds number, which is similar to the *j*-factor as shown in Fig. 2. The pressure loss of fin having tubes becomes much larger for a higher Reynolds number than the plane straight fin (channel) without tube, because of large form drag brought about by the tubes. For a various number of tube rows, the pressure loss for three rows is found to be the lowest in the entire Reynolds numbers. At the lower Reynolds numbers below 520, the pressure loss decreases in the following order as

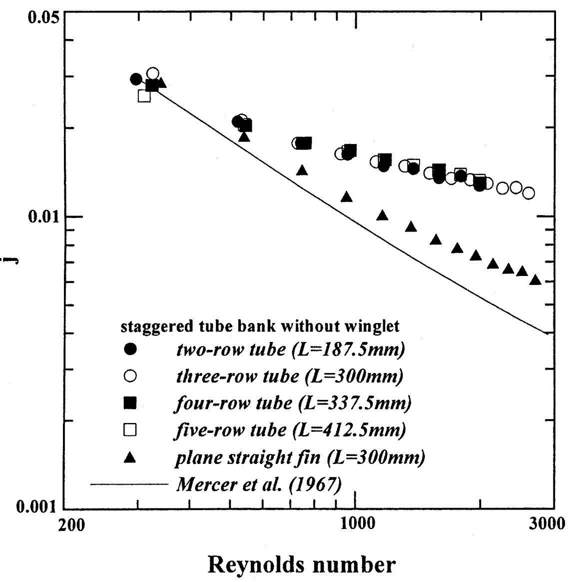

Fig. 2. Heat transfer for the number of tube rows without winglet and for plane straight fin (channel).

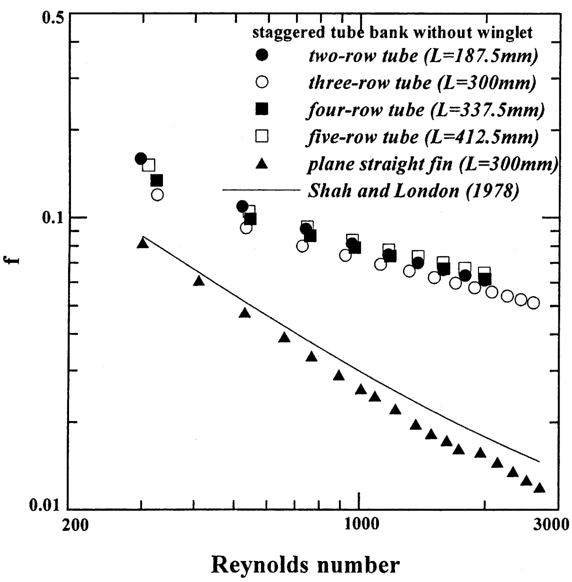

Fig. 3. Pressure loss for the number of tube rows without winglet and for plane straight fin (channel).

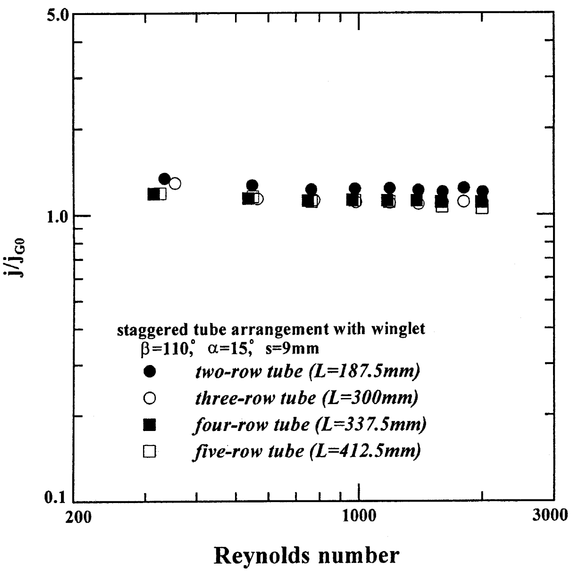

Fig. 4. Heat transfer enhancement for the number of tube rows with a single row of winglet pair.

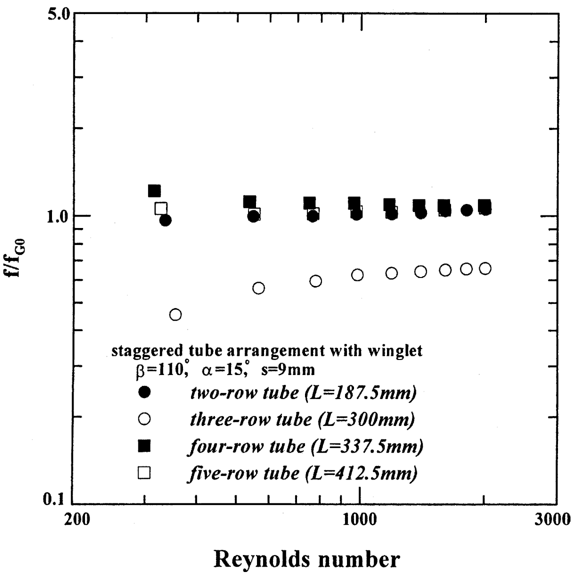

Fig. 5. Pressure loss penalty for the number of tube rows with a single row of winglet pair.

two, five, four and three rows. At the higher Reynolds numbers above 520, the pressure loss for the two rows is equal to that for the four rows, and for the five rows increases slightly higher than for the other numbers of rows.

For a single row of built-in winglet-pairs, Fig. 4 shows that the ratio of heat transfer enhancement, j/j_{G0}, is almost independent of the number of tube rows except for two rows in the entire range of the Reynolds numbers, showing the nearly constant increase of 10–30%. The heat transfer for the two rows is the highest and increases by 20–35%.

Fig. 5 shows that the pressure loss penalty, f/f_{G0}, for two-row, four-row and five-row tube bundles is 1.0, 1.22–1.1 and 1.05, respectively, in Reynolds numbers below the value of 1000, and is nearly equal to each other in Reynolds numbers above the value of 1000. For the three rows, the pressure loss penalty, f/f_{G0}, decreases from 0.66 to 0.45 with the decrease of the Reynolds numbers from 2100 to 350. It means that the pressure loss reduction of 34–55% has been successfully achieved. The existence of the optimal number of tube rows may be explained as follows. The flow is accelerated at the constricted passages between the tube and a pair of winglets, which brings about the separation delay and, as a consequence, the form drag reduction. This flow acceleration reduces the dead-wake zone and restricts its development, which brings about the reduction of the form drag of the downstream rows. It should be noted that the present study has dealt with the effects resulting from a single row of winglet, and the pressure loss penalty is closely related to the variation of longitudinal velocity. Therefore, in the staggered tube bundles, flow acceleration by the single first row of winglets is more effective to the odd number of tube rows than to the even number ones. It may reach the third row without being blocked by the second row of the tubes, though its effect gets weaker downstream. The pressure loss due to the form drag of the winglets is predominantly brought about at the first row, so the pressure loss per unit length decreases with the increase of the number of tube rows. On the other hand, the pressure loss due to the form drag of the tubes decreases up to the third row but turns to increase for the five tube rows, since the effect of the winglets cannot reach the fifth row. This is the reason why the pressure loss penalty is reduced so much for three-rows tube bundle. But, more studies such as flow visualization and local heat transfer experiments need to be conducted.

4. Conclusions

The present study has been performed with a new strategy to obtain the heat transfer and pressure loss for the staggered finned-tube bundles with a single transverse row of the winglet pairs beside the first, front row of tubes. The effect of the number of transverse rows in tube bundle is investigated for two, three, four and five rows. The results are concluded as

(1) For staggered tube bundle without any winglet, the heat transfer is almost independent of the number of rows of tubes. On the contrary, the pressure loss

is slightly dependent inconsistently on the number of tube rows.

(2) The ratio of heat transfer enhancement, j/j_{G0}, was almost independent of the number of tube rows except for two rows in the entire range of the Reynolds numbers, showing the nearly constant increase of 10–30%. The heat transfer for the two rows was the highest and increased by 20–35%.

(3) The pressure loss penalty, f/f_{G0}, for two-row, four-row and five-row tube bundles is 1.0, 1.22–1.1 and 1.05, respectively, in Reynolds numbers (based on two times the channel height) below the value of 1000, and is nearly equal to each other in Reynolds numbers above the value of 1000.

(4) The reduction of the pressure loss penalty for the three-row tube bundle was the largest, in comparison with the other number of tube rows.

(5) For three-row tube bundle, the heat transfer was augmented by 30–10% and yet the pressure loss was reduced by 55–34% with the increase of the Reynolds number from 350 to 2100.

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References

[1] M. Fiebig, N. Mitra, Y. Dong, Simultaneous heat transfer enhancement and flow loss reduction of fin-tubes, in: Proc. 9th Int. Heat Transfer Conf. 4, Jerusalem, 1990, pp. 51–55.

[2] M. Fiebig, Vortex generators for compact heat exchangers, J. Enhanc. Heat Transfer 2 (1995) 43–61.

[3] K. Torii, K.M. Kwak, K. Nishino, Heat transfer enhancement accompanying pressure-loss reduction with winglet-type vortex generators for fin-tube heat exchangers, Int. J. Heat Mass Transfer 45 (2002) 3795–3801.

[4] R. Kawai, K. Nishino, K. Torii, PIV measurement of 3-D velocity distribution around finned tubes with vortex generators, in: 4th International Symposium on Particle Image Velocimetry, Goettingen, September 17–19, 2001, p. 1100.

[5] W.M. Kays, A.L. London, Compact Heat Exchangers, second ed., McGraw-Hill, New York, 1964 (Chap. 5).

[6] S.J. Kline, F.A. McClintock, Describing uncertainties in single sample experiments, Mech. Eng. 75 (1953) 3–8.

[7] W.E. Mercer, W.M. Pearce, J.E. Hitchcock, Laminar forced convection in the entrance region between parallel flat plates, Trans. ASME, J. Heat Transfer (1967) 251–257.

[8] R.K. Shah, A.L. London, Laminar flow forced convection in ducts, Adv. Heat Transfer (suppl. 1) (1978) 160–169.