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Investigation of the semi-dimple vortex generator applicable to fin-and-tube heat exchangers

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A B S T R A C T

The present study examines the air side performance of the fin-and-tube heat exchangers having semi-dimple vortex generator or plain fin geometry. A total of eight samples are made and tested with the corresponding fin pitch (F_p) being 1.6 mm and 2.0 mm and the number of tube row (N) are 1 and 2. The inlet air flow direction is also being tested upon the proposed semi-dimple VG. Test results indicate that the heat transfer performance of the proposed semi-dimple VG with $N = 1$ at a smaller fin pitch of 1.6 mm is slightly higher than that of plain fin geometry. For $N = 1$ with a larger fin pitch of 2.0 mm, the semi-dimple VG is about 10% higher than that of plain fin geometry. The difference in heat transfer performance amid VG and plain fin geometry becomes more pronounced with $N = 2$ and is especially evident when $F_p = 2.0$ mm due to mixing contribution. In general, the difference between plain and semi-dimple geometry becomes more conspicuous at a larger fin pitch because of the comparatively effectively swirled motion. Both geometries show a dependence on fin pitch at $N = 1$ but the effect is almost negligible when N is increased to 2. The inlet air flow direction casts negligible influence on the heat transfer performance of semi-dimple VG. However, the friction factors for the opposite air flow operation is lower than that of normal operation, especially in low Reynolds number region.

1. Introduction

In typical air-cooled heat exchanger applications, normally the air-side thermal resistance accounted for nearly or more than 90% of the total thermal resistance. Hence accommodation of large fin surface area is the generally adopted. In addition, protrusions or interrupted surfaces can be mounted on flat surfaces to provide better heat transfer performance. The surfaces can be in the form of continuous surfaces (e.g. plain, wavy) or interrupted (louver, slit, offset, and the like). Some review articles by Wang [1,2] had reviewed the patents of enhanced surfaces related to the fin-and-tube heat exchangers. From the 80 patents being surveyed, 90% of them are related to the interrupted surfaces. However, interrupted surface normally accompanied appreciable pressure drops. Accordingly, one of the recent designs is to introduce the so-called vortex generator (VG) which may ease the problem of significant pressure drop caused by highly interrupted surfaces. Through some specific protrusions, e.g. wing or winglet-type vortex generators with an angle of attack, desired heat transfer augmentation at the expense of affordable increase in pressure drop can be achieved [3]. For VGs applicable to the air-cooled heat exchangers, the first investigation was done by Edwards and Alker [4] who showed that the local heat transfer coefficient can be increased as much as 40%. Fiebig et al. [5] reported the improvements in heat exchanger performance by punching vortex generators on the primary heat transfer surface. Tiggelbeck et al. [6,7] examined the influence of rectangular wing and delta winglet on the performance of fin-and-tube heat exchanger. Their experimental results found that the inline arrangement is superior to staggered arrangement when VG is applied.

Biswas et al. [8], and Fiebig et al. [9,10] numerically investigated the influences of geometrical configurations of VG such as rectangular wing, triangular winglet and the corresponding geometry parameters like aspect ratio and attack angle. They concluded that an aspect ratio of 2 and an attack angle of 30° provides the best ratio of heat transfer/pressure drop. For an inline arrangement, 55–65% heat transfer enhancement with moderate rise of pressure drop of 20–45%. Wang et al. [11] conducted a water tunnel visualization experiment by utilization of an enlarged scale wave type

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VG applicable to fin-and-tube heat exchanger. Their results clearly indicated that introducing VGs greatly relief the futile transverse vortices behind the tube.

There had been numerous numerical studies associated with the performance of VG type fin-and-tube heat exchangers [12–18], but only very few studies had actually implemented VG in the actual fin-and-tube heat exchangers. For instance, He et al. [19] implemented a triangular winglet VG in a fin-and-tube heat exchanger having inline configuration. Their experimental results show little impact of the 10° array and a moderate heat-transfer improvement up to 32% for the small pair, both introducing additional pressure loss of approximately 20–40%. Wang et al. [20] compared air-side performance between delta-winglet VGs and wavy-fin surface in fin-and-tube heat exchangers under dry- and wet-surface conditions. With the rise of tube row, they found that the performance of the VG surface relative to the wavy-fin surface becomes more evident.

In this study, the authors propose an alternative VG configuration that is based on the dimple design. With the presence of dimple alongside the fin surface, the flip side becomes a hemisphere. As the air flow across the dimple surface, the flow separation may occur and it would generate a re-circulation zone and an upwash flow. The upwash vortices periodically flow out the dimple to give rise to horseshoe vortices and improved the heat transfer process accordingly. Tests are then performed and are compared with plain fin geometry. In essence, the overall objective of this study is therefore to present some detailed comparisons of the air side performance of the semi-dimple VG against and plain fin geometry. The effects of the fin pitch and the number of tube row will be also reported in this study.

2. Experimental setup

As tabulated in Table 1, a total of eight sample coils which includes plain and semi-dimple VG. The detailed dimension and the photo of the semi-dimple VG is schematically shown in Fig. 1. Notice that the fin thickness (δ_f), collar diameter (d_c), transverse pitch (P_t), and longitudinal pitch (P_l) for all the test samples are 0.11 mm, 7.5 mm, 21 mm, and 18.2 mm, respectively. The corresponding fin pitch (F_p) ranges from 1.6 to 2.0 mm and the number of tube row (N) spans from 1 to 2 as shown in Table 1. Detailed construction of the circuitry arrangement is identical to Wang et al. [21]. The experiments are conducted in an open wind tunnel as shown in Fig. 2. The ambient air flow was forced across the test section by means of a 5.6 kW centrifugal fan with an inverter. To avoid and minimize the effect of flow maldistribution in the experiments, an air straightener-equalizer and a mixer were provided. The inlet and the exit temperatures across the sample coil were measured by two T-type thermocouple meshes. The inlet measuring mesh consists of twelve thermocouples while the outlet mesh contains 36 thermocouples. The sensor locations inside the rectangular duct were established following ASHRAE recommendation [22]. These data signals were individually recorded and then averaged. During the isothermal test, the variance of these thermocouples was within ±0.2 °C.

The pressure drop of the test coil was detected by a precision differential pressure transducer, reading to 0.1 Pa. The air flow measuring station was a multiple nozzle code tester based on the ASHRAE 41.2 standard [23]. The working medium in the tube side was hot water. The inlet water temperature was controlled by a thermostat reservoir having an adjustable capacity up to 25 kW. Both the inlet and outlet temperatures were measured by two pre-calibrated RTDs (Resistance temperature device, Pt-100 Ω). Their accuracy was within 0.05 °C. The water volumetric flow rate is detected by a magnetic flow meter with 0.002 L/s resolution.

All the data signals are collected and converted by a data acquisition system (a hybrid recorder). The data acquisition system then transmitted the converted signals through GPIB interface to the host computer for further operation. During the experiments, the water inlet temperature was held constant at 60.0 ± 0.2 °C, and the tube side Reynolds number was approximately 38,000. Frontal velocities of inlet air ranged from 1 to 5 m/s. The energy balance between air side and tube side was within 2%. The water side resistance (evaluated as $1/h_iA_i$) was less than 10% of the overall resistance in all cases. The test fin-and-tube heat exchangers are tension wrapped having an “L” type fin collar. Thermal contact conductance provided by the manufacturers ranged from 11,000 to 16,000 W/m^2 K.

Table 1
Detailed geometric parameters of the test samples.

No.	F_p (mm)	N, row	Geometry
1	1.6	1	Plain
2	1.6	1	VG
3	2.0	1	Plain
4	2.0	1	VG
5	1.6	2	Plain
6	1.6	2	VG
7	2.0	2	Plain
8	2.0	2	VG

3. Data reduction

The *ε-NTU* method is applied to determine the *UA* product in the analysis heat transfer and pressure loss characteristics of the test coil from the experimental data. The detailed derivation of the heat transfer coefficient can be referred to Wang et al. [21] and will not repeat here. The obtained air sider heat transfer coefficient is then in terms of the Colburn *j* factor:

$$j = \frac{h_o}{\rho V_{max} Cp_a} \mathrm{Pr}^{2/3} \quad (1)$$

where $V_{max} = V_{fr}/\sigma$. The term, σ, is the ratio of the minimum flow area to frontal area. All the fluid properties are evaluated at the average values of the inlet and outlet temperatures under the steady state condition. The friction factors are calculated from the pressure drop equation proposed by Kays and London [24]. The relation for the dimensionless friction factor, *f*, in terms of pressure drop is shown below:

$$f = \frac{A_c}{A_o}\frac{\rho_m}{\rho_1}\left[\frac{2\Delta P\rho_1}{G_c^2} - \left(1+\sigma^2\right)\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \quad (2)$$

where A_o and A_c stand for the total surface area and the flow cross-sectional area, respectively. Uncertainties in the reported experimental values were estimated by the method suggested by Moffat [25]. The derived uncertainties of Colburn *j* factors range from 2.4 % to 7.3 % and is 3.6% to 11.2 % of the friction factors.

4. Results and discussion

The test samples are of plain and semi-dimple VG configurations with the number of tube row being 1, and 2. The corresponding fin pitches (F_p) are 1.6 and 2.0 mm, respectively. Test results are in terms of *j* and *f* factors. Fig. 3 denotes the test results for $N = 1$ for plain and semi-dimple VG geometry. For $N = 1$, it appears that the *j* and *f* semi-dimple VG is higher than those of the plain fin geometry. The Colburn *j* factors for VG surface is about 10% higher than that of plain surface when F_p is 1.6 mm and $\mathrm{Re}_{dc} > 1000$ but shows 20–40%