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An experimental study of the air-side performance of fin-and-tube heat exchangers having plain, louver, and semi-dimple vortex generator configuration

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ABSTRACT

In this study, a comparative study of the airside performance of fin-and-tube heat exchangers having plain, louver, and semi-dimple vortex generator (VG) are made. A total of eighteen 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, 2 and 4, respectively. Test results indicate that the heat transfer coefficient for $N = 1$ for louver fin geometry with a smaller fin pitch of 1.6 mm is higher than that of semi-dimple VG and plain fin geometry. For $N = 1$ with a larger fin pitch of 2.0 mm, the semi-dimple VG is marginally higher than that of louver fin geometry when the frontal velocity is lower than 2 m s^{-1}. However, the trend is reversed where the heat transfer coefficient for louver fin outperforms that of semi-dimple VG when the velocity is increased further. For the airside performance for $N = 2$ or $N = 4$, the heat transfer coefficients for louver fin geometry is about 2–15% higher than those of semi-dimple geometry. The difference is increased with the rising velocity and the results prevail for both fin pitches. However, the difference is smaller at a larger fin pitch due to the comparatively effectively swirled motion of the semi-dimple VG. The effect of the number of tube row on the heat transfer coefficients is negligible for louver fin geometry and is also rather small for semi-dimple VG. For the plain fin geometry, the effect of tube row is also small when $N > 1$. The heat transfer performance for $N = 1$ is different from $N = 2$ or $N = 4$ due to its inline configuration.

1. Introduction

Extended surfaces or fins are employed in heat exchangers for effectively improving the overall heat transfer performance. This is especially imperative for air-cooled heat exchanger since the dominant thermal resistance is usually on the air side. The surfaces can be in the form of continuous surfaces (e.g. plain, wavy) or interrupted (louver, slit, offset, and the like). Several review articles by Wang [1–2] had reported the patents of enhanced surfaces relevant to the fin-and-tube heat exchangers. He reported that 90% out of the surveyed patents were related to the interrupted surfaces. However, the interrupted surfaces showed appreciable pressure drops in association with heat transfer performance. In this connection, one of the recent designs is via introduction of the so-called vortex generator through which the heat transfer performance is attainable without pronounced increase of pressure drops. This is because the vortex generator can provides the swirled motion in which the additional transverse velocity components do not directly contribute to the rise of pressure drops as that of longitudinal velocity gradient. As a consequence, the heat transfer performance is improved with only a moderate increase of the pressure drop (Jacobi and Shah, [3]). There are various types of vortex generators used in aerodynamic application (wedge, plough, ramp, scoop, dome, wheeler, wing type, and wave element, ESDU [4]). For VGs applicable to the air-cooled heat exchangers, the early investigation by Edwards and Alker [5]reported a notable increase of local heat transfer coefficient of 76% alongside a delta-winglet vortex generator on a flat plate surface. Tiggelbeck et al. [6,7] examined the influence of rectangular wing and delta winglet on the performance of fin-and-tube heat exchanger. 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

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Nomenclature

A	surface area, m^2	P_t	transverse tube pitch, m
D_c	collar diameter, m	ΔP	pressure drop, Pa
F_p	fin pitch, m	U	overall heat transfer coefficient, W m^{-2} K^{-1}
h	heat transfer coefficient, W m^{-2} K^{-1}	V_{fr}	frontal velocity, m s^{-1}
h_i	inside heat transfer coefficient, W m^{-2} K^{-1}	ε	effectiveness, dimensionless
N	number of tube row	Δ_f	fin thickness, m
NTU	number of transfer unit, dimensionless		
P_l	longitudinal tube pitch, m		

of 2 and an attack angle of 30° results in 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%.

There were many numerical studies concerning the simulation of LVG in the fin-and-tube heat exchangers [11–20], some recent studies [16–20] had focused on the optimization of the implementations. Among these recent efforts, Jang et al. [16] investigated the optimum span angle and transverse location and the effect of different inlet conditions. The results showed that the maximum area reduction ratios reached 14.9–25.5% for the inline arrangement, and 7.9–13.6% of the maximum area reduction ratio was achieved for the staggered arrangement. Hu et al. [17] numerically studied the relationship between the intensity of secondary flow caused by vortex generator of fin-and-tube heat exchangers, and proposed a dimensionless parameter to characterize the intensity of secondary flow. Li et al. [18] proposed a radiantly arranged LVGs for performance improvement of the fin-and-tube heat exchangers. Their numerical results showed that the arrangement of LVGs is totally different from existing publications. In addition to the implementations of LVGs on the fin-and-tube heat exchangers, some other numerical efforts stressed on the LVGs on louvered fin-and-tube heat exchangers [21–23]. Note that most previously published researches aimed at concerning the performance of LVG on fin-and-tube heat exchangers, and only very few studies had actually experimentally implemented VG in the actual fin-and-tube heat exchangers.

Wang et al. [24,25] conducted a water tunnel visualization experiment by utilization of an enlarged scale wave type VG applicable to fin-and-tube heat exchanger. Their results clearly indicated that introducing VGs greatly relief the futile transverse vortices behind the tube. Among the few, He et al. [26] implemented a triangular winglet VG in a fin-and-tube heat exchanger having inline configuration. The experimental results show little impact of the 10 degree array and a moderate heat-transfer improvement up to 32% for the small pair, both introducing additional pressure loss of approximately 20–40%. In view of the shortage of the experimental data for the VG geometry, the present authors propose an alternative VG configuration that is based on dimple design, and test results are compared with plain and louver fin geometry. 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. As a consequence, the overall objective of this study is therefore to present some detailed comparisons of the airside performance of the semi-dimple VG against some counterpart fin geometries, i.e. louver and plain fin geometry. The effects of the fin pitch and the number of tube row will be also examined in this study.

2. Experimental setup

As tabulated in Table 1, a total of eighteen sample coils which includes plain, louver, and semi-dimple VG. The detailed dimension and the photo of the semi-dimple VG is schematically shown in Fig. 1(a) whereas the louver fin is depicted in Fig. 1(b). 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 1.6–2.0 mm and the number of tube row (N) spans from 1 to 4 as shown in Table 1. Detailed construction of the circuitry arrangement is identical to those by Wang et al. [27]. 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. These data signals were individually recorded and then averaged. During the isothermal test, the variance of these thermocouples was within 0.2 °C. In addition, all the thermocouples were pre-calibrated by a quartz thermometer with 0.01 °C precision.

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 [28]. 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.

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	1.6	1	Louver
4	2.0	1	Plain
5	2.0	1	VG
6	2.0	1	Louver
7	1.6	2	Plain
8	1.6	2	VG
9	1.6	2	Louver
10	2.0	2	Plain
11	2.0	2	VG
12	2.0	2	Louver
13	1.6	2	Plain
14	1.6	2	VG
15	1.6	2	Louver
16	2.0	2	Plain
17	2.0	2	VG
18	2.0	2	Louver