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Effect of inclination angle on free convection thermal performance of louver finned heat exchanger

Atipoang Nuntaphan [a,*], Sanparwat Vithayasai [b], Tanongkiat Kiatsiriroat [b], Chi-Chuan Wang [c]

[a] *Mae Moh Training Center, Electricity Generating Authority of Thailand, Mae Moh, Lampang 52000, Thailand*
[b] *Department of Mechanical Engineering, Faculty of Engineering, Chiang Mai University, Chiang Mai 50200, Thailand*
[c] *Energy and Resources Laboratories, Industrial Technology Research Institute, Hsinchu 310, Taiwan, ROC*

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Abstract

The effect of inclination angle on the louver finned tube heat exchanger subject to natural convection condition is reported in this study. It is found that the inclination angle plays an importance role on the performance of the louver finned heat exchanger. Performance of the heat exchanger is associated with the interactions between fin, louver, tube, and inclination angle. The heat transfer performance generally decreases with the rise of the inclination angle. This decrease of heat transfer performance is due to the blockage fin and its reversed heat dissipating direction against the raising air. However, at an inclination angle such as 30–45°, a considerable increase of heat transfer performance is seen. This is because appreciable amount of air flow was directed by the louver, causing a "louver-directed" phenomenon as that of in forced convection. With a further increase of inclination angle, the blockage effect caused by the fin is so strong as to offset the "louver-directed" phenomenon. Unlike those shown in force convection, the heat transfer performance decreased with the number of tube row.

Keywords: Louver fin and tube heat exchanger; Free convection; Inclination

1. Introduction

Louvered fin surfaces are very common for engine cooling, air-conditioning apparatus, aircraft, and air cooler. The popularity of the louver surface may attribute to its superior heat transfer performance by continuous renewing the boundary layer of the air flow. For automotive application, such as radiators, condensers, and evaporators, the louver fins were generally brazed (or soldered, mechanically expanded) to a flat, extruded tube, with a cross-section of several independent passages, and formed into a serpentine or a parallel flow geometry. Fig. 1 shows this kind of louver fin and tube heat exchanger. The exploitation of flat tube instead of round tube is due to its considerable small friction drag and high fin efficiency [1].

There are many researchers regarding to the air-side performance of this heat exchanger such as Webb and Jung [1], Davenport [2], Achaichia and Cowell [3], Rugh et al. [4], Tanaka et al. [5], Chang and Wang [6] who presents considerable amount of air-side data for louver fin geometry. Based on the test results of 91 samples, Chang and Wang [6] and Chang et al. [7] developed a general heat transfer and friction correlation from the foregoing researches.

The previous research works are normally dealing with the air-side performance under force convection. In many process applications, such as the condenser part of a thermosyphon paddy bulk storage and the heater of a small drying machine, the fin tube heat exchangers are operated under natural convection. Despite its low thermal performance,

* Corresponding author.
E-mail address: atipoang.n@egat.co.th (A. Nuntaphan).

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Nomenclature

a	parameter	Ra	Rayleigh number (dimensionless)
A_o	outside surface area of heat exchanger (m^2)	T_a	ambient air temperature (°C)
b	parameter	T_s	surface temperature of heat exchanger (°C)
c	parameter	T_{wi}	inlet temperature of water (°C)
C_{pw}	specific heat of water (J kg^{-1} K^{-1})	T_{wo}	outlet temperature of water (°C)
d_h	hydraulic diameter of flat tube (m)	x	fin spacing (m)
g	gravitational acceleration (9.81 m s^{-2})	y	fin width (m)
h_o	air-side heat transfer coefficient (W m^{-2} K^{-1})	α_a	thermal diffusivity of air (m^2 s^{-1})
k_a	thermal conductivity of air (W m^{-1} K^{-1})	β	volumetric thermal expansion coefficient (K^{-1})
k_f	thermal conductivity of fin (W m^{-1} K^{-1})	δ_f	fin thickness (m)
l	fin length (m)	η	fin efficiency
$\dot{m}_w$	mass flow rate of water (kg s^{-1})	ν_a	kinematics viscosity of air (m^2 s^{-1})
Nu	Nusselt number (dimensionless)	θ	inclination angle
Q	heat transfer rate (W)		

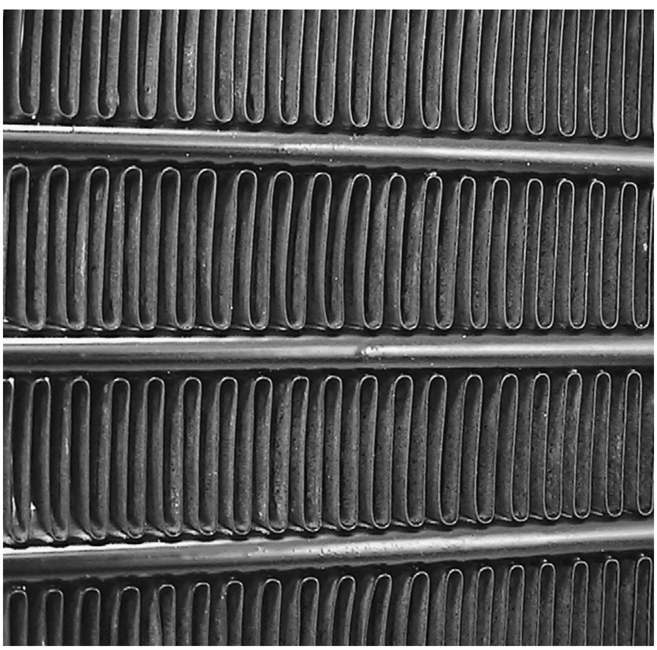

Fig. 1. Louver fin and tube heat exchanger.

benefits of natural convection such as noise-free and power-free makes it quite attractive in certain applications. However, very rare data are available in the open literature regarding to the performance of the louver finned heat exchanger under free convection. Farhadi et al. [8] presented the temperature distribution in a finned tube bundle of an A-type air cooler, and proposed a general correlation for this arrangement under natural convection condition. However, he did not report the effect of inclination despite an inclination angle is usually applied to the A-type cooler. In that regard, it is the objective of this study to examine the influence of inclination angle. This study adopts a commonly used louver finned heat exchanger (automotive radiator). The associated effect caused by induced air flow under natural convection on the heat transfer performance subject to inclination will be examined in this study.

2. Experimental set-up and data reduction

Fig. 2(a) shows the schematic of the experimental apparatus. The louver finned heat exchanger exchanges heat between the hot water flowing in the tube side and heat is transferred to the ambient air by free convection. In this work, the ambient air temperature is approximately 27 °C whereas the three different inlet temperatures of water (40 °C, 60 °C and 80 °C) are used at a fixed flow rate of 1.5 L/min. The water flow rate is measured by a rotameter having ±0.05 L/min accuracy. The inlet and the outlet temperatures of water, the ambient temperature and the external surface of the heat exchanger are measured by a set of K-type thermocouples being calibrated with ±0.1 °C accuracy. From preliminary study, it is found that the difference among these surface temperatures is lower than 3%. Therefore, the average surface temperature is used in this work. In this research, the inclination angle (θ) of the heat exchanger is varied from 0° to 90° from the horizontal line.

The effect of number of tube row (1–4 rows) on the air-side performance is examined in this study. These heat exchangers have the same frontal area, fin spacing, and fin pattern. Fig. 2(b) shows the details of the heat exchanger. Note that, the tested louver fin and flat tube heat exchangers are commercially available automobile radiator. The heat transfer rate of louver fin and tube heat exchanger (Q) can be calculated from the water temperature drop in the tube-side as

$$Q = \dot{m}_w C_{pw}(T_{wi} - T_{wo}). \tag{1}$$

For natural convective heat transfer of this study, the associated fin efficiency can be approximated by Schmidt approximation [9]:

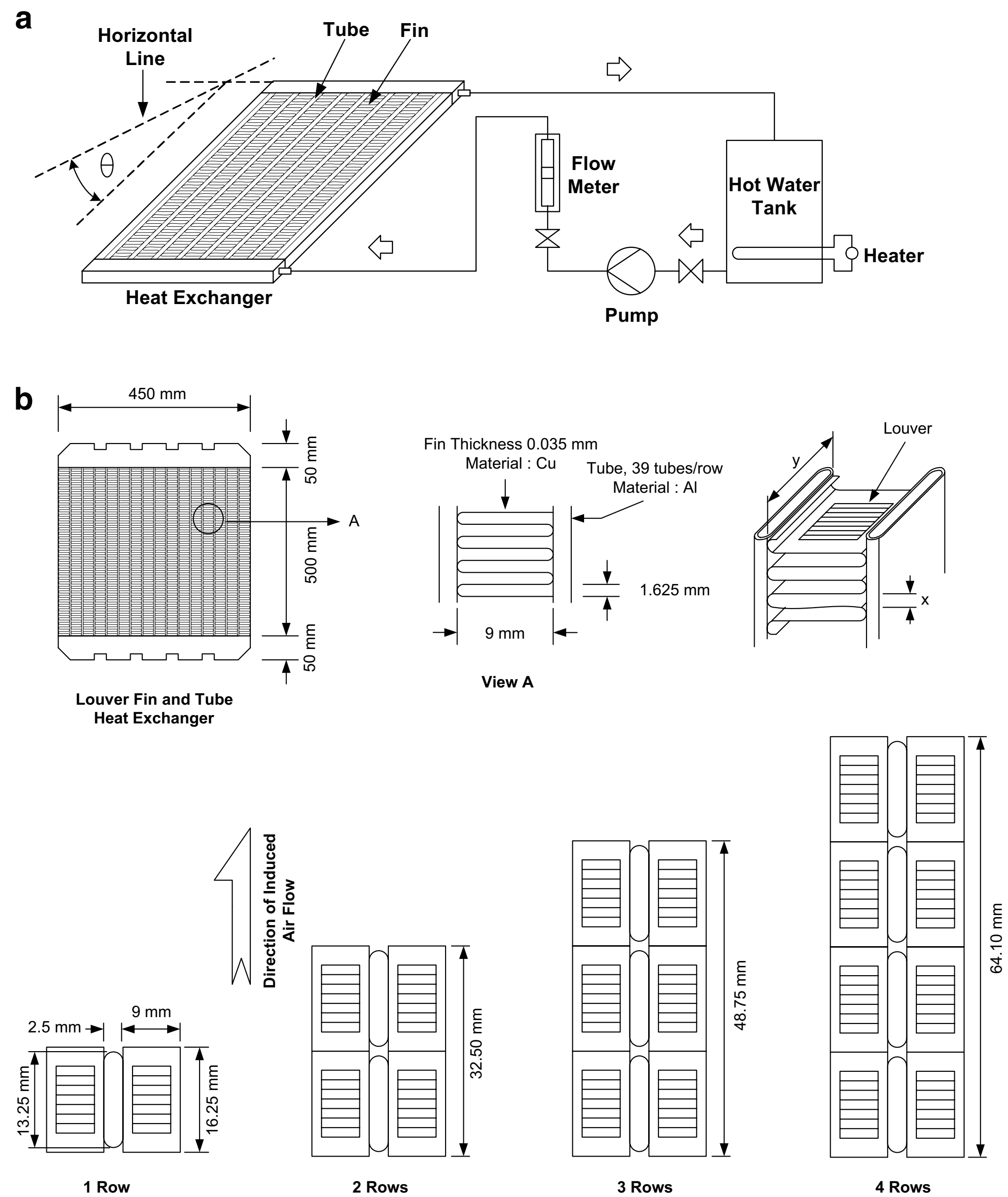

Fig. 2. (a) Schematic of the experimental apparatus and (b) geometrical parameters of the automobile radiator.

$$\eta = \frac{\tan(ml)}{ml}, \tag{2}$$

$$m = \sqrt{\frac{2h_o}{k_f \delta_f}}. \tag{3}$$

In case of natural convection, the fin efficiency is always above 97%. Hence, the heat transfer coefficient can be calculated from

$$h_o = \frac{Q}{A_o(T_s - T_a)}. \tag{4}$$

Uncertainty of the reduced heat transfer coefficient is within 10%. This heat transfer performance is in terms of the Nusselt number (Nu) and the Rayleigh number (Ra) defined as follow:

$$Nu = \frac{h_o d_h}{k_a}, \tag{5}$$

$$Ra = \frac{g\beta(T_s - T_a)d_h^3}{\nu_a \alpha_a}. \tag{6}$$

In this work, the volumetric thermal expansion coefficient is defined as

$$\beta = \frac{1}{0.5(T_s + T_a)}. \tag{7}$$

3. Results and discussion

The effect of inclination angle on the performance of louver finned tube heat exchanger is shown in Fig. 3. The

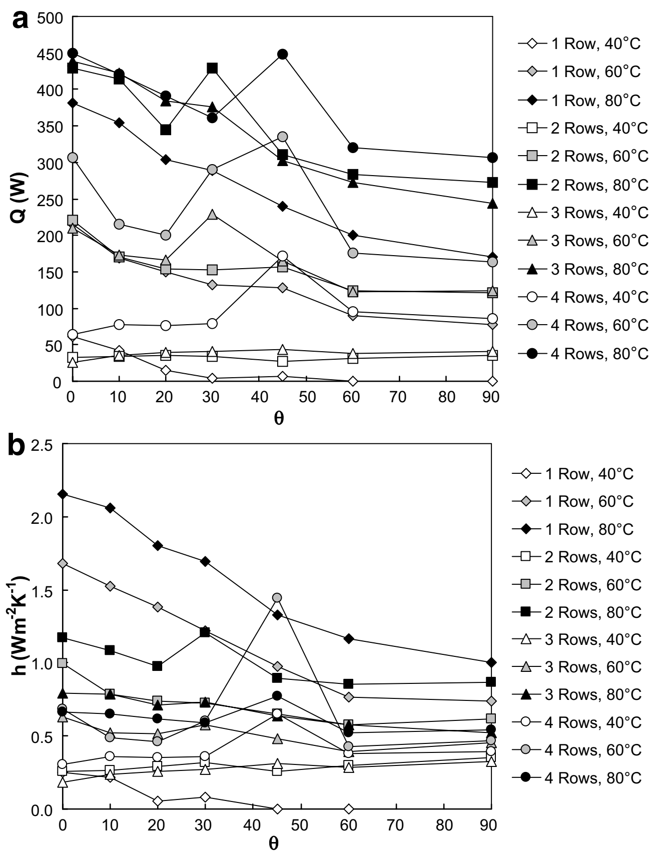

Fig. 3. (a) Heat transfer rate and (b) air-side heat transfer coefficient of heat exchanger at various inclination angles.

water flow rate is fixed and the inlet temperature of waters ranges from 40 to 80 °C. The total heat transfer rate is shown in Fig. 3(a) whereas the corresponding heat transfer coefficients are displayed in Fig. 3(b).

As shown in Fig. 3(a), it is found that the heat transfer rate is generally decreased with the rise of inclination angle. The highest performance occurs at an inclination angle of 0° whereas the poorest performance is at $\theta = 90°$. The decline of heat transfer performance vs. inclination angle is associated with the blockage of fin. A schematic showing the interactions of induced air flow with fin, tube, and louver is shown in Fig. 4. For $\theta = 0°$ as shown in Fig. 4(b), the un-louvered fin is almost parallel to the direction of induced airflow. Hence, most of the induced airflow is directed by the un-louvered fin. The results are in line with those reported by Davenport [2] and Achaichia and Cowell [3] from automotive multi-louver fin surfaces at very low Reynolds number. This is a rather interesting phenomenon for louver fin surfaces. Webb and Trauger [10] found that at low Reynolds number some of the air streams bypass the louvers and act as "duct flow" between the fin channels. Achaichia and Cowell [3] identified the flow pattern as "fin directed flow" for low Reynolds number region and "louver directed flow" for high Reynolds number region.

With the rise of inclination angle, the induced air flow is no longer all parallel to the un-louvered fin. As can be seen from Fig. 4(c) at the extreme case of $\theta = 90°$, the un-louvered fin surface is perpendicular to rising airflow direction. Hence the un-louvered fin acts as a blockage of rising airflow. In addition, half of the heat dissipated from the un-louvered fin is heading downwards from the fin which is against the direction of hot rising air. As a consequence one can see the poorest heat transfer performance occurs at $\theta = 90°$. This phenomenon becomes more pronounced when the inlet water temperature is increased. This is expected because the heat transfer coefficient for natural convection is increased with the rise of temperature difference. Hence, the blockage effect of hot rising air become more and more severe when the temperature difference is increased. For an inlet temperature of 80 °C and an ambient temperature of 27 °C, it is found the heat transfer performance for $\theta = 0°$ outperforms that of $\theta = 90°$ for more than 100%.

Note that the heat transfer performance is not constantly decreased with the inclination angle. It is interesting to see a very special result occurred near $\theta = 30$–$45°$ where a reversed trend showing an appreciable increase of heat transfer performance is seen. The unexpected increase of heat transfer performance is due to the interactions of air flow from un-louvered fin, louver, and tube. The presence of un-louvered fin deteriorates the heat transfer performance when the inclination angle is increased. However, as shown in Fig. 4(d), the amount of induced air flow flowing across the fin portion is split into two streams as the inclination angle is increased. Originally, the flow pattern is duct flow with very little induced airflow passing across louver portion. More and more induced air flow passes across the louver portion when the inclination angle is increased. Eventually the flow pattern may become louver directed at certain inclination angle, leading to a dramatic increase of heat transfer performance due to relatively good mixing between adjacent fins. Notice that the louver angle of the present fin geometry is 30–45°. A further increase of inclination angle, the negative influence of blockage from the un-louvered fin offsets the benefits from louver portion, giving rise to a decline of heat transfer performance accordingly. The effect of tube row on the heat transfer performance is also shown in Fig. 3. Despite the increase of row number generally increase the overall heat transfer rate (Fig. 3(a)), the heat transfer coefficient decreases with the number of tube row. The results are contrary to those tested in forced convection at higher Reynolds number. The effect of the number of tube row on the heat transfer performance of louver fin geometry surface is negligible either for a flat tube geometry [1–7] or for a round tube configuration [11,12]. This is due to the louver directed flow pattern in high Reynolds number region cause significant mixing.

Based on the test results, one can see considerable influence of inclination angle and the number of tube row. Hence, an empirical fit of the test data is made, yielding the following correlation:

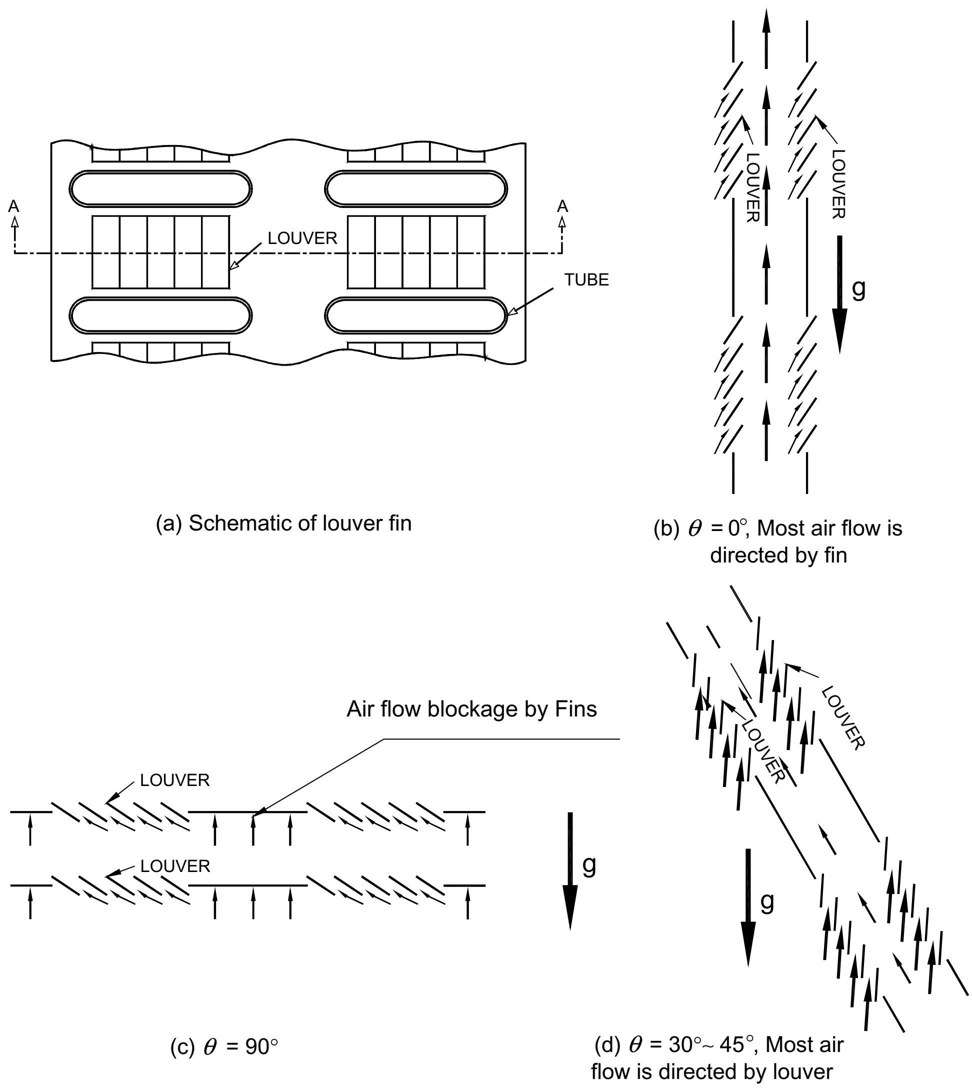

Fig. 4. Flow patterns of louver fin and tube heat exchanger.

For $0^\circ \leqslant \theta \leqslant 20^\circ$ or $60^\circ \leqslant \theta \leqslant 90^\circ$,

$$Nu = 2.1225Ra^a\left(\frac{x}{y}\right)^b(1+\cos\theta)^c, \tag{8}$$

$$a = -0.7922 + 1.3168\left(\frac{x}{y}\right) + 0.7460\cos\theta - 18.7330Ra^{-1},$$

$$b = -0.2647 + 0.5594\cos\theta,$$

$$c = -1.7627 - 3.4659\left(\frac{x}{y}\right) + 7.1466Ra^{-1}.$$

For $20^\circ < \theta < 60^\circ$,

$$Nu = 7.3239\times 10^5 Ra^a\left(\frac{x}{y}\right)^b(\cos\theta)^c, \tag{9}$$

$$a = -2.4027 - 0.8707\left(\frac{x}{y}\right) + 0.7822\cos\theta - 103.9500Ra^{-1},$$

$$b = -0.0235 + 1.6825\cos\theta - 55.3850Ra^{-1},$$

$$c = 1.8589 - 8.8131\left(\frac{x}{y}\right).$$

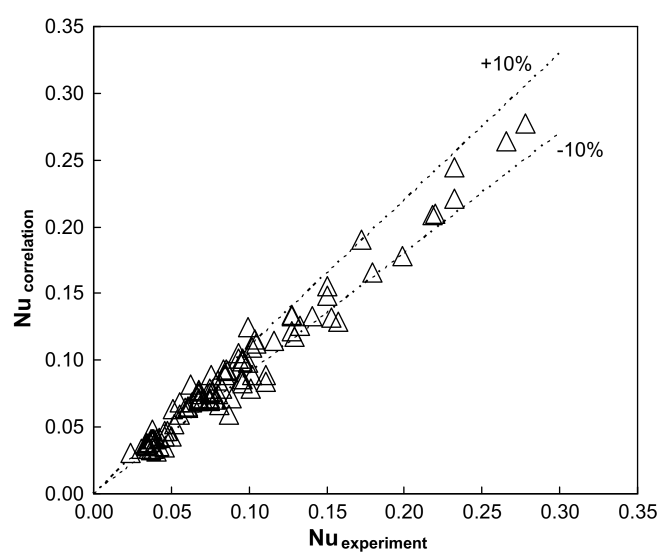

Fig. 5. Comparison of Nu from experiment and correlation.

Note that, the parameters x and y denotes spacing between adjacent fin and the width of fin from the first row to the last row, respectively. Fig. 5 shows the comparison of the Nusselt number from the experiment and the developed correlation and it is found that the proposed correlation can predict 71.4% of the experimental data with in ±10%.

4. Conclusion

This study examines the effect of inclination angle on the louver finned tube heat exchanger subject to natural convection. It is found that the inclination angle plays an importance role on the performance of the louver finned heat exchanger. Performance of the heat exchanger is associated with the interactions between un-louvered fin, louver, tube, and inclination angle. A short summary of the test results is as follows:

1. The heat transfer performance generally decreases with the rise of the inclination angle. This decrease of heat transfer performance is due to the blockage of un-louvered fin and its reversed heat dissipating direction against the raising air.
2. At an inclination angle such as 30–45°, a reversed trend showing a considerable increase of heat transfer performance is seen. This is because appreciable amount of air flow was flowing across louver portion, causing a "louver-directed" phenomenon as that in forced convection. However, a further increase of inclination angle, the blockage effect by the un-louvered fin is so strong as to offset the "louver-directed" phenomenon.
3. Unlike those shown in force convection, the effect of the number of tube row on the heat transfer performance becomes more and more pronounced with the temperature rise. Generally the heat transfer performance decreased with the number of tube row.
4. A correlation is proposed to describe the associated influence of inclination angle. The correlation can describe 71.4% of the experimental data within ±10%.

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References

[1] R.L. Webb, S.H. Jung, Air-side performance of enhanced brazed aluminum heat exchangers, ASHRAE Trans. 98 (Pt2) (1992) 391–401.
[2] C.J. Davenport, Correlation for heat transfer and flow friction characteristics of louvered fin, AIChE Symp. 79 (1983) 19–27.
[3] A. Achaichia, T.A. Cowell, Heat transfer and pressure drop characteristics of flat tube and louvered plated fin surfaces, Exp. Therm. Fluid Sci. 1 (2) (1988) 147–157.
[4] J.P. Rugh, J.T. Pearson, S. Ramadhyani, A study of a very compact heat exchanger used for passenger compartment heating in automobiles, Compact Heat Exchangers for Power and Process Industries, ASME-HTD 201 (1992) 15–24.
[5] T. Tanaka, M. Itoh, M. Kudoh, A. Tomita, Improvement of compact heat exchangers with inclined louvered fins, Bull. JSME 27 (1984) 219–226.
[6] Y.J. Chang, C.C. Wang, A generalized heat transfer correlation for louver fin geometry, Int. J. Heat Mass Transfer 40 (3) (1997) 533–544.
[7] Y.J. Chang, K.C. Hsu, Y.T. Lin, C.C. Wang, A generalized friction correlation for louver fin geometry, Int. J. Heat Mass Transfer 43 (12) (2000) 2237–2243.
[8] F. Farhadi, N. Davani, P. Ardalan, New correlation for natural convection of finned tube A-type air cooler, Appl. Therm. Eng. 25 (17–18) (2005) 3053–3066.
[9] Th.E. Schmidt, Heat transfer calculation for extended surface, Refrig. Eng. (1949) 351–357.
[10] R.L. Webb, P. Trauger, Flow structure in the louvered fin heat exchanger geometry, Exp. Therm. Fluid Sci. 4 (2) (1991) 205–217.
[11] C.C. Wang, Y.P. Chang, K.Y. Chi, Y.J. Chang, An experimental study of heat transfer and friction characteristics of typical louver fin and tube heat exchangers, Int. J. Heat Mass Transfer 41 (4–5) (1998) 817–822.
[12] C.C. Wang, C.J. Lee, C.T. Chang, S.P. Lin, Heat transfer and friction correlation for compact louvered fin-and-tube heat exchangers, Int. J. Heat Mass Transfer 42 (11) (1999) 1945–1956.