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Thermal characteristics of louvered fins with a low-reynolds number flow †

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Abstract

A heat recovery system is crucial for the effective use of energy where heat rejection from production processes is unavoidable and must be reused. The response of the louvered fins to the low-Reynolds number hot gas is yet to be reported in the literature for the application of a heat exchanger on low-speed hot plume arising from heat sources in production processes. This study focuses on the effects of the louvered fin heat exchanger's design parameters, which include the louver pitch and louver angle, on the convective heat transfer, which defines the thermal interaction between the hot, buoyant, naturally-induced air and the louvered fins. The resulting Colburn factors (*j*) are compared with those derived under forced convection with a similar range of low Reynolds number (233 to 1024). All experiments are done on a 15:1 scaled-up model. The fin aspect ratios between the fin spacing and louver pitch are set at 0.75, 1, and 1.5, while the louver angles are set at 18°, 23°, 30°, 35°, and 40°. The Colburn factor strongly depends on the louver angle, especially at the lower range of the Reynolds number. The decreasing aspect ratio induces more hot buoyant air into the louver-formed channels, increasing the heat transfer rate. When the fin angle increases towards 30°, a larger Colburn factor is produced. However, the heat transfer characteristic drops as the angle goes beyond 30°. The highest *j* for the low speed flow is attained when the louver angle is 30° and the fin aspect ratio is 1.

Keywords: Cooing/reheating system; Energy saving; Compact heat exchanger

1. Introduction

Louvered fin compact heat exchangers (Fig. 1) are used in a variety of applications, including in heat rejection systems in automobiles, residential air-conditionings, oil coolers, and radiators. One of the most important objectives of past investigations on compact heat exchangers has been the development of high performance heat exchangers which do not sacrifice lightness and small volume. With its large heat transfer area per unit volume, the compactness of the louvered fins is structured for recovering energy from a heat source. Achaichia and Cowell [1] were the first to experimentally verify that it is relatively important to design louvered fins with a louver-directed flow rather than a duct-directed flow [Fig. 2(a) and Fig. 2(b)]. Their study indicated an increase in heat transfer coefficients when the flow was in transition from being duct-directed to being louver-directed. They used the heat transfer characteristics of the louvered fins to determine the flow efficiency. For the forced convection flow over the cooler louvered fins, Tafti et al. [2] reported that the flow inside the multi-louvered fin array was unstable when ReL_p was 900. Webb and Trauger [3] also reported that, under a critical Reynolds number, the flow characteristics depend on the Reynolds number and the louver angle.

Zhang and Tafti [4] suggested that the Reynolds number, fin pitch, louver thickness, and louver angle have effects on the flow efficiency of multi-louvered fins. Their results showed that the flow efficiency is strongly dependent on the geometrical parameters, especially the low Reynolds number. The flow efficiency increases with the Reynolds number and louver angle, but it decreases with the fin pitch and thickness ratio. Webb and Trauger [3] used a dye injection technique in their 10:1 scale model. The geometric parameters, such as the louver pitch, louver angle, and fin pitch, were varied to determine their effect on the flow structure. The tests covered an Re range of 400-4000, based on the louver pitch. When the louver angle and louver pitch increase, the flow efficiency increases. This represents a louver-directed flow, where heat transfer is increased. However, if the fin pitch increases the flow efficiency decreases. Davenport [6] also reported a relation between the Reynolds number of the flow within the louvered fin compact heat exchanger and the flow direction.

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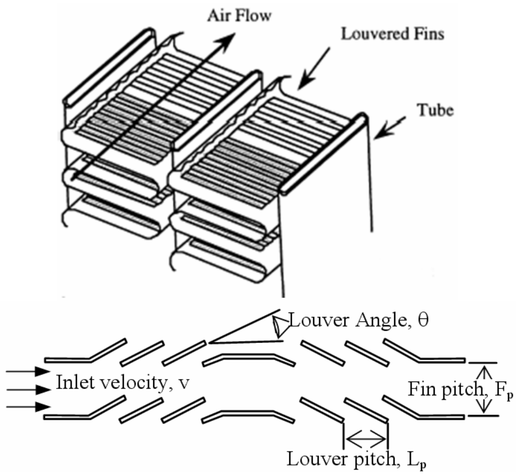

Fig. 1. Illustration of the louvered plate fin heat exchanger and the geometric parameters of the louvered fins [5].

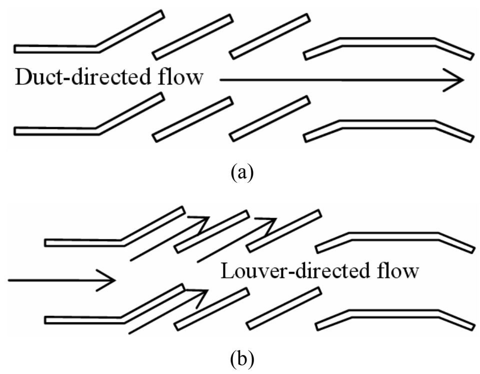

Fig. 2. (a) Duct-directed flow and (b) Louver-directed flow through a typical louver array [5]

Kim and Bullard [6] also showed the inverse relation between the heat transfer characteristic (Colburn factor, j) and the Reynolds number of the inlet flow for a slow flow. Their results are in good agreement with the results found by Davenport [5].

Lyman et al. [8] presented a method for evaluating the spatially resolved louver heat transfer coefficients using various reference temperatures, such as the bulk flow temperature and adiabatic wall temperature, to define the convective heat transfer coefficients. Based on this, larger fin pitches and higher louver angles yield better performance at low Reynolds numbers. At high Reynolds numbers, the performance is highly influenced by how many louvers the heated wakes influence downstream. Louver geometries in which the louvers are nearly aligned do not provide a good design from a heat transfer perspective, particularly for high Reynolds numbers. There are large variations in the performance of various louver models at lower Reynolds numbers compared to higher Reynolds numbers. The heat transfer coefficients based on the bulk flow temperature for a particular louvered fin correlates closely with its local thermal field, indicating the importance of the thermal field around a particular louver. Chang and Wang [9] proposed the use of a heat transfer characteristic as a function of the design parameters, most of which is included in this current investigation for the case of forced convection:

$$j = \mathrm{Re}_{L_p}^{-0.49}\left(\frac{\theta}{90}\right)^{0.27}\left(\frac{F_p}{L_p}\right)^{-0.14}\left(\frac{F_l}{L_p}\right)^{-0.29}\left(\frac{T_d}{L_p}\right)^{-0.23}\left(\frac{L_l}{L_p}\right)^{0.68}\left(\frac{T_p}{L_p}\right)^{-0.28}\left(\frac{\delta}{L_p}\right)^{-0.05} \quad (1)$$

The objectives of previous literature were to design a heat exchanger, such that a high-performance and a low pressure drop can be achieved. In this way, the forced convection heat transfer mostly dominates during cooling. Moreover, potential heat recovery from the hot air plume was investigated, and its upward movement was found to be induced by the temperature difference between the heat source and the quiescent and cooler surrounding, rather than between the fins and the hot air itself as in regular, natural convection. In other words, the hot flow with low Reynolds number is fed into the louvered fin heat exchanger instead of being induced.

In contrast, this research work uses a 15:1 scaled-up model of the louvered fins to study the effects of the geometry in relation to the low Reynolds number flow of hot, buoyant air plume. This has never been investigated before. It focuses on the application of heating instead of cooling. The objective of this work is to study the effects of the geometrical parameters, including the fin pitch (F_p), louver pitch (L_p), and louver angle (θ) of the louvered fins, to achieve better design parameters for the louvered fin compact heat exchanger. The application in interest is a heat recovery system utilizing hot and low-speed buoyant air plume, which is normally rejected from processing machines that are widely used in food and agricultural industries nationwide.

2. Experimentation

The heat transfer characteristics of louvered fins are described in terms of the Colburn factor (j). To achieve this, a testing tunnel for the experiments is constructed. Four columns of nine louvered fins are set into a 20x30x70 cm^3 test section. Cooled by water flow on both ends, each louvered fin is heated up by hot, buoyant air underneath. The natural convective heat transfer is then gauged by circulating the cooling water on both sides of each louver blade using a controlled centrifugal pump [Fig. 3(b)]. Their flow rates are maintained at 0.025 kg/s by a flow meter and a Rota meter. The water inlet and outlet separate in two ducts [Fig. 3(c)]. Heat balance leads to the calculation of the corresponding heat transfer rate. The adjustable fin pitch and fin angle of attack allows for alteration of the ratio F_p/L_p and θ. The louvered fin length (L_l) is kept at 4.5 cm. A buoyant plume of hot air is generated from the electrical heater installed at the bottom of the apparatus [Fig. 3]. Powered by five 1000 W heaters, the buoyant, hot air is naturally induced into the test section at a low speed, pass-

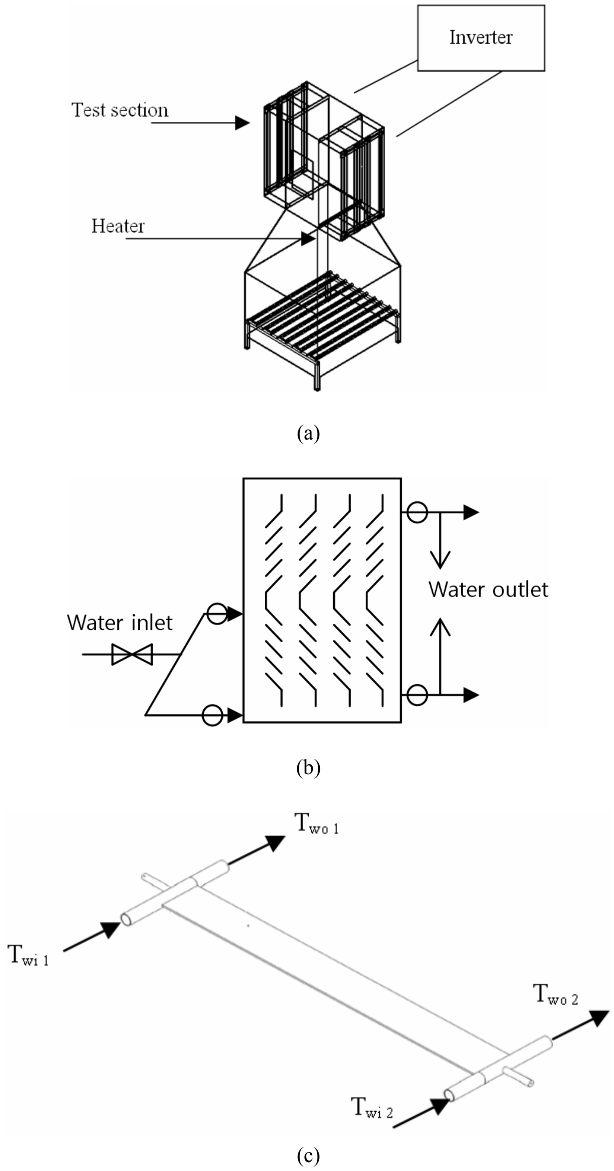

Fig. 3. (a) Experimental set-up; (b) the test section; and the (c) louver fin design with a scale-up of 15:1.

ing upward through the louvered fins installed within. The power of the heater can vary as the voltage input is altered. Confirmed by anemometer and Pitot tube, its speed is controlled between 0.1-1.2 m/s, corresponding to an air inlet temperature between 40-100 °C. In the experiment, all of the louver surfaces are attached to thermocouples, such that the surface temperatures are recorded under steady-state conditions to determine the average louver surface temperature.

From Newton's law of cooling, the predictive equation for the convective heat flux [W/m^2] is represented by:

$$q'' = \frac{Q}{A} = h\left(T_w - T_{ref}\right) \tag{2}$$

where h is heat transfer coefficient

A is the louver surface area

T_w is louver temperature

T_{ref} is the free steam temperature used as the reference temperature

For a low speed external flow application, the free stream temperature is taken as the reference temperature, and for the internal flow applications, the bulk flow temperature is taken as the reference temperature, similar to the work of Lyman et al. [8].

Heat transferred into each fin blade eventually cools down with water at both ends of the fin. Then, Q in Eq. (2) is calculated using heat balance, providing that the mass transfer rate of cooling water is controlled on both sides:

$$Q = mC_P\left(\Delta T_1 + \Delta T_2\right) \tag{3}$$

where ΔT_1 and ΔT_2 are the water inlet and outlet temperature differences on both ends. The convection heat transfer coefficient of air can be represented in the form of the Colburn factor, j, as:

$$j = \frac{h\,\mathrm{Pr}^{\frac{2}{3}}}{GCp} \tag{4}$$

where G is the maximum mass velocity, which can be written as:

$$G = \frac{\dot{m}_a}{A_{ff}} = \frac{\rho A_p v}{A_{ff}} \tag{5}$$

where

A_{ff} is the minimum cross-sectional free flow area

A_p is the test cross-sectional area

The heat transfer characteristics of the louvered fins are then determined from heat exchange with the buoyant air plume. study on F_p/L_p and θ is also performed. To be able to compare the results with the forced convective flow at the same speed, a pressure-driven flow is also set-up into the experimental apparatus. Using a small fan with an adjustable speed to draw hot air into the test section, the forced convective heat transfer coefficient is then determined and compared with that of the naturally-induced flow at a comparable range of Reynolds number.

3. Results and discussions

Data was collected for three values of the fin-to-louver pitch ratio (F_p/L_p = 0.75, 1, and 1.5) and five louver angles (θ = 18°, 23°, 30°, 35°, and 40°). Controlled by adjusting the electrical power delivered to the heater, the velocity of the hot air plume was reported as a Reynolds number based on the louver pitch, i.e., in spite of the fact that the flow was induced by buoyancy, the heat transfer characteristics were represented with the flow's Reynolds number, instead of the more common pa-

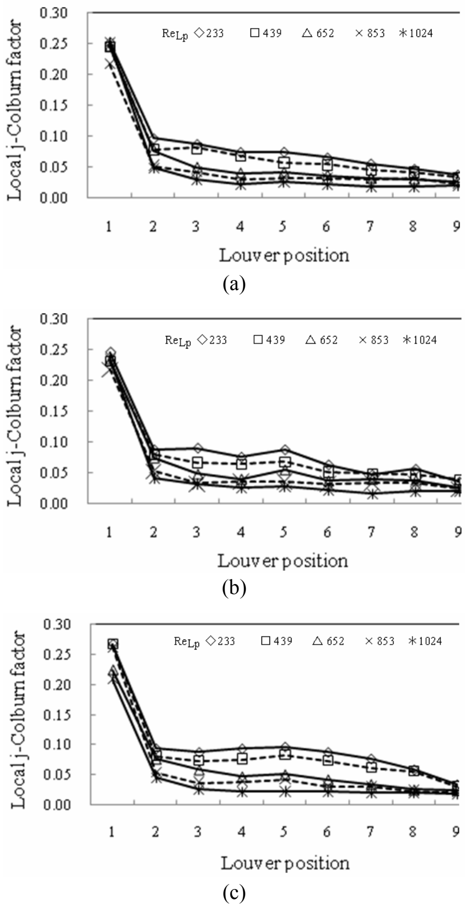

Fig. 4. Colburn factor for each position at $\theta = 18°$: (a) $F_p/L_p = 0.75$; (b) $F_p/L_p = 1$; and (c) $F_p/L_p = 1.5$.

rameter: the Grashof number. This was done so that the data could be compared with the equivalent pressure-driven flow from other studies (Lyman et al. [8]) and because the flow induced from the temperature difference between the heat source and the quiescent surrounding was fed into the louver fins. All Reynolds numbers were low to simulate the slow, hot, buoyant flow. It was limited to the range of 233 to 1024. All temperature readouts were commenced after a steady-state condition was attained.

3.1 Local Colburn factor of each louvered fin

The Colburn factor of each louvered fin was determined along the flow axis for 5 louver angles. All results were similar. An example showing the results at two selected angles of attack are shown in Fig. 4. The corresponding distance from the test section's inlet for each louver position is shown in Table 1. The Colburn factors of the louvered fin at the entry positions were larger than that at the other positions. Thus, a higher value of convective heat transfer coefficient was observed. The fins downstream, however, showed a lower performance since the temperature difference between the flow and the fin temperature decreases along the flow axis. At all angles, the heat transfer characteristic was found to be inversely proportional to the corresponding Reynolds number of the flow. The angle of attack showed moderate effects on the Colburn factor as it caused the heat transfer to increase when the louver angle increased. This factor shows a greater effect at lower speed flows. On the contrary, the effects of F_p/L_p were limited. For F_p/L_p in the range of 0.75 to 1.00, the corresponding Colburn factors did not change significantly.

Table 1. Louver position measured from inlet of test section (inlet-to-center).

Louver number #	1	2	3	4	5	6	7	8	9
Distance from inlet (cm)	10	15	20	25	35	45	50	55	60

3.2 Effect of fin-to-louver pitch ratio and louver angle

As F_p decreases, the duct-directed flow seemed to become unfavorable. A naturally induced flow would normally follow the path where the hydraulic resistance is less. Its direction would depend on the friction of the flow path in the louver direction relative to that in the duct direction. Figs. 5-7 show the influences of the fin-to-louver pitch ratio, where the inverse effects of higher Reynolds numbers were confirmed. At a small angle of attack, a small F_p/L_p allowed hot air to pass through more, and the flow might be more louver-directed. As F_p/L_p increased, it appeared that this wider fin spacing reduced the hydraulic resistance in the duct direction relative to that in the louver direction. This is a possible reason why the heat transfer factor dropped for larger F_p/L_p. The speed of the flow only slightly affected the heat transfer, especially at higher louver angles. The Colburn factors were almost similar at high Reynolds numbers, and the minimum effect of F_p/L_p was observed in this range. Changing the louver angle, however, produced a slight improvement in heat transfer. At an inclination angle of 40°, more buoyant, hot air seemed to be drawn into the spacing among louvers as its Colburn factor at low Reynolds number increased and the effect from F_p/L_p was completely reversed.

Figs. 5-7 also reveal the maximum Colburn factor ($9.0\text{x}10^{-2}$) that the flow could attain was at 30° louver angle and at F_p/L_p of 1. This is emphasized in Fig. 8, where the maximum Colburn factors that the flow could attain are shown at each Reynolds number.

3.3 Comparison with pressure-driven flow

To be able to compare the heat transfer characteristic of naturally-induced flow and pressure-driven flow, the results from this work were then compared with those from Lyman et al. [8] as shown in Figs. 9 to 11 for three different louver angles. To confirm the findings, experiments on forced convection via the use of a controllable electrical fan was also performed, and its results were compared with Lyman's data at every louver angle. Our forced convection results agree fairly with Lyman's data. The heat transfer characteristic of the

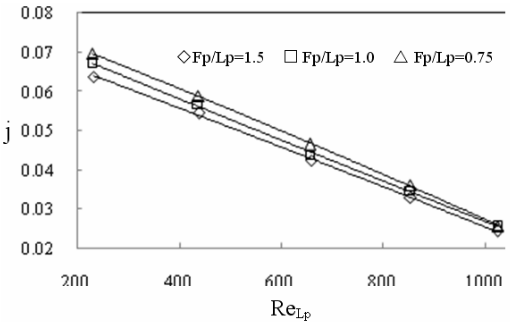

Fig. 5. Relation between the average Colburn factor and the corresponding Reynolds number at different F_p/L_p (θ= 18°).

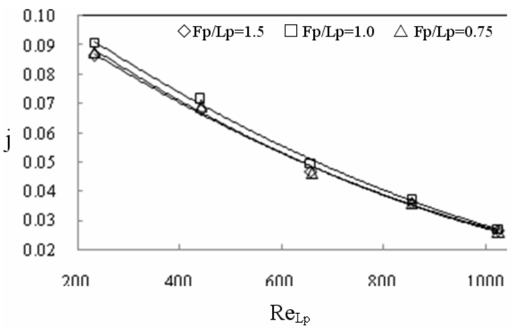

Fig. 6. Relation between the average Colburn factor and the corresponding Reynolds number at different F_p/L_p (θ= 30°).

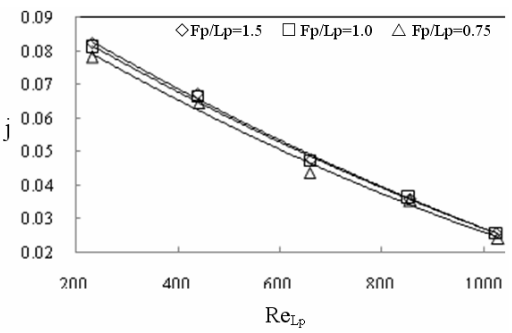

Fig. 7. Relation between the average Colburn factor and the corresponding Reynolds number at different F_p/L_p (θ= 40°).

pressure-driven flow taken from the study by Lyman et al. [8] was lower than that of the naturally-induced flow for the whole range of Reynolds numbers.

3.4 Heat balance and uncertainty analysis

The average percentages of heat loss are shown in Table 2. The largest margin occurred for those experiments at a louver angle of 18°. The corresponding uncertainty in the velocity measurement from the experiments was about 6-10% from its nominal value and that of the temperature difference was within 20%. Based on literature, the gas properties used in this work were estimated with an error of 1.0%, and errors from the geometric dimensions were small. With this estimation of measuring errors, the uncertainty of j could be calculated (Holman [10]) to be about 18.5%. These estimated uncertainties seem to be acceptable.

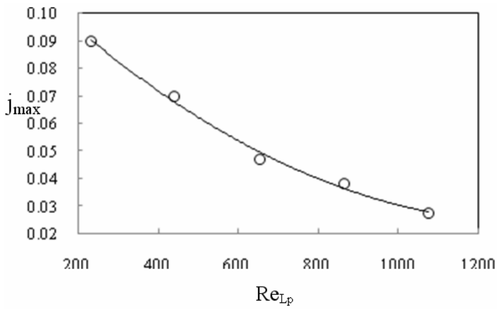

Fig. 8. Maximum Colburn factor that the flow at each Reynolds number can attain.

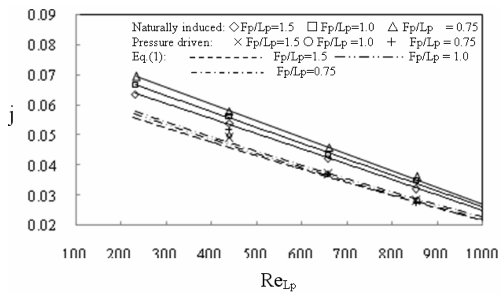

Fig. 9. Comparison between the Colburn factors of naturally induced and pressure-driven flows at a louver angle of 18°.

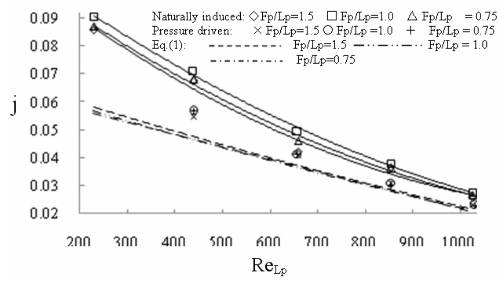

Fig. 10. Comparison between the Colburn factors of naturally induced and pressure-driven flows at a louver angle of 30°.

4. Conclusions

The optimized geometrical parameters of the louvered fins needed to obtain the most effective heat transfer using low-speed, buoyant, hot air were investigated. The experiments were conducted using a 15:1 scaled-up louver model with varied fin-to-louver pitch ratios and louver angles. The heat transfer characteristics of the louvered fins were determined in terms of the Colburn factor. The Colburn factor showed a strong dependence on geometrical parameters, such as louver pitch, fin pitch, and louver angle, especially at a low speed. A small fin angle of attack could improve the average heat transfer characteristics of the flow. However, if the angle was beyond some threshold value, the Colburn factor decreased. A louver angle of 30° and F_p/L_p=1 yielded the maximum Col-

Table 2. Heat loss during the heat transmission from naturally induced hot air to cooling water.

Louver Angle	18°	23°	30°	35°	40°
Heat loss (%)	25.2	20.1	18.1	13.2	10.0

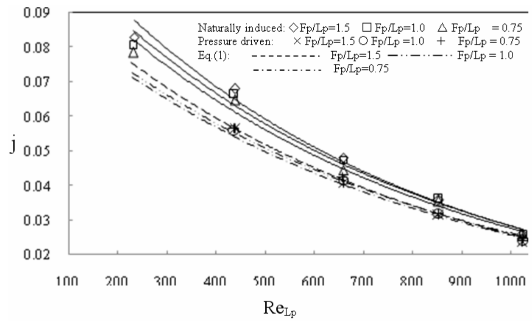

Fig. 11. Comparison between Colburn factors of naturally induced and pressure-driven flows at the louver angle of 40°.

burn factors at all Reynolds number. This suggested a possible conceptual design of the heat exchanger for recovering heat from a low-speed hot air plume, which would otherwise be rejected into the atmosphere, such as the case in some food industries. The heat transfer characteristics of the louvered fins between a naturally-induced and pressure-driven flow at equivalent Reynolds number were also compared. For a low speed flow with Reynolds numbers ranging from 233-1024, the hot, buoyant air plume yielded larger heat transfer characteristics than its counterpart pressure-driven flow.

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Nomenclature

m : Mass flow rate of cooling water (kg/s)
A_{ff} : Minimum cross sectional free flow area (m^2)
A_p : Test section area (m^2)
A_s : Louver area (m^2)
c_p : Specific heat/ heat capacity (kJ/kg.K)
h : Convective heat transfer coefficient (W/m^2.K)
j : Colburn factor
P : Pressure (Pa)
Pr : Prandlt number
Q : Rate of heat transfer (W)
Re_{Lp} : Reynolds number based on louver pitch and

$$\mathrm{Re}_{Lp} = \frac{\rho v L_p}{\mu}$$

T_{wo} : Outlet water temperature (K)
T_{wi} : Inlet water temperature (K)
V : Air speed (m/s)
ρ : Air density (kg/m^3)
μ : Viscosity (kg/m.s)

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