OPTIMIZATION OF THE LOUVER FIN-AND-TUBE HEAT EXCHANGERS—A PARAMETRIC APPROACH

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3D turbulent flow numerical simulations are performed to study heat transfer and flow friction characteristics of louver fin-and-tube heat exchangers. The effects of louver angle, fin pitch, transversal tube pitch, and longitudinal tube pitch on Colburn and friction factors are investigated in detail. Three stages of the effect of louver angle on the Colburn j factor are identified, including rising, level-off, and increasing again. A wide range for louver angle ranging from 0° (plain fin) to 80° is considered. Results show that the Colburn factor increases from a louver angle of 0° (plain fin) to 20°, followed by a level-off or slight decline until the louver angle reaches 45°, and finally, the Colburn factor is increased again with a further rise of louver angle. However, the friction factor increases with an increase in louver angle from 0° (plain fin) to 80°. Colburn and friction factors are almost independent of fin pitch while they decrease with an increase in transversal and longitudinal tube pitches. The optimum louver angle is around 20° upon carrying out the full factorial method in maximizing Colburn factor and minimizing friction factor.

KEY WORDS: heat transfer enhancement, louver fin-and-tube heat exchanger, louver angle, fin pitch, tube pitch, numerical simulation

1. INTRODUCTION

A heat exchanger is a device for carrying out heat exchange between two fluids in different temperatures. Air-cooled heat exchangers, containing hot fluid and air flow inside and outside of tubes, respectively, are widely used in refrigeration, air conditioning systems, and automobiles. Fins are usually required on the air side of these types of heat exchangers because the heat transfer coefficient of air is extremely low. Either flat, wavy, or louver fins can be adopted as fin surfaces

NOMENCLATURE

A_c	flow cross-sectional area (m ²)	St	Stanton number
A_f	fin surface area (m ²)	T	temperature (K)
A_o	total heat transfer area (m ²)	t	fin thickness (mm)
C	heat capacity (J/kg·K)	u	velocity (m/s)
D_c	tube collar outside diameter (mm)	W	width of louver region (mm)
	(m shall be used in equations)	Y^+	Y plus
f	friction factor	Greek Symbols	
F_p	fin pitch (mm)	α_k	inverse turbulent Prandtl numbers for k
G	air mass flux (kg/m ² ·s)	α_ε	inverse turbulent Prandtl numbers for ε
G_k	turbulent kinetic energy generation caused by mean velocity gradients (J/kg)	ΔP	pressure drop (Pa)
h	heat transfer coefficient (W/m ² ·K)	ΔT_{lm}	logarithmic mean temperature difference (K)
j	Colburn factor	ε	dissipation rate (m ² /s ³)
k	turbulent kinetic energy (J/kg)	η	fin efficiency
k_f	fluid thermal conductivity (W/m·K)	η_o	surface efficiency
k_{fin}	fin thermal conductivity (W/m·K)	θ	louver angle (degree)
K_c	entrance loss coefficients	μ	dynamic viscosity (kg/m·s)
K_e	exit loss coefficients	μ_t	turbulent dynamic viscosity (kg/m·s)
L	length of louver region (mm)	ρ	density (kg/m ³)
m	thermo geometry parameter (m ⁻¹)	σ	minimum to frontal flow area ratio
N	number of tube rows	ψ	fin efficiency parameter
N_l	number of louvers	Subscripts	
Nu	Nusselt number	i, j, k	tensor index
P	pressure (Pa)	in	inlet
P_l	longitudinal tube pitch (mm)	m	mean
P_t	transversal tube pitch (mm)	out	outlet
Pr	Prandtl number	s	solid wall
$\dot{Q}''$	heat transfer rate to the fluid per surfaces area (W/m ²)		
r	tube radius (m)		
r_f	equivalent circular fin radius (m)		
Re_{D_c}	Reynolds number based on tube collar outside diameter		

to remedy the poor performance of air in typical air-cooled heat exchangers. By employing the periodic boundary-layer renewal, the heat transfer coefficients (HTCs) in louver fins are much higher than flat and wavy fins, and it also comes with the price of an appreciable pressure drop penalty. The presence of louvers not only offers boundary layer restarting for heat transfer enhancement but also provides sufficient airflow mixing. The louvers are formed by punching the metal and are protruded out from the base metal (Shah and Sekulic, 2003). Although wide

studies on flat, wavy, and slit fins can be found in the literature (Awad and Muzychka, 2011; Glazar et al., 2012; Jang and Yang, 1998; Jiao and Baek, 2005; Kim and Cho, 2015; Kim and Youn, 2013; Lin et al., 2008; Sadeghianjahromi et al., 2018b, 2020; Taler, 2004; Tao et al., 2007; Wang et al., 1996, 1999, 2000), louver fins still attract a lot of attention for further optimization from the aspects of experimental and numerical investigations, Kays and London (1950) were the first, who summarized experimental results related to louver fins. The first paper regarding two-dimensional numerical simulations on louver fins was prepared by Baldwin et al. (1987). Some recent experimental and numerical studies on louver fins including their major conclusions are summarized in Table 1.

TABLE 1: Recent experimental and numerical studies on louver fins

No.	Research	Research type/ Tube type	Major conclusions
1	Achaichia and Cowell (1988)	Experimental/ Flat	<ol style="list-style-type: none"> 1. This paper was the first publication for louver fin-and-flat tube heat exchangers. 2. Flow is duct flow in low Re. 3. Correlations were developed for St and f.
2	Pauley and Hodgson (1994)	Experimental/ No tube	<ol style="list-style-type: none"> 1. Flow visualization was performed using dye injection. 2. The mixing angle increases with rise in Re. 3. The larger θ causes mixing angle to be larger.
3	Cowell et al. (1995)	Experimental/ Flat	<ol style="list-style-type: none"> 1. Operating mechanisms in multilouver fins were discussed. 2. Louver fins are suitable for applications with smaller size, less weight, and lower pumping power.
4	Atkinson et al. (1998)	Numerical/Flat	<ol style="list-style-type: none"> 1. 2D and 3D numerical simulations were discussed. 2. Both 2D and 3D models predict pressure losses accurately, while 3D models predict heat transfer higher than 2D models.
5	Wang et al. (1998a)	Experimental/ Circular	<ol style="list-style-type: none"> 1. N has negligible effect on f. 2. Effect of F_p on h is insignificant for $Re_{D_c} > 2000$. 3. Correlations for j and f factors were developed.
6	Leu et al. (2001)	Numerical/ Circular and oval	<ol style="list-style-type: none"> 1. 10% and 41% decrease in h and ΔP are obtained, respectively by using oval tube instead of circular tube. 2. Both h and ΔP increase with decrease of Louver pitch and axis ratio. 3. ΔP increases significantly with increase in θ.
7	Kim and Bullard (2002)	Experimental/ Flat	<ol style="list-style-type: none"> 1. 45 multilouver fin geometries were studied. 2. Correlations for j and f factors were developed with errors of $\pm 14.5\%$ and $\pm 7\%$, respectively.
8	Hsieh and Jang (2006, 2012)	Numerical/ No tube	<ol style="list-style-type: none"> 1. Louver fin with variable θ was investigated. 2. Up to 25.5% area reduction can be achieved in louver fin with variable θ in comparison with plain fin. 3. Optimization of louver fin-and-tube heat exchangers was performed using Taguchi method.

TABLE 1: (continued)

No.	Research	Research type/ Tube type	Major conclusions
9	Malapure et al. (2007)	Numerical/Flat	<ol style="list-style-type: none"> 1. At lower Re, flow is along fin, while at higher Re, flow is directed along louvers. 2. St and f decrease with F_p increase.
10	Li and Wang (2010)	Experimental/ Flat	<ol style="list-style-type: none"> 1. 4-region and 6-region louver fins have improvements in comparison with traditional fin. 2. Developed correlations for j and f can predict 85% of experimental results within $\pm 10\%$ and $\pm 20\%$, respectively.
11	T'Joel et al. (2011a)	Numerical and experimental/ No tube	<ol style="list-style-type: none"> 1. Inclined louver fin, which is a compound design of slit and louver fins, was investigated. 2. At low Re, flow is changed to unsteady because of recirculation zones that exist on the inclined regions in the fin array. 3. Inclined louver fin can perform better than louver and slit fins at low Re.
12	T'Joel et al. (2011b)	Numerical and experimental/ No tube	<ol style="list-style-type: none"> 1. A hybrid design of slit fin and standard louver fin was presented. 2. Flow visualization was performed in the water tunnel.
13	Vaisi et al. (2011)	Experimental/ Flat	In louver fins, symmetrical arrangement results in 9.3% rise in h and 18.2% reduction in ΔP in comparison with asymmetrical arrangement.
14	Dai et al. (2012)	Analytical/Flat	<ol style="list-style-type: none"> 1. Comparisons between heat exchangers made from metal foam with typical louver fin heat exchanger were performed. 2. Metal foam heat exchanger is smaller and lighter than louver fin heat exchanger in the same fan power, but it is more expensive.
15	Han et al. (2013)	Numerical/Oval and circular	<ol style="list-style-type: none"> 1. Wavy and louver fin-and-tube heat exchangers were studied. 2. Using oval tube in heat exchangers decreases flow resistance and augment heat transfer capacity, which results in rise in fin efficiency.
16	Čarija et al. (2014)	Numerical/Flat	<ol style="list-style-type: none"> 1. CFD simulations were done in $70 < Re < 350$. 2. Flat fin and louver fin were compared. 3. Using louver fin, up to 58% rise of heat transfer performance is found in $Re = 350$ in comparison to flat fin.
17	Dogan et al. (2015)	Experimental/ Flat	<ol style="list-style-type: none"> 1. Double-row louver fin was compared with triple-row louver fin. 2. Considering NTU and effectiveness, double-row louver fin is more effective.

TABLE 1: (continued)

No.	Research	Research type/ Tube type	Major conclusions
18	Jang and Chen (2015)	Numerical and experimental/ No tube	<ol style="list-style-type: none"> 1. Louver fin with variable θ was optimized. 2. At $133 < Re < 1199$, optimum design proposes 48.5%–55.2% reduction in area.
19	Karthik et al. (2015b)	Numerical and experimental/ Elliptical	For increasing the overall h , influence of rise of air velocity is greater than rise of water mass flow rate.
20	Karthik et al. (2015a)	Numerical and experimental/ Flat	<ol style="list-style-type: none"> 1. Decrease in F_p leads to the higher ΔP at a given frontal air velocity, but h increases in proportion with F_p. 2. The effect of increase in both P_t and P_l results in higher ΔP, while the effect of these parameters on h is very minimal.
21	Wang et al. (2015)	Experimental/ Circular	<ol style="list-style-type: none"> 1. 18 samples of fin-and-tube heat exchangers with plain fin, louver fin, and semi-dimple vortex generator were tested. 2. For plain fin in $N > 1$, louver fin, and semi-dimple vortex generator, effect of N on h is insignificant.
22	Srinivasu et al. (2016)	Numerical/Flat	<ol style="list-style-type: none"> 1. Using water/ethyl glycol and Al_2O_3 nano fluid in a radiator with louver fin was investigated. 2. Rise of volumetric concentration of nanoparticles enhances the efficiency of heat transfer.
23	Erbay et al. (2017)	Numerical/Flat	<ol style="list-style-type: none"> 1. 2D numerical simulations were performed. 2. f decreases with increasing F_p and Re. j increases when Re decreases. 3. Using volume goodness factor, maximum performance is achieved at $\theta = 20^\circ$ and $F_p = 1.5$ mm.
24	Javaherdeh et al. (2017)	Numerical/Flat	<ol style="list-style-type: none"> 1. Increasing θ leads to increase in h and ΔP and suitable θ varies from 24° to 32°. 2. Better performance can be obtained by smaller louver pitch and higher louver number. 3. Optimization of the inlet and exit nonlouver regions were carried out.
25	Hrnjak et al. (2017)	Experimental/ No tube	<ol style="list-style-type: none"> 1. Effects of θ (15°, 27°, and 39°) for three F_p (2.12, 1.41, and 1.69 mm) on ΔP and overall h of serpentine louver fin under periodic frosting were studied. 2. Performance of heat exchangers under frosting condition shall be in accordance with several-cycle experiments. 3. Under frosting condition, a larger louver pitch outperforms smaller ones.

TABLE 1: (continued)

No.	Research	Research type/ Tube type	Major conclusions
26	Okbaz et al. (2018)	Experimental/ Circular	<ol style="list-style-type: none"> 1. Flow visualization using PIV (particle image velocimetry) method was performed. 2. Louver fin was compared with wavy fin with the same size and specifications. 3. Thermal and hydraulic performances of louver fin are higher than wavy fin.
27	Sadeghian-jahromi et al. (2018a)	Numerical/ Circular	<ol style="list-style-type: none"> 1. Proposed correlations regarding j and f factors can predict 100% and 86% of data obtained by numerical simulation in $\pm 15\%$, respectively. 2. Optimum θ using genetic algorithm was obtained by utilizing developed correlations.
28	Asadi et al. (2018)	Numerical/ No tube	The constructal law was used to optimize the channel geometry with louver fins by changing louver pitch ratio, louver angle ratio, and inlet to outer louver length ratio.
29	Qian et al. (2018)	Numerical/ Circular	<ol style="list-style-type: none"> 1. Linear-up velocity profile is compared with linear-down profile in louver fins. 2. Linear-up velocity profile has 13%–32% lower Nu and 61%–66% lower ΔP compared to uniform velocity profile. 3. Linear-down velocity profile has 26%–34% higher Nu and 80%–89% higher ΔP compared to uniform velocity profile.

Upon Table 1, the following summary is concluded:

1. Most of the studies have been carried out on louver fin-and-flat tube heat exchangers (Achaichia and Cowell, 1988; Atkinson et al., 1998; Čarija et al., 2014; Cowell et al., 1995; Dai et al., 2012; Dogan et al., 2015; Erbay et al., 2017; Javaherdeh et al., 2017; Karthik et al., 2015a; Kim and Bullard, 2002; Li and Wang, 2010; Malapure et al., 2007; Srinivasu et al., 2016; Vaisi et al., 2011), while louver fin-and-circular tube heat exchangers have been taken into consideration by comparatively few researchers.
2. Complete elaborations regarding parametric study (especially those parameters which depend on geometry) requires numerous and tedious models for simulations, which is time-consuming. Therefore, in spite of this fact that many phenomena can be discovered in parametric studies, researchers are less interested in parametric study. So, sufficient parametric studies and optimizations have not been performed on louver parameters, especially on louver angles (Hsieh and Jang, 2012; Jang and Chen, 2015; Javaherdeh et al., 2017). Furthermore, the ranges of louver angle considered in different studies are limited, while louver angles ranging from 0° to near 90° have not been investigated in different studies.

- Although experimental studies have been carried out on louver fins, observations and results obtained by performing numerical simulations cannot be achieved by experimental analysis because the dimensions of louvers are very small and exact louver angles are difficult to measure correctly, thereby some uncertain errors in the experimental measurements may prevail. Also, for such compact geometries, using the experimental instruments such as pressure or temperature measurement tools to measure the local thermal-hydraulic characteristics might offer another interference upon the flow pattern. Thus, it is only probable to measure bulk parameters such as inlet and outlet pressure or temperature differences and nothing about the flow pattern details, recirculation zones, and so on are in hand. Conversely, modeling of louver fins with exact dimensions and performing flow visualization between louvers are easily possible in numerical analysis.

In this study, three-dimensional simulations are performed on both flat and louver fin-and-tube heat exchangers with circular tubes considering turbulent flow. The effects of different geometrical parameters on heat transfer (Colburn factor) and flow friction (friction factor) characteristics of louver fin-and-tube heat exchangers in wide ranges of louver angles from 0° (flat fin) to 80° are analyzed numerically. Detailed study of flow visualization between tubes and louver fins is executed. Also, optimization of louver angle is done by full factorial method. The results can be taken into consideration in design and optimization of louver fins, which are the most efficient fins and are widely used in applications and industries. Because louver angle is the most effective parameter of louver fins, the maximum ranges from 0° to 80° have been considered for louver angle in order to investigate its behavior thoroughly, which has been never done before.

2. MODEL DESCRIPTION

Figure 1 shows physical model and geometrical properties of a simulated louver fin-and-tube heat exchanger. The computational domain is also illustrated in Fig. 2. Basic values for simulated geometries in the validating cases (flat fin and louver fin) and parametric studies are specified in Table 2.

FIG. 1: Physical model of simulated louver fin (all dimensions are in mm)

FIG. 2: Computational domain of simulated louver fins

TABLE 2: Basic geometrical parameters for validation and parametric studies

Parameters	Parametric study	Validation of louver fin	Validation of flat fin
D_c (mm)	10.42	10.42	10.23
N	2	2	2
t (mm)	0.115	0.115	0.13
L (mm)	15	15	—
N_l	4	4	—
W (mm)	12.7	12.5	—
F_p (mm)	3.75	2.06	2.24
P_l (mm)	19.05	19.05	22
P_t (mm)	25.4	25.4	25.4
θ ($^\circ$)	0–80	13.5	—

2.1 Governing Equations

Governing equations for numerical simulations including continuity, momentum (RANS), and energy can be obtained on the basis of the following assumptions:

- The flow is steady, three-dimensional, incompressible, and turbulent.
- Air is working fluid, while its properties are assumed to be constant.
- Thermal radiation and natural convection can be ignored.

Therefore, the governing equations can be expressed as follows:

Continuity equation (Sadeghianjahromi et al., 2018a):

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (1)$$

Momentum equation (Sadeghianjahromi et al., 2018a):

$$\frac{\partial}{\partial x_j} \rho (\bar{u}_i \bar{u}_j) = -\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu_{eff} \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \rho (\overline{u'_i u'_j}) \right] \quad (2)$$

Energy equation (Sadeghianjahromi et al., 2018a):

$$\frac{\partial}{\partial x_j} \rho C (\bar{u}_j \bar{T}) = \bar{u}_j \frac{\partial \bar{P}}{\partial x_j} + \bar{u}'_j \frac{\partial \bar{P}'}{\partial x_j} + \frac{\partial}{\partial x_j} \left[k_f \frac{\partial \bar{T}}{\partial x_j} - \rho C \overline{u'_j T'} \right] \quad (3)$$

where ρ , P , u , C , T , and k_f are density, pressure, velocity, heat capacity, temperature, and fluid thermal conductivity, respectively. Mean values are denoted by bar signs, and fluctuations relative to mean values are shown by prime signs. μ_{eff} is the effective turbulent viscosity and is defined as follows:

$$\mu_{eff} = \mu + \mu_t \quad (4)$$

To simulate the turbulent viscosity, μ_t , the renormalization group (RNG) k - ϵ model was used as the turbulence model. It consists of two extra equations to calculate kinetic energy (k) and dissipation rate (ϵ) as follows:

$$\frac{\partial}{\partial x_i} (\rho k \bar{u}_i) = \frac{\partial}{\partial x_j} \left(\alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right) + G_k - \rho \epsilon \quad (5)$$

$$\frac{\partial}{\partial x_i} (\rho \epsilon \bar{u}_i) = \frac{\partial}{\partial x_j} \left(\alpha_\epsilon \mu_{eff} \frac{\partial \epsilon}{\partial x_j} \right) + C_{1\epsilon} \frac{\epsilon}{k} (G_k) - C_{2\epsilon}^* \rho \frac{\epsilon^2}{k} \quad (6)$$

One can find more details about RNG theory in Orszag et al. (1993). $Y^+ < 5$ is considered in numerical simulations since enhanced wall function is utilized for wall function.

2.2 Boundary Conditions and Numerical Method

Boundary conditions for numerical simulations are specified in the computational domain (Fig. 2). Air inlet velocity to the computational domain (x direction) is uniform and in the range of 1–6.5 m/s, while its components in other directions (y and z) are zero. Also, air inlet temperature (T_{in}) is considered constant at 303.15 K. Atmospheric pressure is assigned for pressure at air outlet. Normal gradients are supposed to be zero for side boundaries (symmetric planes). Wall temperatures are constant, which are equal to 333.15 K and 338.15 K for flat fin and louver fin validations, respectively. Parametric studies are performed with constant wall temperature of 338.15 K and no-slip condition for solid surfaces. Top and bottom boundaries are assumed to be periodic.

Governing equations are solved using traditional computational fluid dynamics (CFD) software, Ansys Fluent software (version 2019 R3). A SIMPLE algorithm is used to couple pressure and momentum equations. The discretized equations are solved by software until convergence occurs for which residuals are set to be less than 10^{-6} .

A grid study is done for both flat fin and louver fin in maximum inlet velocity by different number of grid cells, as illustrated in Fig. 3. Based on the grid study, grid cell sizes are selected less than 0.2 mm. It should be noted that for keeping $Y^+ < 5$, some cells are adopted near solid wall surfaces.

FIG. 3: Grid study for modeling of (a) flat fin and (b) louver fin

2.3 Parameter Definitions

In order to consider the effect of properties of fluids in Stanton number, this nondimensional parameter is modified to a new nondimensional parameter, Colburn factor (j), which is almost independent of fluid properties for $0.5 \leq \text{Pr} \leq 10$. Colburn factor is usually used for illustration of heat transfer performance (Shah and Sekulic, 2003). Colburn j factor can be described as follows (Wang et al., 1996):

$$j = \frac{\text{Nu}}{\text{Re}_{D_c} \cdot \text{Pr}^{1/3}} \quad (7)$$

where Prandtl number is denoted by Pr. Reynolds number, Re_{D_c} , which is defined based on tube collar outside diameter (D_c), can be determined from the following correlation (Wang et al., 1996):

$$\text{Re}_{D_c} = \frac{\rho V_{max} D_c}{\mu} \quad (8)$$

where V_{max} , ρ , and μ are maximum velocity, density, and dynamic viscosity of air, respectively.

Nusselt number (Nu) is defined based on tube collar outside diameter (D_c) as $\text{Nu} = h D_c / k_f$ where k_f and h are fluid thermal conductivity and heat transfer coefficient, respectively. Since all solid surfaces have constant temperatures, surface efficiency (η_o) is related to fin efficiency (η) and shall be considered for heat transfer rate ($\dot{Q}''$). Therefore, Nusselt number (Nu) can be attained from numerical simulations in accordance with the following correlation:

$$\text{Nu} = \frac{\eta_o \dot{Q}'' D_c}{\Delta T_{lm} \cdot k_f} \quad (9)$$

where ΔT_{lm} is logarithmic mean temperature difference, which can be obtained from the following formula (Nemati and Moghimi, 2014; Shah and Sekulic, 2003):

$$\Delta T_{lm} = \frac{T_{out} - T_{in}}{\ln[(T_s - T_{in}) / (T_s - T_{out})]} \quad (10)$$

where constant wall surface temperature is denoted by T_s and T_{out} and T_{in} represent air outlet and inlet temperatures, respectively. Surface efficiency (η_o) is described in accordance with the following correlation (Wang et al., 1996):

$$\eta_o = 1 - \frac{A_f}{A_o} (1 - \eta) \quad (11)$$

where A_o represents total surface area and A_f is fin surface area. Fin efficiency (η) is determined by the following formula (Nemati and Samivand, 2014):

$$\eta = \left[\frac{\tanh(m(r_f - r)\psi)}{m(r_f - r)\psi} \right]^\psi \quad (12)$$

where r and r_f are tube radius and equivalent circular fin radius, respectively. m is a thermo-geometry parameter and can be calculated as follows (Nemati and Samivand, 2014):

$$m = \sqrt{\frac{2h}{k_{fin}t}} \quad (13)$$

where h is fluid heat transfer coefficient and k_{fin} and t are fin thermal conductivity and thickness, respectively. ψ is fin efficiency parameter, which can be obtained from the following equation (Nemati and Samivand, 2014):

$$\psi = 1 + 0.17912 \ln(r_f/r) \quad (14)$$

where r and r_f are tube radius and equivalent circular fin radius, respectively and r_f/r can be determined from the following equation (Wang et al., 1996):

$$r_f/r = 1.27 (P_t/2r) \left[\left(0.5 \sqrt{P_l^2 + (P_t/2)^2} \right) / (P_t/2) - 0.3 \right]^{1/2} \quad (15)$$

where P_t and P_l are transversal and longitudinal tube pitches, respectively.

Friction factor (f) can be calculated as follows based on Kays and London's definition (Kays and London, 1984):

$$f = \frac{A_c}{A_o} \frac{\rho_m}{\rho_{in}} \left[\frac{2\rho_{in}\Delta P}{G^2} - (K_c + 1 - \sigma^2) - 2 \left(\frac{\rho_{in}}{\rho_{out}} - 1 \right) + (1 - \sigma^2 - K_e) \frac{\rho_{in}}{\rho_{out}} \right] \quad (16)$$

where σ is minimum to frontal flow area ratio. A_o represents total heat transfer area, and A_c is flow cross-sectional area. K_c and K_e , which are assumed zero in calculations, are entrance and exit loss coefficients, respectively. ρ_{in} , ρ_{out} , and ρ_m are air densities at inlet, outlet, and mean value, respectively. ΔP is pressure drop, while air mass flux is denoted by G .

2.4 Model Validation

Experimental data obtained by Wang and colleagues for fin-and-tube heat exchangers having plain fin (Wang et al., 1996) and louver fin (Wang et al., 1998a) with circular tubes is used for validation of the numerical simulations. Geometrical parameters are specified in Table 2. Comparisons of calculated Colburn (j) and friction (f) factors from numerical simulations of flat and

louver fin-and-tube heat exchangers with experimental data are shown in Figs. 4 and 5, respectively. The mean deviations for j and f factors for both flat and louver fins in comparison with the experimental results are specified in Table 3. Also, the reported experimental uncertainties for j and f factors for plain fin (Wang et al., 1996) and louver fin (Wang et al., 1998a) are listed in Table 3. The results show the applicability of the present numerical simulations.

3. PARAMETRIC STUDY

Parametric studies for investigation of effects of louver angle (θ), fin pitch (F_p), transversal tube pitch (P_t), and longitudinal tube pitch (P_l) on Colburn factor (j) and friction factor (f) of louver fin-and-tube heat exchangers were performed. Basic geometrical parameters were the same as those in Table 2.

FIG. 4: Comparison of numerical results with experimental data from Wang et al. (1996) for flat fin

FIG. 5: Comparison of numerical results with experimental data from Wang et al. (1998a) for louver fin

TABLE 3: Reported experimental uncertainties and validation mean deviations

Parameters	Reported experimental uncertainties		Validation mean deviations	
	Flat fin (Wang et al., 1996)	Louver fin (Wang et al., 1998a)	Flat fin	Louver fin
Colburn factor	3.9%–9.4%	5.2%–20.2%	28%	13%
Friction factor	1.3%–17.7%	7.3%–23.4%	10%	30%

3.1 Effect of Louver Angle

In order to examine the effect of louver angle on the heat transfer and friction factor of louver fin-and-tube heat exchangers, louver angle was raised from 0° (flat fin) to 80° and other parameters were fixed as tabulated in Table 2. u_{in} is changed from 1 to 6.5 m/s with corresponding Reynolds number ranging from 1248 to 8111. Figure 6 shows j and f factors vs. louver angles in different Reynolds numbers. The tendency of j factor vs. louver angle is characterized into three stages. Initially, the j factor increases with the rise of louver angle up to a certain threshold value (around 20°), followed by a slight level-off or decline with a louver angle toward 45° , and finally a slight uplift is seen when surpassing the 45° louver angle. In the first stage, it is clear that j factor increases linearly (approximately with slope of 3% for $Re = 1248$) with variation of louver angle (θ) from 0° to 20° , followed by a detectable decline when the louver angle is raised from 20° to 45° . Finally, it increases linearly again (approximately with slope of 1% for $Re = 1248$) from louver angle 45° to 80° . This phenomenon persists for all Reynolds numbers (from 1248 to 8111). It is quite clear that the Colburn factor of louver fin, regardless of louver angle, is much higher than the Colburn factor with plain fin (louver fin with $\theta = 0^\circ$). It is illustrated that the slopes of variations of Colburn factor in the higher Reynolds number regime are less pronounced than the one in lower Reynolds numbers. Also, it is quite clear that with the rise in louver angle from $\theta = 0^\circ$ to $\theta = 60^\circ$, friction factor increases rather slightly (with a very small slope of 1%) for all Reynolds numbers. However, once the louver angle surpasses 60° , the louvers act as obstacles that block the airflow considerably, and thereby causing a surging rise of friction factor. Moreover, it is indicated that Colburn (j) factor and friction (f) factor decrease with increase

FIG. 6: Effect of louver angle on j and f factors in different Reynolds numbers

in Reynolds numbers in the same louver angle. Both heat transfer ($\dot{Q}''$) and pressure loss (ΔP) increase with increase in Reynolds number, but the amount of increase in Reynolds number is much higher than increase in heat transfer ($\dot{Q}''$) and pressure loss (ΔP). Consequently, j and f factors decrease.

For further elaboration of the influence of louver angle on Colburn factor, air flow temperature contours with velocity vectors and 3D pathlines on the left symmetry plane of Fig. 2 for four fin rows are shown in Figs. 7 and 8 subject to different louver angles ranging from $\theta = 0^\circ$

FIG. 7: Air flow temperature contours with pathline for louver angles from 0° to 20° for four fin rows: (a) louver angle 0° ; (b) louver angle 10° ; (c) louver angle 20°

FIG. 8: Air flow temperature contours with pathline for louver angles from 30° to 60° for four fin rows: (a) louver angle 30°; (b) louver angle 45°; (c) louver angle 60°

to $\theta = 60^\circ$ at an air inlet velocity of 1 m/s. It is clear that air flow in zero louver angle (plain fin) is somehow similar to the developing flow between two parallel plates [Fig. 7(a)]. Increasing the louver angle to 10° [Fig. 7(b)], some airflow is directed by the louver to mingle the airstream of the adjacent channels. However, even though the airflow can flow across the louver to fulfill appreciable mixing when compared with the plain, there is only a small fraction of flow that can go through the louvers, so the penetration of directed airflow is less intense. The results indicate that the flow pattern is more like a channel flow rather than fin directed flow. For louver fin and

flat-tube heat exchangers, the flow visualization conducted by Webb and Trauger (1991) concluded that increase of louver angle increases the flow efficiency, which is defined as the actual transverse distance of the airflow divided by the ideal transverse distance of the airflow. Hence, the boundary-layer renewal becomes more pronounced for offering more effective mixing with further rise of louver angle. With the increase in louver angle to 20° [Fig. 7(c)], the flow pattern is disturbed significantly, and a considerable fraction of flow passes through the louvers and penetrates into more fin channels, suggesting a higher flow efficiency. This change in flow pattern increases heat transfer and consequently the Colburn j factor. In fact, louver angle around 20° shows the optimum performance for flow patterns from parallel flow to the disturbed flow. The performance peak for a louver angle of 20° is also reported by Čarija et al. (2014). For louver angle from 20° to 60° , the alignment of louvers and consequently the alignment of passages between louvers are important. Louvers in different fin rows become more aligned when varying the louver angle from 20° to 45° [Fig. 8(a)] where the air flows alongside the louver direction in parallel, and the mixing alongside the louver direction is actually decreasing, which results in a level-off or decreasing trend of Colburn j factor. For a louver angle of 45° [Fig. 8(b)], the flow goes along louvers after impinging to the first louver and since all louvers are completely aligned, the boundary layer grows and consequently heat transfer and Colburn factor decrease [i in Fig. 8(b)]. For a louver angle of 60° [Fig. 8(c)], the airflow passes the louver and hits the louver located at the neighboring fin in a staggered manner, forcing the redevelopment of the boundary layer. As a consequence, heat transfer performance is augmented. Yet, the staggered manner of the louvers in the nearby fins also leads to a rise in maximum flow velocity across the louver. Note that some parts of this airflow impinge onto the first louver and incur noticeable flow separation [i in Fig. 8(c)]. This separation causes other parts of flow to flow upward between the second and the third louvers of the second upper fin row. Velocity of flow in this part is much higher while this higher velocity causes more heat transfer and increases the Colburn j factor accordingly [ii in Fig. 8(c)].

In essence, increases in the Colburn j factors when the louver angles exceed 45° are mainly attributed to the staggered configuration regarding the arrangement of louvers in nearby fins. Small vorticities generated in top of the first half-louver increases the heat transfer slightly while vortices generated between the last louver and tube are increased with the rise of louver angle, which also promotes heat transfer for speeding up the maximum velocity.

3.2 Effect of Fin Pitch

To investigate the effects of fin pitch (F_p) on j and f factors of louver fin-and-tube heat exchangers, fin pitch is increased from 2.5 mm to 5 mm. Longitudinal tube pitch is constant at 15 mm and louver angle is raised from $\theta = 0^\circ$ to $\theta = 60^\circ$. Other geometrical parameters are the same as listed in Table 2. From Fig. 6, an air inlet velocity of 1 m/s yields the maximum j and f factors in the range of 1–6.5 m/s. Hence, the examination of the effects of fin pitch on these factors is mainly conducted at an inlet velocity of 1 m/s. The maximum possible louver angle is limited by the fin pitch. Therefore, the maximum louver angle is less than 60° in some cases. Figure 9 shows j and f factors vs. louver angle. Similarly, as the influence of louver angle, the results for the Colburn j factor can be categorized into three stages while friction factors are termed with two regions. However, the influence of fin pitch is rather small. This is applicable either for the Colburn j factor or friction factor. The results are in line with the test results for louver fin-and-tube heat exchangers without redirection louver by Wang et al. (1998a) and louver fin-and-tube heat exchangers with redirection louver by Wang et al. (1998b).

FIG. 9: Effect of F_p on j and f factors in different louver angles

3.3 Effects of Transversal and Longitudinal Tube Pitches

The effects of transversal (P_t) and longitudinal (P_l) tube pitches on j and f factors of louver fin-and-tube heat exchangers were studied further. For this purpose, the transversal tube pitch was fixed at 25.4 mm and values of 15 mm, 19.05 mm, and 30 mm were selected as longitudinal tube pitch. The simulations were then followed by changing transversal tube pitch from 25.4 mm to 40 mm while fixing longitudinal tube pitch at 19.05 mm. Louver angle was changed from $\theta = 0^\circ$ to $\theta = 60^\circ$ and $u_{in} = 1$ m/s, while other geometrical parameters are listed in Table 2. j and f factors in different louver angles for different transversal and longitudinal tube pitches are shown in Figs. 10 and 11, respectively. As seen in the figures, it is clear that raising either longitudinal or transversal tube pitches imposes impact on the Colburn j factor. The first reason for this decline with the increase in either of pitches may impair the fin efficiency and reduce the effective temperature difference between airflow and the fin/wall surfaces. The second reason is decrease of mixing intensity for the hot fluid near the surface and the cold fluid far from the

FIG. 10: Effect of P_t on j and f factors in different louver angles

FIG. 11: Effect of P_l on j and f factors in different louver angles

surface since louver regions are comparatively distant away from each other. Decrease in j factor from $P_l = 15$ mm to $P_l = 30$ mm increases from a louver angle of 0° (plain fin) to a louver angle of 60° . This phenomenon can also be seen for changing the transversal tube pitch from $P_t = 25.4$ mm to $P_t = 40$ mm.

For further investigation of the effects of transversal and longitudinal tube pitches on friction factor, pressure contours related to Figs. 10 and 11 for a louver angle of 20° in the $y = 0.3$ mm plane are shown in Fig. 12. As illustrated in Fig. 12(a), with increase in longitudinal tube pitch, the effect of the louvers on friction factor will not change while a decrease of the ratio of the louver area to the nonlouver area of plain fin causes a relatively small decrease in friction factor. Consequently, friction factor decreases with the rise in longitudinal tube pitch.

For different transversal tube pitches, variations of friction factor are very small for louver angles from 0° to 20° . As mentioned earlier, the effects of louvers are comparatively weak because the major airflow is flowing on the fin channel rather than in the louvers. Friction factor decreases with the increase in transversal tube pitch for louver angles from 20° to 60° . As shown in Fig. 12(b), when louver angle exceeds 20° , smaller fraction of inlet fluid passes through the louvers with increase in transversal tube pitch. Also, pressure losses upon the tubes are lessened. Consequently, significant decrease in friction factor occurs.

It is clear that the effect of transversal tube pitch (P_t) on friction factor is more pronounced than the effect of longitudinal tube pitch (P_l). The main reason is due to the contraction/expansion between adjacent tubes. Furthermore, two different zones can be seen in f factor for both transversal and longitudinal tube pitches, the same as in Fig. 6.

4. OPTIMIZATION BY FULL FACTORIAL METHOD

In the full factorial method, some desirable values should be selected for each parameter and numerical simulations must be performed for all combinations of parameters. So, all parameters participate in the optimization. Based on the parametric studies, fin pitch imposes rather small effects on heat transfer and friction factor. Hence, fin pitch will not be included in the optimization. Transversal tube pitch, longitudinal tube pitch, louver angle, and air inlet velocity are the major parameters the optimizations, and the related parameters for optimization are tabulated in Table 4.

FIG. 12: Static pressure contours in different (a) longitudinal and (b) transversal tube pitches in louver angle of 20° in plane of $y = 0.3$ mm

A total of 112 numerical simulations were performed by combination based on Table 4. Optimizations of j and f factors were carried out using numerical simulation data by full factorial method. The optimum criteria were maximizing Colburn factor while minimizing friction factor whenever possible. The final results for optimization are shown in Fig. 13. This figure consists of

TABLE 4: Parameter values used for optimization

Parameter	Values used for optimization						
Transversal tube pitch (P_t) (mm)	25.4	30	—	—	—	—	—
Longitudinal tube pitch (P_l) (mm)	15	19.05	—	—	—	—	—
Air inlet velocity (V) (m/s)	1	2	4	6.5	—	—	—
Louver angle (θ) (deg.)	0	10	20	30	45	60	80

FIG. 13: Results of optimization by full factorial method

four columns and three rows, which are separated by grey lines. Each column is assigned to one of the studied parameters, which is mentioned as the header of that column, while the upper and lower values are shown under each parameter. For optimization purpose, it is essential to attain the maximum Colburn factor and minimum friction factor, simultaneously, which is defined as *composite desirability*. In this regard, there may be more than one optimum working condition, the degree of optimization can be judged by composite desirability factor. The objective is to approach this factor toward unity as the degree of optimization. The composite desirability of each working point for each parameter (based on Table 4) is shown in the first row while it is related to j and f factors, and the calculated optimal values of other parameters are indicated in the second and third rows, respectively. For example, there are four dots in the velocity column (third column), meaning four values are selected for velocity in the range of 1 to 6.5 m/s which is based on Table 4 and the calculated in optimal values of P_t , P_l , and θ .

As shown in the figure, the maximum Colburn factor is 0.0330 and the minimum friction factor is 0.1102, which are indicated by blue dash lines. Accordingly, the working points are among those candidates where the maximized Colburn factor and the minimized friction factor

are indicated by vertical solid red lines, and its value is also mentioned in red in the header of each column. Further information regarding the fundamentals of the full factorial method can be found in Cavazzuti (2013).

As mentioned earlier, optimum values for friction factor (f) and Colburn factor (j) are 0.1102 and 0.0330, respectively, which can be obtained by a transversal tube pitch (P_t) of 25.4 mm, a longitudinal tube pitch (P_l) of 15 mm, an air inlet velocity (V) of 1 m/s, and a louver angle (θ) of 20° . The calculated optimum louver angle is in accordance with the optimized louver angle obtained by using the genetic algorithm in Sadeghianjahromi et al. (2018a). Optimization in this paper has been performed using a full factorial method, in which all of the parameters used in parametric study are directly participating in optimization. However, the genetic algorithm, which was used by Sadeghianjahromi et al. (2018a) for optimization, needs an input function and they used the developed correlations based on their parametric study as algorithm input, while their developed correlations have mean deviations of 2.19% and 7.20% for j and f factors, respectively, which can cause some errors and deviations in optimization.

5. CONCLUSIONS

A detailed parametric study was carried out regarding the fin-and-tube heat exchanger having louver fin configuration. The effects of louver angle, fin pitch, transversal tube pitch, and longitudinal tube pitch on heat transfer performance and friction factor were investigated. Moreover, Colburn and friction factors were optimized using numerical results. All in all, the following conclusions are drawn:

- The Colburn j factor increases linearly with louver angle variation from 0° to 20° , followed by a level-off or slight decline when the louver angle is less than 45° , and finally the level-off trend of the Colburn j factor is reversed and rising when louver angle exceeds 45° . The slopes of increase or decrease of j factor decrease with increase in Reynolds number.
- Friction factor increases linearly (approximately with slope of 1%) with louver angle increase from $\theta = 0^\circ$ to $\theta = 60^\circ$. However, louvers act as obstacles against fluid flow when louver angles surpass 60° , thereby causing significant increase in friction factor.
- j and f factors are almost independent of fin pitch in the range from 2.5 to 5 mm and $Re_{D_c} < 2000$.
- j and f factors decrease with an increase in transversal and longitudinal tube pitches. For different transversal tube pitches, variations of friction factor are not so significant for louver angles from 0° to 20° . The influence of transversal tube pitch on friction factor is more pronounced than that of longitudinal tube pitch, especially when the louver angle exceeds 20° .
- The optimum louver angle for maximizing j and minimizing f is obtained around 20° by full factorial method.

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