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ABSTRACT

Recent developments in materials have opened opportunities for non-metallic materials to be used for constructing heat exchangers. In particular, polymers can be superior to conventional metals in chemical resistance, weight, and material cost. While polymers generally have a very low thermal conductivity and are limited in mechanical strength and temperature, these drawbacks can be alleviated by reducing the characteristic lengths of the heat exchanger core. In this paper, the technical potential of a polymer-tube-bundle heat exchanger for liquid-to-gas applications is assessed. Design parameters of the polymer heat exchanger are determined to match the thermal-hydraulic performance of a conventional metallic heat exchanger under the same operating conditions. Results show that the polymer heat exchanger can achieve significantly reduced core weight and volume as well as material cost.

1. INTRODUCTION

Heat exchangers using single-phase liquids in HVAC&R applications are typically constructed of metals. These applications include secondary-loop supermarket display cases, fan-coil units for comfort heating and air conditioning, and automotive heat exchangers. While metallic heat exchangers have good mechanical properties and provide excellent thermal performance, metals are heavy, relatively expensive, and susceptible to surface fouling and corrosion. As a potential alternative, polymers can overcome some of the shortcomings of metals. However, polymers generally have a very low thermal conductivity (less than 0.5 W/m-K) and other limitations. Nevertheless polymers can be suitable heat exchanger materials, especially for single-phase liquid applications.

Major benefits of polymers as heat exchanger materials are: (1) reduced weight and cost, (2) manufacturing flexibility for complex designs, and (3) chemical stability. The main weaknesses of polymers in comparison to more conventional heat exchanger materials (*i.e.* metals and alloys) are low thermal conductivity and strength. If the metallic materials of a heat exchanger are simply replaced by polymers without any design modifications, a significant increase in the overall heat transfer resistance will result. For this reason, polymer heat exchangers can require a larger surface area (and volume) than the conventional metallic counterparts unless the design is changed. In addition to the low thermal conductivity, comparatively lower limits of operating temperature or mechanical strength are the weaknesses of polymers. The use of polymer-based composites can mitigate these thermal and structural issues and improve thermal conductivity. However, such composite materials may not be as attractive in terms of material cost and manufacturing, and their usage should be justified by other benefits such as reduced weight or chemical stability.

The low thermal conductivity of polymers has two practical implications for polymeric heat exchanger designs: (1) the use polymers as primary heat transfer path should be minimized, i.e. walls for heat conduction (e.g. tube or plate) should be made thin, and (2) long-distance heat conduction (e.g. fins) through polymers should be avoided. Thus, instead of relying on conduction through extended surfaces, a polymer heat exchanger design should increase the primary (i.e. base) heat transfer surface area. The needs for reduced conduction resistance and higher mechanical strength apparently conflict with each other. The limitations in mechanical strength can be resolved by adopting small length scales, i.e. instead of using large flow channels with thick walls which degrade thermal performance, the flow can be divided into multiple smaller channels with thinner walls. Due to the manufacturing flexibility, polymers are particularly suitable for producing a highly compact, multi-mini-channel heat exchanger design.

Polymer heat exchangers have been most widely used in the chemical processing industry, where the heat transfer fluids are sometimes highly corrosive to conventional metals. Polymeric heat exchangers can be found in liquid-to-liquid, liquid-to-gas, and gas-to-gas heat transfer applications (Reay, 1989; Zaheed and Jachuck, 2004). The thermal-hydraulic performance of heat exchangers for liquid-to-liquid applications, made of polymer tube bundles or polymer hollow fibers, has been previously investigated (Liu *et al.*, 2000; Zarkadas *et al.*, 2005). These studies targeted chemical-reactor applications, motivated by the advantages of polymers in corrosion resistance and weight reduction. They demonstrated the potential superiority of polymer heat exchangers. However, the effects of polymer tube-bundle design parameters have not been fully explored in that prior work, and quantitative modeling of the performance of polymer-tube-bundle heat exchangers for liquid-to-gas applications is not found in the literature.

In this paper, we will evaluate the potential of polymers to replace metallic heat exchangers by considering thermal-hydraulic performance, mechanical strength, size, weight, and material cost. To manage the conduction thermal resistance of the polymer, a thin-walled tube bundle without fins will be adopted instead of the conventional fin-and-tube configuration. An analytical method will be developed to model the tube bundle geometry. Important performance requirements, i.e. heat transfer rate, pumping power, and mechanical strength, will be prescribed and met by tuning the tube bundle geometry. An attempt was made to produce an optimal geometry, but it was found that all the geometrical parameters had monotonic effects on the selected figures of merit as discussed later.

2. HEAT EXCHANGER CONFIGURATION

2.1 Baseline: Conventional Metallic Heat Exchanger

As a baseline reference for a liquid-to-gas application, a conventional metallic heat exchanger was selected from the literature (Wang *et al.*, 1998) as shown in Figure 1. The baseline is considered to have louvered aluminum fins and copper tubes. The geometrical parameters of this baseline are given in Table 1.

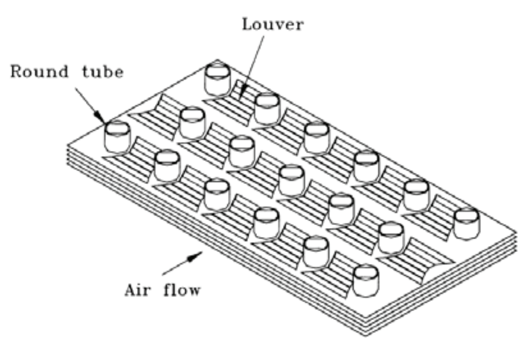


Figure 1: Louvered fin-and-tube heat exchanger (Figure from Wang et al. (1998))

Table 1: Specifications of louvered fin-and-tube heat exchanger (baseline)

Parameter	Value	Parameter	Value
Fin pitch (mm)	2.08	Transverse tube pitch (mm)	25.4
Louver pitch (mm)	2.4	Longitudinal tube pitch (mm)	19.05
Louver height (mm)	1.4	Number of tube rows	4
Fin thickness (mm)	0.115	Core with (mm)	600
Fin collar diameter (mm)	10.42	Core height (mm)	355
Tube thickness (mm)	0.35		

2.2 Alternative: Polymer-Tube-Bundle Heat Exchanger

From the earlier discussion, the proposed liquid-to-air heat exchanger uses a polymer tube bundle without fins. In order to reduce tube wall conduction resistance, the tubes should be made sufficiently thin while maintaining the structural integrity under normal operating conditions. As typical heat exchangers operate with pressurized liquid inside the tubes, the pressure and the operating temperature can become the limiting factor for the polymer tubes.

Raman *et al.* (2000) conducted a careful examination a number of polymeric materials for use in solar collectors. They selected several candidate materials by screening with national codes and standards for plumbing applications. Combining the thermal and mechanical limits of individual materials, they were able to calculate the minimum tube thickness for a given tube diameter. In summary, they proposed using large number of thin, small-diameter polymer tubes for solar collectors. They suggested using extruded polymer tubes and fiber-reinforced polymer composite headers. In a related study (Liu *et al.*, 2000), prototype designs of shell-and-tube and immersed-tube-bundle heat exchangers were modeled with the consideration of heat duty and thermal/mechanical properties.

We will take a similar approach by adopting a large number of small and thin polymer tubes as a liquid-to-gas heat exchanger. For a given pressure, the tube diameter will be determined such that sufficient mechanical strength and reasonably low wall conduction resistance are attained. The basic configuration of the polymer-tube-bundle heat exchanger and the design parameters are shown in Figure 2, where the heat exchanger has a staggered tube bundle in a cross-flow configuration. Important geometrical parameters include tube diameter, wall thickness, transverse and longitudinal tube pitches, tube length, core height, and number of tube rows. Depending on the overall core dimensions, multiple cross-flow "modules" can be stacked parallel or perpendicular to the gas-flow direction. Table 2 gives essential material properties of selected metals and polymers that will be used for the performance modeling in the next section. The two polymers were identified as the best candidates for tube materials in solar water heating systems by Raman *et al.* (2000).

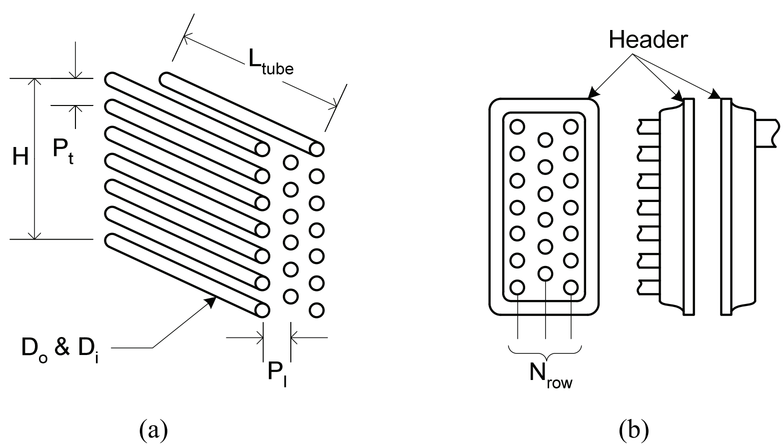


Figure 2: Design parameters of polymer-tube-bundle heat exchanger; (a) tube bundle, (b) header

Table 2: Properties of selected metals and polymers

Parameter	Copper	Aluminum	HTN	PEX
Density (kg/m^3)	8918	2696	1130	952
Thermal conductivity (W/m-K)	398	237	0.31	0.38
Long-term tensile strength* (MPa)			12.6	2.8
Material cost (US$/kg)	8.3	2.9	7.7	3.2

* Estimated at 82°C (Raman *et al.*, 2000)
HTN: High temperature nylon, PEX: Cross-linked polyethylene

3. PERFORMANCE MODELING AND ANALYSIS

Empirical correlations from the literature were used to predict the air- and tube-side heat transfer and pressure drop for the baseline heat exchanger and the polymer-tube-bundle heat exchanger. All correlations were formulated as closed-form mathematical expressions which could be evaluated for any given set of input parameters. Care was taken to ensure that the present modeling is conducted within the original parameter space for each correlation. The rest of the present analysis used water and air as the heat exchange fluids.

3.1 Baseline: Conventional Metallic Heat Exchanger

The airside performance was predicted from correlations by Wang *et al*. (1999) for louvered plate-fin-and-tube heat exchangers. Their correlations predicted 95.5% and 90.8% of the *j* and *f* data in the original database within ±15%. Colburn *j* and Fanning friction factors are defined below. The selected baseline geometry in Table 1 was taken from the samples in the original database, and therefore reasonable accuracy is expected in the present modeling. The correlation equations have been omitted here. The methods of the tube-side calculations are essentially the same as those for the polymer-tube-bundle heat exchanger described in the next section.

$$j = \frac{Nu}{RePr^{1/3}} \tag{1}$$

$$f = \frac{A_c \rho_{ave}}{A_{tot}\rho_1}\left[\frac{2\rho_1 \Delta P}{\rho_{ave} u_c^2} - \left(1+\sigma^2\right)\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \tag{2}$$

3.2 Alternative: Polymer-Tube-Bundle Heat Exchanger

For given system variables (i.e. inlet temperatures, mass flow rates), the liquid-side heat transfer and pressure drop were calculated from the Nusselt number and Darcy friction factor correlations for fully-developed laminar and turbulent pipe flows (Gnielinski, 1976; Incropera and DeWitt, 1996; Petukhov, 1970) . The air-side heat transfer and pressure drop were estimated from closed-form correlations for the Nusselt number, Leveque number (*Lq*), and Hagen number (*Hg*), as summarized by Martin (2002). Once air-side and water-side convective heat transfer coefficients were obtained, an effectiveness-NTU relation was used to calculate heat transfer rate. The present analysis used an effectiveness-NTU relation for a single-pass, cross-flow configuration with both fluids unmixed (see equations (3)-(10)). The minimum thickness of polymer tube was determined such that the tube hoop stress induced by the differential pressure does not exceed the tensile strength (see equation (11)).

$$Nu_i = \frac{h_i D_i}{k_l} \tag{3}$$

$$f_i = \frac{D_i}{L_{tube}} \frac{2\Delta P_i}{\rho_l u_l^2} \tag{4}$$

$$Nu_a = \frac{h_a D_o}{k_a} = 0.404 Lq^{1/3} \tag{5}$$

$$Lq = 0.92 Hg Pr_a\, function\left(X_t, X_l, X_d\right) \tag{6}$$

$$Hg = \frac{\Delta P D_o^2}{\rho N_l \nu_a^2} = function\left(Re_a, X_t, X_l, N_l\right) \tag{7}$$

$$\frac{1}{UA_{tot}} = \frac{1}{h_l A_l} + \frac{\ln\left(D_o / D_i\right)}{2\pi k_{tube} L_{tube} N_{tube}} + \frac{1}{h_a A_{tot}} \tag{8}$$

$$\varepsilon = \frac{Q}{C_{min}\left(T_{l1} - T_{a1}\right)} = 1 - \exp\left\{C_r^{-1} NTU^{0.22}\left[\exp\left(-C_r NTU^{0.78}\right) - 1\right]\right\} \tag{9}$$

$$NTU = UA_{tot} / C_{min}\ ; \qquad C_{min} = \min\left(\dot{m}_a c_{pa}, \dot{m}_l c_{pl}\right) \tag{10}$$

$$\delta_{tube} \geq \frac{D_i}{2\sigma / P_i - 1} \tag{11}$$

4. RESULTS AND COMPARISON

For the selected baseline heat exchanger geometry, if air and water velocities are varied, the air-side and the water-side pressure drops as well as the heat transfer rate change. The performance evaluation criteria used in this section are based on a matching performance—i.e. the design parameters of polymer-tube-bundle heat exchanger are determined such that the same heat transfer rate and air/water pressure drop (or pumping power) as those for the baseline heat exchanger are achieved under the same system variables (mass flow rates, inlet fluid temperatures). Thus, the two heat exchangers are tuned so that they perform exactly the same function.

In Figure 3, air face velocity for the baseline sample is varied under a constant tube internal pressure, tube pitch ratios, and water velocity in the baseline tubes. The plot presents the contribution tube-wall conduction resistance to the total heat transfer resistance for the polymer tube bundle heat exchanger. Clearly, decreasing tube diameter results in reduced contribution from tube-wall conduction resistance due to the thinner tube wall. For the parameter space in the figure, it was found that the air-side convective heat transfer resistance is the largest—the increasing wall conduction resistance contribution is simply due to a decreasing denominator (R_{total}) with increasing air velocity. The discontinuities of the plotted slopes in the figure (also in other figures) are due to transitions between separate correlations within the range of flow rates.

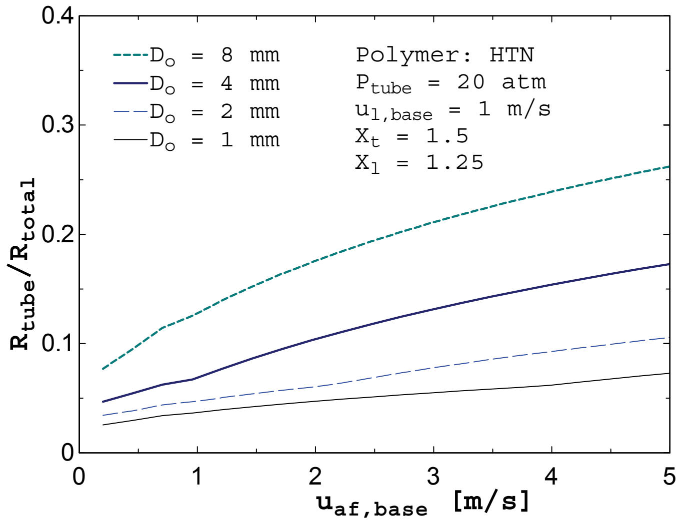


Figure 3: Fraction of total heat transfer resistance by tube wall conduction for polymer heat exchanger

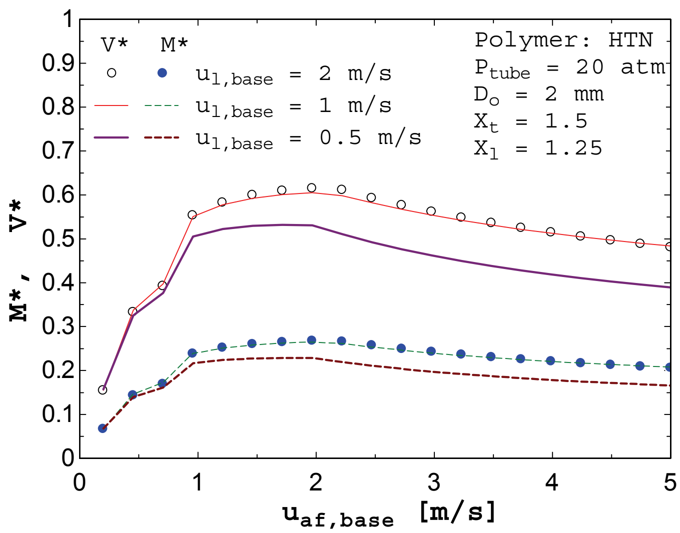


Figure 4: Core volume and mass ratios for HTN tube bundle heat exchanger—effect of air and water velocity

The ratio of polymer heat exchanger core mass to baseline core mass, and a similar ratio of core volume are presented in Figure 4. For the parameter space represented here, the polymer-tube-bundle heat exchanger has significantly lower core mass and volume. Interestingly, the peaks near the baseline air face velocity of 2 m/s correspond to the *least* advantageous condition for the polymer heat exchanger. The effect of baseline water velocity appears negligible above 1 m/s, while a greater superiority is realized at 0.5 m/s. Figure 5 presents the effect of longitudinal and transverse tube pitch ratios on the core mass and volume of polymer heat exchanger. Clearly, reducing both tube pitch ratios results in reduced core mass and volume. However, the longitudinal tube pitch has a greater impact than the transverse tube pitch. Furthermore, core volume is more dramatically affected than core mass. Consequently, some configurations with a large longitudinal tube pitch have a core volume greater than the baseline core.

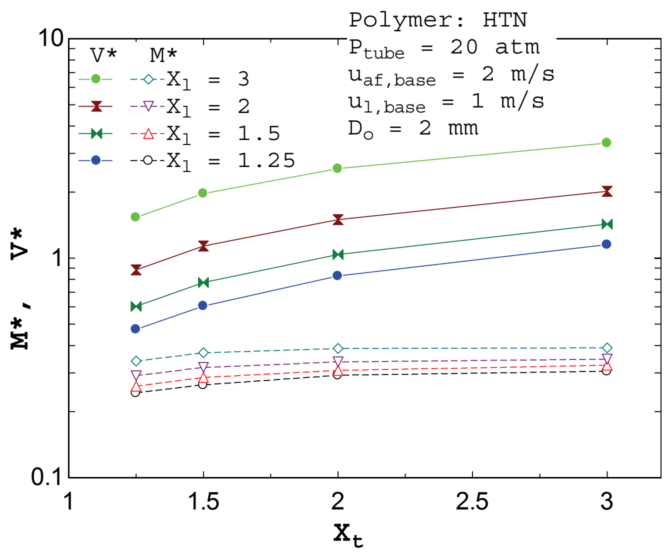
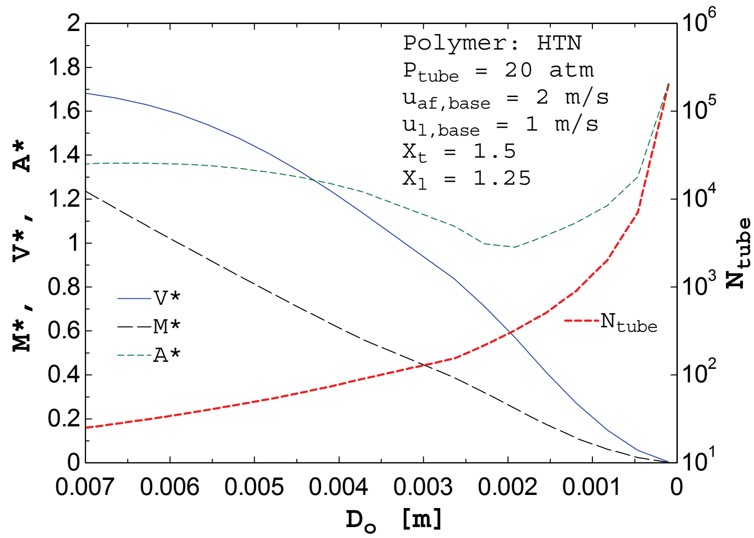


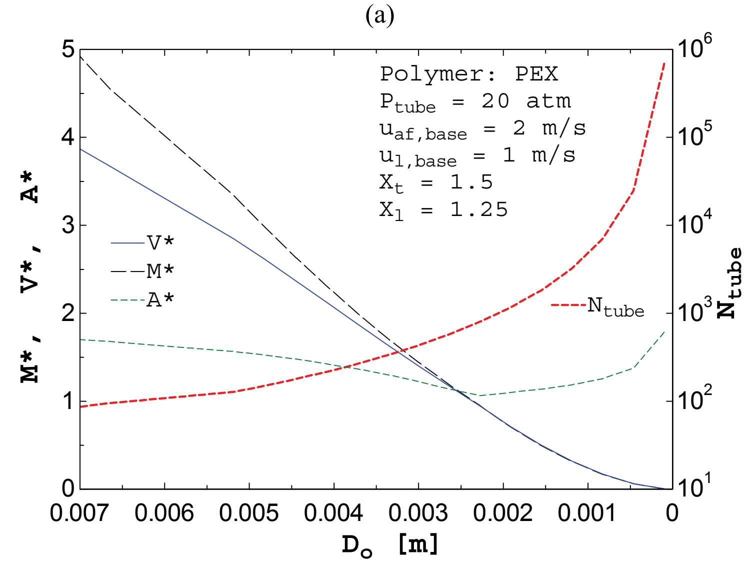
Figure 5: Core volume ratio and core mass ratio for HTN tube bundle heat exchanger—effect of tube pitch ratios

Figure 6(a) and (b) show the impact of tube diameter on various merit characteristics of the polymer-tube-bundle heat exchanger. The core mass is lower than the baseline mass when tube outer diameter is smaller than 6 mm for the HTN tubes, and 2.5 mm for the PEX tubes. In contrast, the volume saving threshold appears at a much smaller diameter (3.5 mm) for the HTN tubes, whereas much less difference exists for the PEX tubes. At a glance, the HTN tube bundle manifests a distinctive superiority over the baseline heat exchanger. On the other hand, the PEX tube bundle is restricted to a much smaller tube diameter to obtain mass and volume savings. The main cause for this is the low tensile strength of PEX tubes. Meanwhile, the material cost of HTN and PEX per unit mass is 123% and 50% of the baseline material, respectively. Thus, for applications where core volume and mass are less important, the PEX tube bundle may be as good or better than the HTN tube bundle, due to the reduced material cost. Another interesting observation can be drawn from the variation of core face area ratio, which shows a minimum at a tube diameter ~2 mm. Contrary to an initial concern that the polymer tube bundle may have a significantly larger face area, the figure shows only a moderate increase of face area for polymer heat exchangers. Another important factor in assessing the viability of the polymer-tube-bundle heat exchanger is the number of tubes in the core. The figure indicates that the number of tubes quickly rises above 1000 for a tube diameter smaller than 1 mm. Although the cost of fabrication has not been included in this study, assembling an extremely large number of thin polymer tubes with headers might be expensive. Overall, for the selected baseline heat exchanger for water-to-air applications, an HTN tube bundle heat exchanger with a tube outer diameter of 2-3 mm appears to be an excellent replacement.

The target applications of the present polymer-tube-bundle heat exchanger can be extended beyond liquid-to-gas forced convection applications. For example, natural convection on the gas side or a two-phase refrigerant flow on the tube side can be considered. Such applications will require specific attention to uniquely related issues such as achieving sufficient flow rate by buoyancy effect or solving problems with refrigerant/gas absorption and permeation into polymer tubes.



(a)



(b)

Figure 6: Core volume, mass, and face area ratios for polymer tube bundle heat exchanger—effect of tube diameter; (a) HTN (high temperature nylon) tube, (b) PEX (cross-linked polyethylene) tube

5. CONCLUSIONS

A polymer-tube-bundle heat exchanger was conceptualized as a potential replacement to a conventional metallic plate-fin-and-tube heat exchanger for water-to-air applications. The lower thermal conductivity and mechanical strength of polymers were systematically overcome by employing a larger number of thin-walled small-diameter tubes. The main findings are listed below:

- The tube-wall conduction resistance of the polymer tube bundle can be significant.
- In order to reduce core volume, mass, and material cost, it is always desirable to have (1) small tube diameter, (2) small tube pitch ratios, and (3) low air and water flow rates.
- An optimal geometry was not found within the parameter space covered in the present study. However, a refined model capable of yielding a design optimum may be obtained by including other factors such as manufacturing cost.
- While HTN (high temperature nylon) tubes can be more effective in reducing core mass and volume, PEX (cross-linked polyethylene) tubes have an advantage from significantly lower specific material cost.

NOMENCLATURE

A	area	(m^2)
A^*	facial area ratio, $A_{af}/A_{af,baseline}$	(–)
c_p	specific heat	(J/kg-K)
C	heat capacity, $\dot{m}c_p$	(J/K)
C_r	heat capacity ratio, C_{min}/C_{max}	(–)
D	diameter	(m)
f	friction factor	(–)
h	convective heat transfer coefficient	(W/m^2-K)
j	Colburn j factor	(–)
k	thermal conductivity	(W/m-K)
$\dot{m}$	mass flow rate	(kg/s)
M^*	core mass ratio	(–)
NTU	number of transfer unit	(–)
N	number of tubes (or tube rows)	(–)
Nu	Nusselt number	(–)
P	tube pitch; gauge pressure	(m; Pa)
ΔP	pressure drop	(Pa)
Pr	Prandtl number	(–)
Q	heat transfer rate	(W)
Re	Reynolds number	(–)
T	temperature	(°C)
u	velocity	(m/s)
U	overall transfer coefficient	(W/m^2-K)
V^*	core volume ratio	(–)
X	tube pitch ratio, e.g. $X_t = P_t/D_o$	(–)
Greek symbols		
δ	thickness	(m)
ε	effectiveness	(–)
η	surface efficiency	(–)
ν	kinematic viscosity	(m^2/s)
ρ	density	(kg/m^3)
σ	tensile strength	(Pa)

Subscripts

1; 2	inlet; outlet
a	air
c	minimum flow passage
af	air facial
ave	average
core	heat exchanger core
d	diagonal
i	tube internal
l	liquid or longitudinal
o	airside total
t	transverse
tot	total

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