OPTIMAL DESIGN OF MICRO BARE-TUBE HEAT EXCHANGER

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ABSTRACT. An optimum design tool based on the simulated annealing technique has been developed for a compact heat exchanger that employs only small diameter tubes without conventional fins, and the performance achieved by the present design is compared to commercially available compact heat exchangers. For all design problems tested, the present heat exchanger design offers a significant degree of improvement in terms of the pumping power, the heat transmission efficiency, and/or the core volume size. The range of tube diameter in the optimum design is found to be 0.3-0.5 mm. Laboratory experiments with a prototype heat exchanger with 0.5 mm OD bare tubes confirm the numerical results.

1. INTRODUCTION

With the growing importance of global environmental problems, efficient energy utilization becomes an ever more urgent target in science and technology. Among diverse elementary techniques to be improved, heat exchanger is one of the major components common in a wide variety of thermal energy handling processes, such as conversion, transport, consumption and storage. Improvement of heat exchanger performance affects both directly and indirectly the performance of various devices and systems, and it would lead to better utility and industrial energy plants, air-conditioning systems, manufacturing processes, transportation systems, and even information devices, all of which should contribute to reduction of emission of greenhouse effect gases. Hence, the present work is aimed at developing an optimum design tool of high-performance heat exchangers. Emphasis is laid upon compact heat exchangers for automobile air-conditioning system.

Generally, heat transfer enhancement is achieved by employing extended heat transfer surfaces, such as louvered fins and offset-fins. Although the modern heat exchanger performance has been admirably increased with these elaborate heat transfer surfaces, it has also resulted in the increased manufacturing cost due to ad-hoc fabricating processes. Moreover, further improvement in the heat exchanger performance seems to be saturated after a great

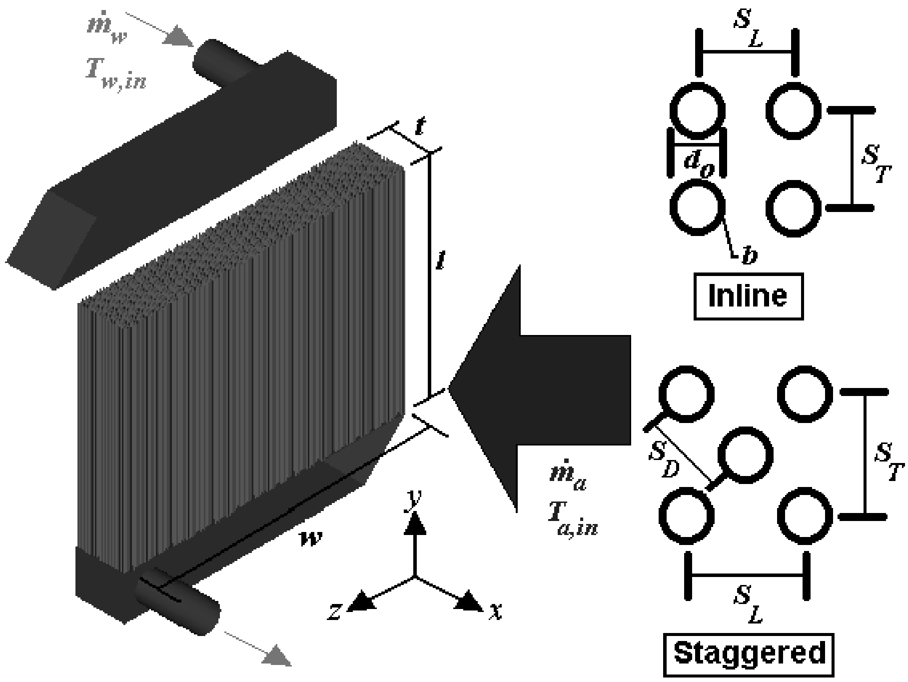

Figure 1. Micro bare-tube heat exchanger.

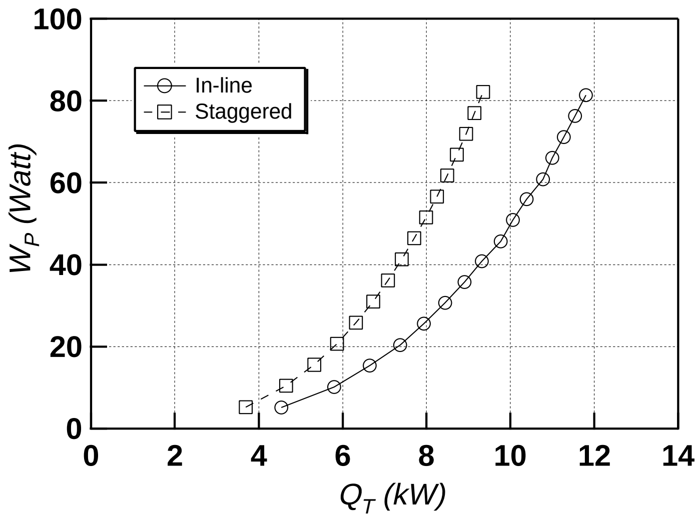

Figure 2. Comparison between in-line and staggered tube arrangements.

deal of industrial effort with this sort of sophisticated extended heat transfer surfaces of complex geometry. Alternatively, one would be aware of the fact that, given a Nusselt number ($Nu = h \cdot L / k$) for the heat transfer system under study, the heat transfer coefficient h itself would simply increase as the characteristic length scale L is reduced. That is, we might conceive a high performance heat exchanger by reducing the hydraulic diameter of its flow passage.

In the present work, we introduce a new design of compact heat exchanger that consists of many small diameter tubes without any extended surfaces, i.e., *micro bare-tube heat exchanger*. As a result, we meet a large pressure drop across the heat exchanger core. Hence, we should compromise the heat transfer augmentation and pressure penalty. To do this, we have developed an optimum design tool for micro bare-tube heat exchanger, and evaluate its performance in several design cases.

2. DESIGN PROCEDURE

The type of heat exchanger we consider in the present work is a single-pass single-phase air-water crossflow heat exchanger utilized for automobile applications. The heat exchanger core is depicted schematically in Fig. 1 with its design parameters. We considered two kinds of tube arrangements, namely, in-line and staggered arrangements as shown together in Fig. 1. Preliminary calculations, however, showed that the in-line arrangement would be superior to the staggered one for small diameter tubes considered here ($100 < Re_{max} < 1000$). Figure 2 shows the pumping powers W_p required to achieve a specified amount of heat exchange Q_T. It is obvious that the pumping power for the staggered tube bank is larger than that for the in-line one. Therefore, the in-line tube arrangement is employed in all calculations in the present paper.

The basic equations for calculations of heat exchange and pressure drop, and the simulated annealing method are given in the following sections.

<u>Heat exchange calculation</u>

In order to calculate the total heat exchange Q_T of heat exchanger, the heat transfer coefficients of water flow inside the tube and air flow across tube bank are estimated by using empirical formulae.

<u>Flow inside tube.</u> Due to very small diameters considered, the tube length to diameter ratio l/d_o is generally very large, so that it is reasonable to assume that the flow inside tube is both hydrodynamically and thermally fully-developed. Since the Reynolds number of this flow can be assumed to be less than 2000, the analytical solution of Nusselt number with isothermal wall boundary condition [6] is employed:

$$Nu_i = \frac{h_i d_i}{k_i} = 3.66, \qquad (1)$$

where h_i, d_i and k_i are the heat transfer coefficient, tube inner diameter, and fluid thermal conductivity, respectively. Note that, we have assumed the constant tube wall temperature and used Eq. (1), which is not strictly

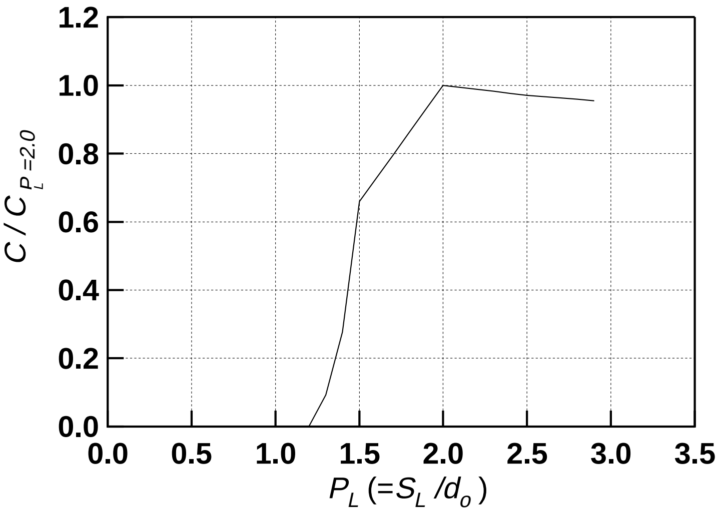

Figure 3. The correction factor for Eq. (2) when Re_{max} < 1000.

satisfied. However, it is still acceptable because the heat transfer resistance in the water side is much smaller than the air side as shown later.

Flow across tube bank. The Nusselt number for flow across the tube bank is estimated with the following correlation equation given by Zhukauskas [2]:

$$Nu_o = \frac{h_o d_o}{k_o} = C \cdot Re_{max}^m \cdot Pr_f^{0.36} \cdot \left(\frac{Pr_f}{Pr_{wall}} \right)^{0.25}, \tag{2}$$

where h_o, d_o and k_o are the heat transfer coefficient, tube outer diameter, and fluid thermal conductivity, respectively, while Pr_f and Pr_{wall} are the Prandtl numbers evaluated at the bulk temperaure and the wall temperature. The Reynolds number Re_{max} is based on the tube outer diameter and the mean velocity at the minimum cross section of the tube bank. Zhukauskas [2] suggested that the values of empirical constants C and m depend only on Re_{max}. However, the use of Eq. (2) at low Re_{max}(<1000) should result in significant overestimation of Nu , if the effect of pitches of tube bank (S_T ,S_L) is not taken into account as can be seen in his experimental data [2]. Therefore, we determine the value of C for various non-dimensional longitudinal piches $P_L(=S_L/d_O)$ from a series of numerical simulations of flow across a tube bank at $P_T\ (=S_T/d_O) = 2.0$ with a commercial CFD software package FLUENT/UNS (Fluent Inc.). Figure 3 shows the correction factor for C obtained for the 10-row tube bank at Re_{max} = 600 [3].

With the heat transfer coefficients of the flow inside the tube and the flow across the tube bank obtained from Eqs. (1) and (2), we can calculate the overall heat transmission coefficient U by the following equation,

$$\frac{1}{U} = \frac{d_o}{d_i} \cdot \frac{1}{h_i} + \frac{1}{h_o} + \frac{d_o}{k} \cdot ln\left(\frac{d_o}{d_i} \right), \tag{3}$$

where k is the tube thermal conductivity. In the present work, pure copper is assumed to be the tube material with k = 401.0 [W/(m·K)] in all calculations.

Finally, the total heat exchange Q_T can be calculated by using the effectiveness-Ntu method [6]. For crossflow heat exchanger with both fluid streams unmixed, it yields

$$Q_T = \left(1 - e^{\left[\frac{1}{C_r} \cdot Ntu^{0.22} \cdot \left[e^{(-C_r \cdot Ntu^{0.78})} - 1 \right] \right]} \right) C_{min} \Delta T_{inlet}, \tag{4}$$

where ΔT_{inlet} is the absolute difference in working fluid inlet temperatures. The quantity C_r is the heat capacity ratio defined as $C_r = C_{min}/C_{max}$, where C_{min} and C_{max} are the smaller and larger values of heat capacities of working fluids, respectively. The Ntu, or number of transfer unit, is defined as $Ntu = UA/C_{min}$, where U is obtained from Eq. (3) whilst A is the heat exchange area of heat exchanger.

Pressure drop calculation

The formulae used for estimating the pressure drop of flow inside the tube and that across the tube bank are summarized below.

Flow inside tube. The core pressure drop of flow inside the tube is calculated by the following formula given by Kays and London [4].

$$\Delta P_{core} = \frac{\rho u_m^2}{2} \cdot \left(K_c + K_e + c_f \frac{l}{d_i} \right), \tag{5}$$

where c_f is the Darcy friction factor (= 64/Re for fully-developed laminar flow), while K_c and K_e are the core entrance and exit pressure-loss coefficients, respectively. Note that, the friction loss is dominant in the core pressure drop for all cases examined. The total pressure drop including the pressure drop at inlet and outlet manifolds is assumed to be three times the core pressure drop ($\Delta P_i = 3\Delta P_{core}$).

Flow across tube bank. For the external flow across the tube bank, the empirical equation proposed by Zhukaukas [2] is employed:

$$\Delta P_o = N_L \cdot \chi \cdot \left[\frac{\rho \cdot u_{max}^2}{2} \right] \cdot f, \tag{6}$$

where the friction factor f and the geometrical correction factor of tube bank χ are given graphically in [2].

Pumping power. The total pumping power can be given as the sum of the pumping power driving the internal (water) and external (air) flows:

$$W_p = \frac{\dot{m}_a \Delta P_o}{\rho_a} + \frac{\dot{m}_w \Delta P_i}{\rho_w}, \tag{7}$$

where $\dot{m}_a$, $\dot{m}_w$, ρ_a , ρ_w are the air and water mass flow rates and their densities, respectively.

Simulated annealing method

With the above set of equations, design parameters are optimized by using the simulated annealing method [5] in such a way that they should minimize a cost function described below. Both the number of temperature steps and the number of random steps per temperature step are chosen as 10000. The temperature of the system is decreased according to a logarithmic annealing schedule with an arbitrary high starting temperature and the terminal temperature set at 3e-8.

3. RESULT AND DISCUSSIONS

In order to evaluate the performance of the present design, we refer to typical data from two commercially available louvered-finned tube compact heat exchangers for comparison as shown in Table 1 [7]. For each set of data, we pose three design problems with different cost functions defined as follows.

Optimization Case 1: Minimization of pumping power

In this case, the cost function is the total pumping power (W_P) defined by Eq. (7). The core dimension (l, w, t), inlet temperature of both working fluids ($T_{a,in}$, $T_{w,in}$), and the total heat exchange (Q_T) are set at the same values as the commerical heat exchanger data, while other five parameters, i.e., the tube outer diameter (d_o), the transverse and longitudinal pitches (S_T, S_L), and the required mass flow rates ($\dot{m}_a$, $\dot{m}_w$) are simultaneously optimized to obtain minimum W_P.

The comparison of the required pumping power between the optimal micro-bare tube and the commercial heat

Table 1. Specification of commercial compact heat exchangers [7].

	Definition	Q_T (kW)	W_P (W)	w (cm)	l (cm)	t (cm)	V (cm^3)	$T_{w,in}$ (^OC)	$T_{a,in}$ (^OC)	$\dot{m}_a$ (kg/s)	$\dot{m}_w$ (kg/s)	ΔP_a (Pa)	ΔP_w (kPa)
H1	Air heater	7.0	35.0	20.0	15.0	3.50	1050	80	15	0.116	0.20	343	2.93
H2	Air heater	6.4	21.0	20.0	15.0	2.50	750	80	15	0.116	0.20	196	7.33

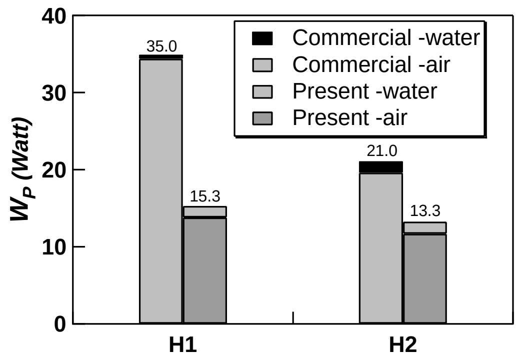

Figure 4. Performance comparison in Case 1.

exchangers is shown in Fig. 4. It is evident that the present design require smaller pumping power than those of the commercial ones, i.e., 63% for H1 and 44% for H2. The optimum tube diameter is found to be around 0.5 [mm] with P_T = 4.2 and P_L = 1.5. Note that about 90% of W_P is the pumping power on the air side, which major improvement has been made if compared with the commercial one. On the other hand, the pumping power on the water side is somewhat increased in the present design.

Optimization Case 2: Maximization of heat exchange rate

In this case, $-Q_T$ is chosen as the cost function to maximize the heat exchange rate. The five parameters, i.e., d_o, S_T, S_L, $\dot{m}_a$, and $\dot{m}_w$ are simultaneously optimized, while the other parameters including W_P are kept at the same values as the commercial heat exchanger data.

The comparison of total heat exchange rate at the same pumping power is illustrated in Fig. 5. It is obvious that the present design gives higher heat exchange rates, by 33% for H1 and 17% for H2, respectively. The optimum tube diameter is found to be around 0.4 [mm] with P_T = 4.0 and P_L = 1.5.

Optimization Case 3: Minimization of core volume

In Case 3, we keep the total heat exchange Q_T, the pumping power W_P and also the core width w at the same values of the commercial heat exchanger data, while optimize seven parameters d_o, S_T, S_L, $\dot{m}_a$, $\dot{m}_w$, l, and t that would give the minimum core volume ($V = lwt$).

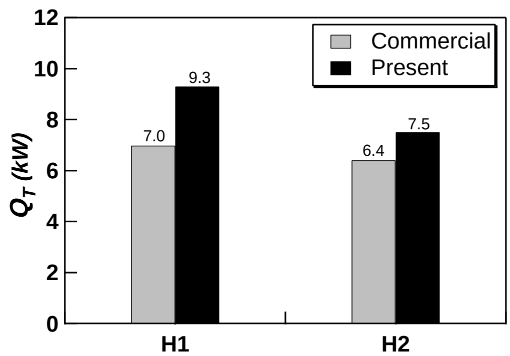

Figure 5. Performance comparison in Case 2.

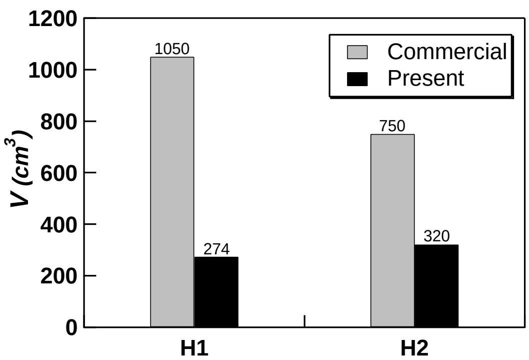

Figure 6. Performance comparison in Case 3.

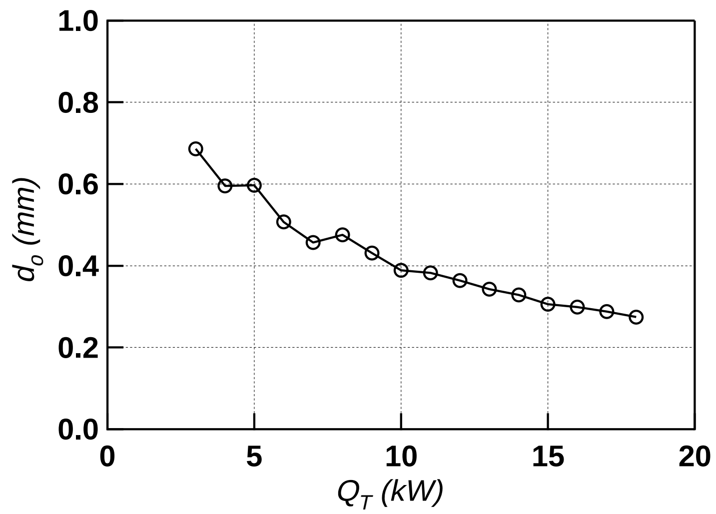

Figure 7. Optimum tube diameter vs total heat exchange for Case 1 (H1).

The core volume of the present design is significantly reduced as shown in Fig. 6. The volume optimized is as small as 26% and 43% of those of H1 and H2, respectively. In this case, $d_o \approx 0.3$ [mm], while $P_T \approx 3.0$ and $P_L \approx 2.0$ for the optimal arrangement.

Characteristics of optimum designs

Figure 7 shows the optimum tube diameter in Case 1 (H1) for various heat exchange rate Q_T from 3 to 18 kW. The optimum value is about 0.7 [mm] at $Q_T = 3$ [kW] and decreases monotonically with increasing Q_T.

Figure 8 shows Q_T / W_P in Case 1 (H1) as d_o and P_T are changed. For each diameter given, there exists a maximum value at some specific P_T. Although $d_o = 0.1$ [mm] does not show a peak, we estimate it will have one at $P_T > 10$. Note that, at the low value of non-dimensional transverse pitch, the bigger tube is more advantageous than the smaller one. However, the overall maximum value occurs at $P_T \approx 4.0$ when $d_0 = 0.5$ [mm] as previously mentioned in the optimization case 1 above.

Validation of the numerical results

In order to verify the optimum design in the present work, we constructed a prototype heat exchanger by using the optimum design parameters in Case 2 (H1). A capilliary copper tube with outer diameter of 0.5 [mm] and the wall thickness of 0.05 [mm] is employed instead of the actual optimum value ($d_o = 0.4$ [mm]). The non-dimensional transverse and longitudinal pitches in this prototype are 3.7 and 2.0, respectively. The total number of copper tubes used is about 3000.

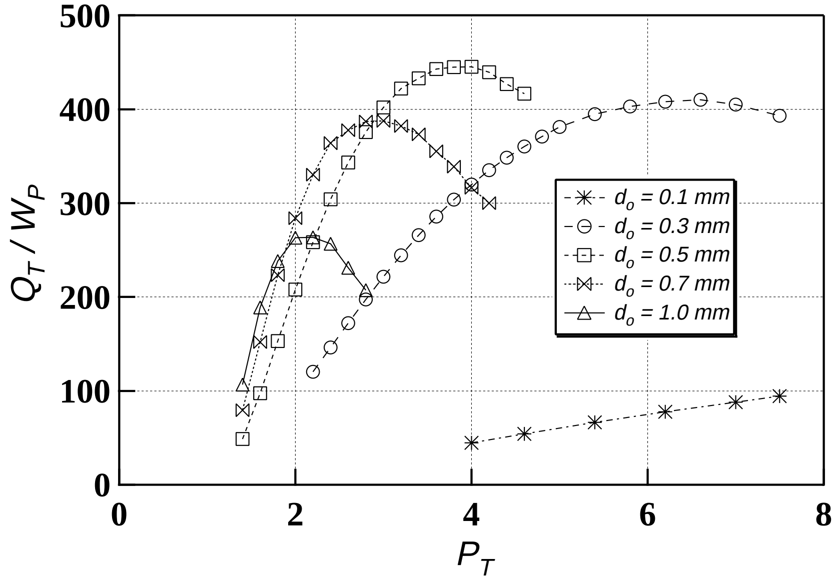

Figure 8. Effect of non-dimensional transverse pitch in Case 1 (H1).

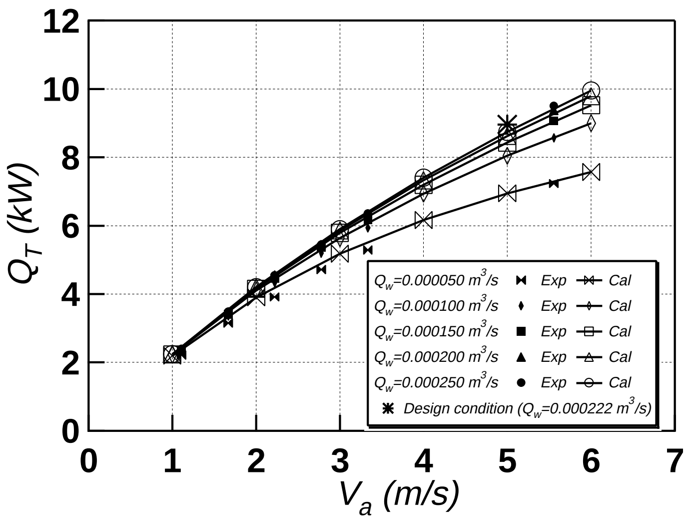

Figure 9. Comparison of total heat transfer between experiment and calculation.

The total heat transfer of this prototype heat exchanger for various frontal air velocities is shown in Fig. 9. The agreement between experiments and calculations is strikingly good. At the design condition, the actual heat transfer is only 3% less than the predicted value. Note that, the heat transfer coefficient on the water side is around 5000 [W/(m^2K)], which is much larger than that for the air side (around 300 [W/(m^2K)]).

4. CONCLUSIONS

An optimum design tool for micro bare-tube heat exchanger has been developed by employing empirical formulae and simulated annealing optimization technique. For three optimization cases examined, the present design has significant advantages over typical commercial heat exchanger as summarized below.

Case 1: With the same heat exchange rate and core volume, the pumping power is decreased by 37-56%.

Case 2: With the same pumping power and core volume, the heat exchange rate is increased by 17-33%.

Case 3: With the same heat exchange rate and pumping power, the core volume is decreased by 57-74%.

Note that, in all cases, the present designs require smaller fan power but larger water pumping power than those of the commercial ones in order to achieve the same desired operation condition. The optimum tube outer diameter ranges from 0.3 mm to 0.5 mm and is found to be decreased gradually with the increase of heat loaded. The performance of the present design is confirmed through physical experiments of a prototype.

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