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ABSTRACT

The major limitation of any air-to-refrigerant Heat eXchanger (HX) is the air side thermal resistance which can account for 90%, or more, of the overall thermal resistance. For this reason, the secondary heat transfer surfaces (fins) play a major role in these HX's by providing additional surface area. Many researchers extensively investigate how to improve the performance of fins. The most common passive heat transfer augmentation method applied to fins uses surface discontinuity; providing an efficient disruption-reattachment mechanism of the boundary layer. Such approach is leveraged by louvers, slits and even vortex generators. In some applications, however, these concepts are not adequate especially when there is high fouling or frosting, which is the case of many HVAC&R systems including heat pumps for cold climates. In such cases a continuous fin surface is required, which can usually be plain or wavy. The latter provides larger surface area and can induce turbulent flows improving the heat transfer. Normally the wavy fins either have a smooth sinusoidal or Herringbone profile, longitudinal to the airflow direction. In this paper, we propose a novel wavy fin design method using Non-Uniform Rational B-Splines (NURBS) on the longitudinal direction as well. The tools used in this work include automated CFD simulations, metamodeling and Multi-Objective Genetic Algorithm (MOGA). The analysis comprises optimizing a conventional Herringbone wavy fin and uses it as a baseline. While maintaining tube diameter, tube pitches, and number of rows, fin spacing and thickness we perform an optimization on the fin profile using NURBS and compare the potential thermal-hydraulic performance improvements.

1. INTRODUCTION

Applications such as heat pumps in cold climates face an extra challenge of frosting in the outdoor unit, which ultimately compromises the overall system performance. For these systems, the commonly known high performance fins like slits and louvers are not suitable. It has been shown empirically that louver fins have a poorer performance compared to plain wavy and flat fins, respectively under frosting/defrosting operating modes (Silva et al., 2011; Huang et al., 2014). Wavy fins are an elegant way of balancing the thermal-hydraulic performance with, and without, frost accumulation, compared to louver and flat fins, which are compensating fin types for each of the operating conditions. Wavy fins are well understood and discussed extensively in the literature (Kays, 1960; Kays & London, 1984; Wang et al., 1999). There is so much that can be done with wavy fins. One way to improve the performance and increase compactness is by optimizing the wave shape. Such type of study is more common on wavy fins and flat tube HX's. Dong et al. (2010) investigated the performances of wavy fin and flat tube with smooth and Herringbone profiles. Recently Song et al. (2015) presented a wavy-fin channel optimization using Constructal Theory (Bejan & Lorente, 2008). Internally enhanced tubes have been subject of studies regarding fin profile optimization (Fabbri, 1997; Fabbri, 1998).

To the author's knowledge there are no studies on the external fin shape optimization for fin and round tube HX's, especially for tube diameters below 5.0mm. In this paper, we present a wavy fin design method using NURBS to find and optimum periodic profile. The purpose is to compare with a conventional Herringbone wavy fin. This study consists of optimizing the shape of the fin while maintaining all other dimensions and air velocity fixed.

2. METHODOLOGY

2.1 Design and optimization framework

The numerical optimization framework (Figure 1) consists of an Approximation Assisted Optimization (Abdelaziz, Azarm, Aute, & Radermacher, 2010), which involves four main steps: a) Problem specification and Design of Experiments (DoE) development; b) CFD modeling and Simulations; c) Metamodel development; d) Multi-Objective Optimization.

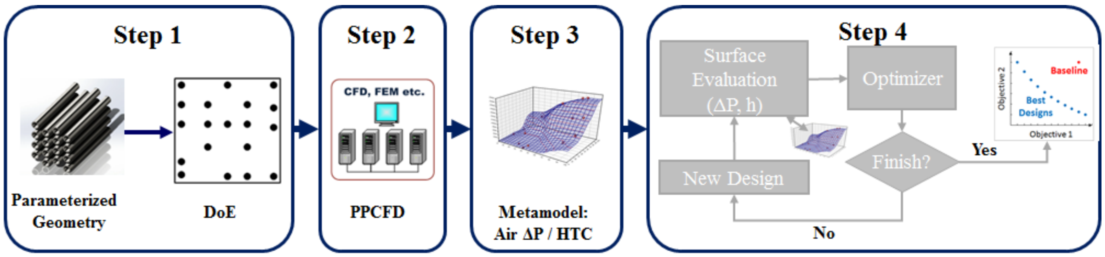
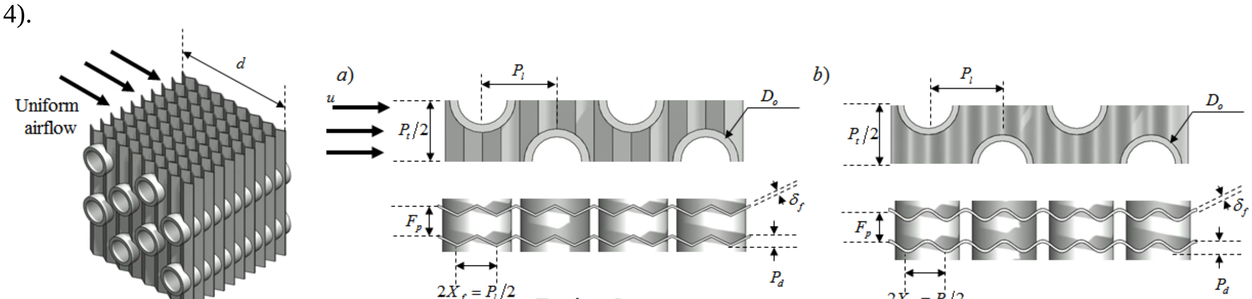


Figure 1: Optimization framework.

2.2 Problem Specification

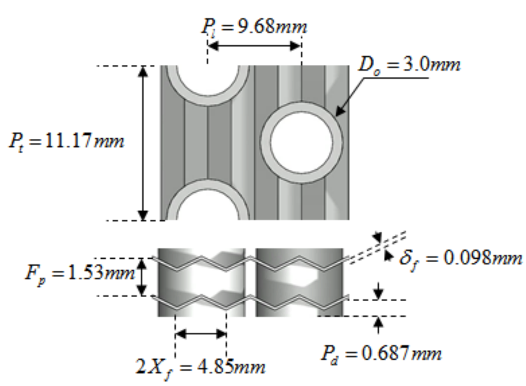
In this paper, we are optimizing a wavy fin and round tube surface using small tube diameters. Conventional wavy fins can be either smooth or Herringbone (Figure 2). As a baseline we picked the latter and optimized the surface that targeting maximum NTU and minimum C_f using the design variables showed in Figure 2. We have selected a design that has a tube diameter of 3. For the same scaling and topology variables (tube diameter, tube pitches, tube banks, fin spacing, fin thickness and air velocity) we optimize the wave profile for the same amplitude and frequency as the Herringbone using NURBS curve with 7 control points and 6th degree interpolation (Figure 4). The design variables are reduced to 6, which are the coordinate pairs of 3 control points. Four additional control points are pre-determined in order to ensure continuity and differentiability at the boundary vertices guaranteeing a smooth periodicity (Figure 4).



Design Space:

D_o (mm)	P_l/D_o (-)	P_t/D_o (-)	F_p (mm)	P_d/X_f (-)	δ_f (mm)	N (-)	u (m/s)
2.0 - 5.0	1.25 – 4.0	1.25 – 4.0	0.5 – 2.5	0.088 – 0.84	0.05 – 0.1	2 - 20	0.5 – 7.0

Figure 2: Conventional wavy fin and tube design space: a) Herringbone; b) Smooth.



Metric	unit	Value
D_h	mm	2.02
u	m/s	1
NTU	-	1.46
C_f	-	0.129
h	W/m².K	68.9
ΔP	Pa	6.14
η_o	-	0.928

Figure 3: Baseline Herringbone fin.

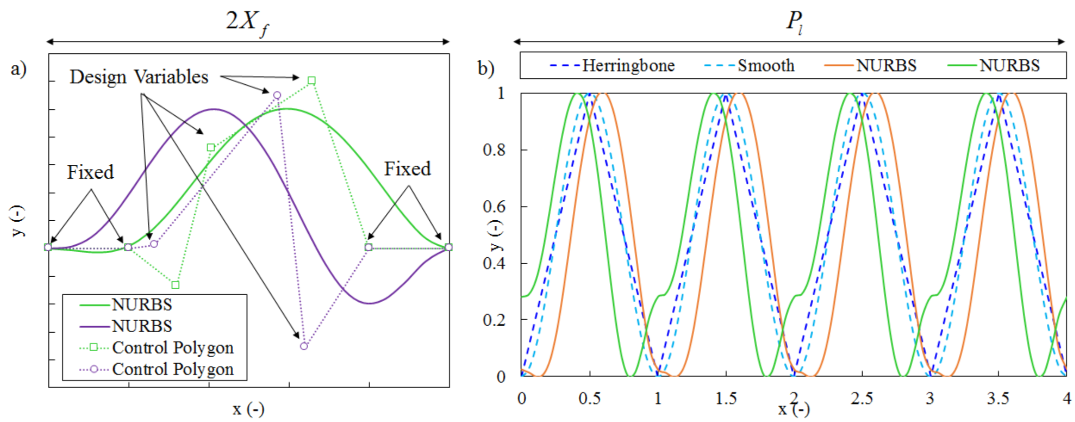


Figure 4: Fin profiles: a) Sample NURBS waves; b) Comparison between sample NURBS and conventional waves.

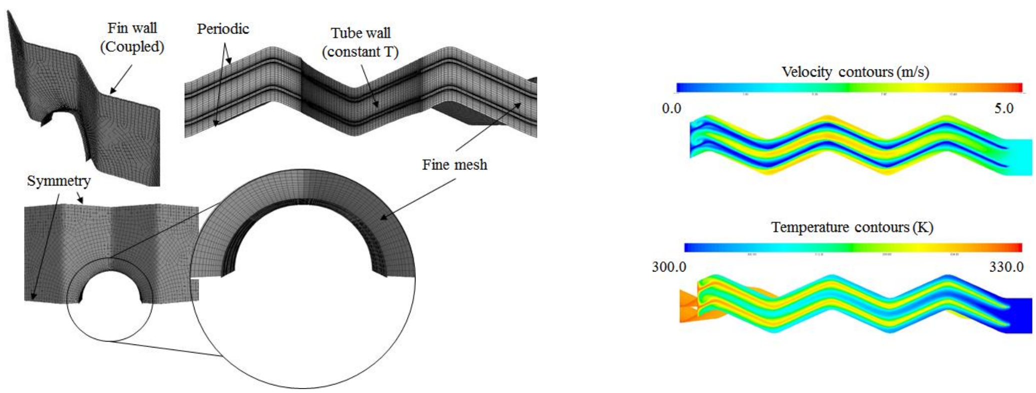


Figure 5: Computational domain, mesh and contour plots.

2.3 CFD Modeling

The CFD computational domain is a two dimensional cross section segment of the HX, assuming any end effects to be negligible. The inlet boundary has uniform velocity and uniform temperature (300K), whereas the outlet boundary is at constant atmospheric pressure. The upper and lower boundaries are periodic, and the tube walls are at constant temperature of 330K, whilst the fin walls are coupled to the tubes. The faces parallel to the fins on the sides are periodic. The fluid properties use ideal gas model, and the turbulence is evaluated using the k-ε realizable model. The convergence criteria used is 10^{-5}. The near wall region mesh is a fine map scheme with growing layers at a ratio of 1.2 (Figure 5). The core of the computational domain is a pave mesh scheme with an average element size equal to the last row of the boundary layer mesh.

2.4 Data reduction

Since the CFD models serve to determine the airside thermal and hydraulic resistances, there is no need to account for additional thermal resistances. Thus with constant wall temperature, the capacitance ratio yields $C_{min} / C_{max} = 0$, then the heat transfer coefficient can be easily calculated through ε-NTU method as per equations (1-3). The pressure drop is determined as the difference between inlet and outlet static pressures, assuming that local losses are negligible.

$$NTU = -\ln(1-\varepsilon) = -\ln\left[1-\left(T_{out}-T_{in}\right)/\left(T_{wall}-T_{in}\right)\right] \tag{1}$$

$$\eta_o h = UA/A_o = NTU \cdot C_{\min}/A_o \tag{2}$$

$$\eta_o = 1-\frac{A_{fin}}{A_o}(1-\eta), \eta = f(h) \tag{3}$$

The fin effectiveness is obtained using the Schmidt (Schmidt, 1949) approximation method.

2.5 Parallel Parameterized CFD
In order to handle a large number of designs the Parallel Parameterized CFD (PPCFD) (Abdelaziz et al., 2010) is a suitable method that allows one to automate CFD simulations. The code consists of reading and writing data and it communicates with the CFD modeling and simulation environments in an automated fashion. The analyses in this study used the ANSYS® platform, more specifically Gambit 2.4.6 for geometry and meshing, and Fluent 14.5 for the simulation runs.

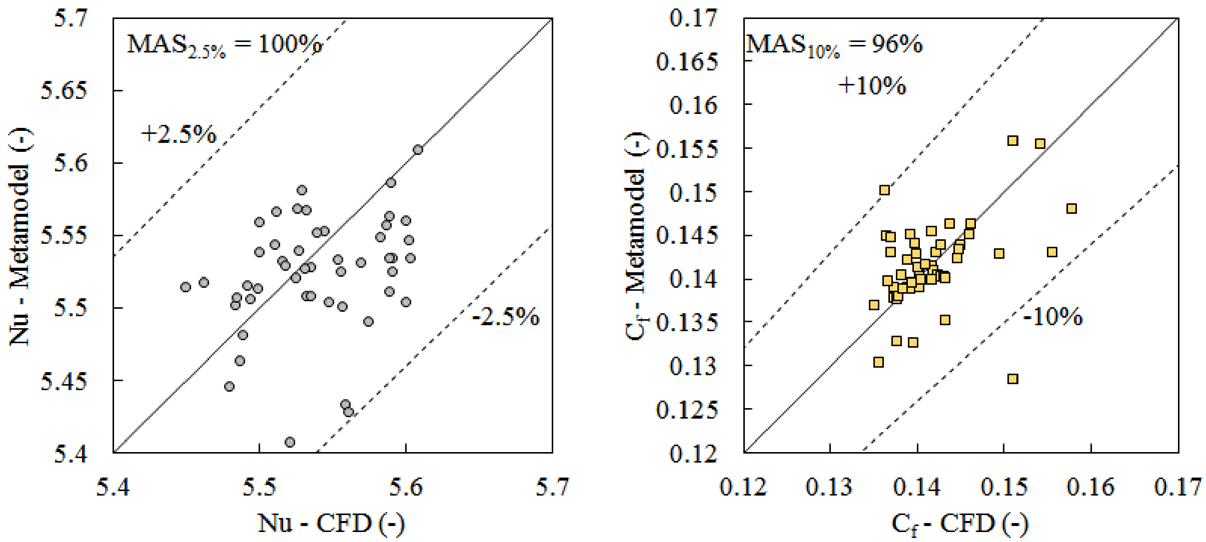


Figure 6: Metamodel verification against random designs.

2.6 CFD Metamodels
We have generated a Design of Experiments using Latin Hypercube Sampling method, containing 170 samples. These were simulated using the PPCFD method previously explained. The post-processed data was used to develop a metamodel correlating heat transfer and pressure drop to the control points defining the fin profile. The metamodel accuracy verification comprises evaluating its ability of predicting responses from random CFD simulations (in addition to the DoE). In this study, the metamodel "goodness" is evaluated using the Metamodel Acceptability Score (MAS) (Hamad, 2006). The MAS value indicates the fraction of predicted responses from a set of random simulations, of which the Absolute Relative Error is equal or less than an established threshold ($e_{MAS} = 10\%$). In this work, the metamodel is acceptable when: $\{MAS \geq 1 - e_{MAS}\}$. The relative error (e_i) (eq. 4) compares the predicted response ($\hat{y}(i)$) with the actual CFD response ($y(i)$). The metamodel results are shown in Figure 6.

$$e_i = \left|\hat{y}(i) - y(i)\right| / y(i) \tag{4}$$

3. PERFORMANCE COMPARISON ANALYSIS

3.1 Multi-Objective Optimization
In this paper, we are optimizing the surface performance. Although the surface area between the baseline and the new designs should be very similar, it is important to quantify the heat transfer performance in terms of thermal resistance, thus characterizing the improvement in heat transfer coefficient but also the area enhancement factor. For this reason, the optimization problem (eq. 5) consists of maximizing the airside NTU (eq. 6) by the friction factor (C_f) (eq. 7) and the HX surface area (eq. 8). Typically, the HX optimization targets more surface area to reduce thermal resistance. In this case the reduction of the surface area has two benefits: the first is if the performance is improving with minimum surface area, then we are seeking designs with higher heat transfer coefficient which is a more noble way of reducing thermal resistance. Secondly, if such surfaces are used under potential frosting conditions then the less surface for frost to grow the better. This problem is solved using Multi-Objective Genetic Algorithm (MOGA) with a population size of 150, 10% replacement and 500 iterations.

$$\begin{aligned}&\max\ NTU/C_f\\&\min\ A_o\\&s.t.\\&NTU/C_f \geq 1.46/0.129\end{aligned} \tag{5}$$

$$NTU = \eta_o h A_o / C_{\min} \tag{6}$$

$$C_f = \Delta P / \left(0.5 \rho u_c^2\right) \cdot D_h / \left(4d\right) \tag{7}$$

$$D_h = 4 A_c d / A_o \tag{8}$$

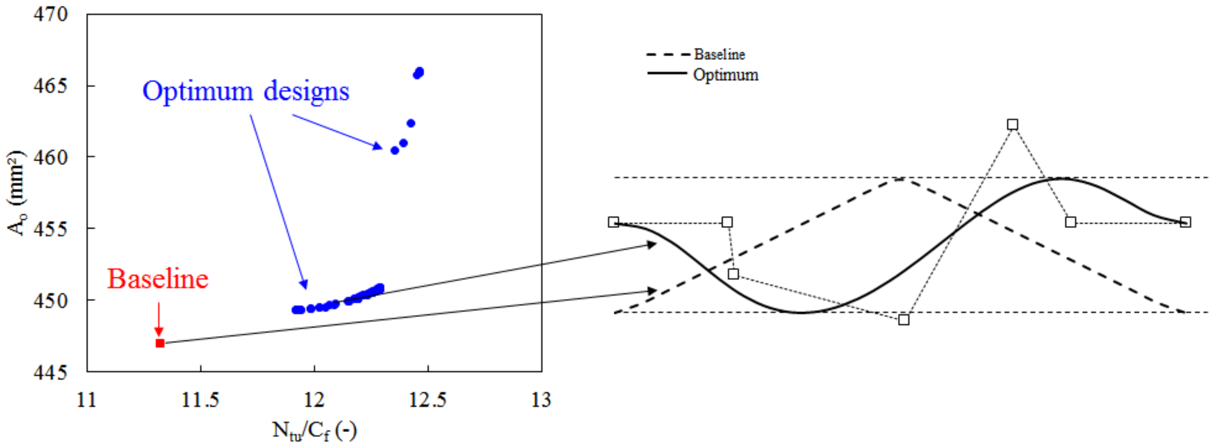


Figure 7: Optimization Results and wave profiles.

The optimization objectives are not conflicting thus the unusual Pareto set. As the area increases there is a positive impact on NTU. Figure 7 shows a comparison of the fin profile between the baseline and one of the optimum designs. All optimum designs have similar profile with slight differences on the x-coordinates.

3.2 CFD Analysis

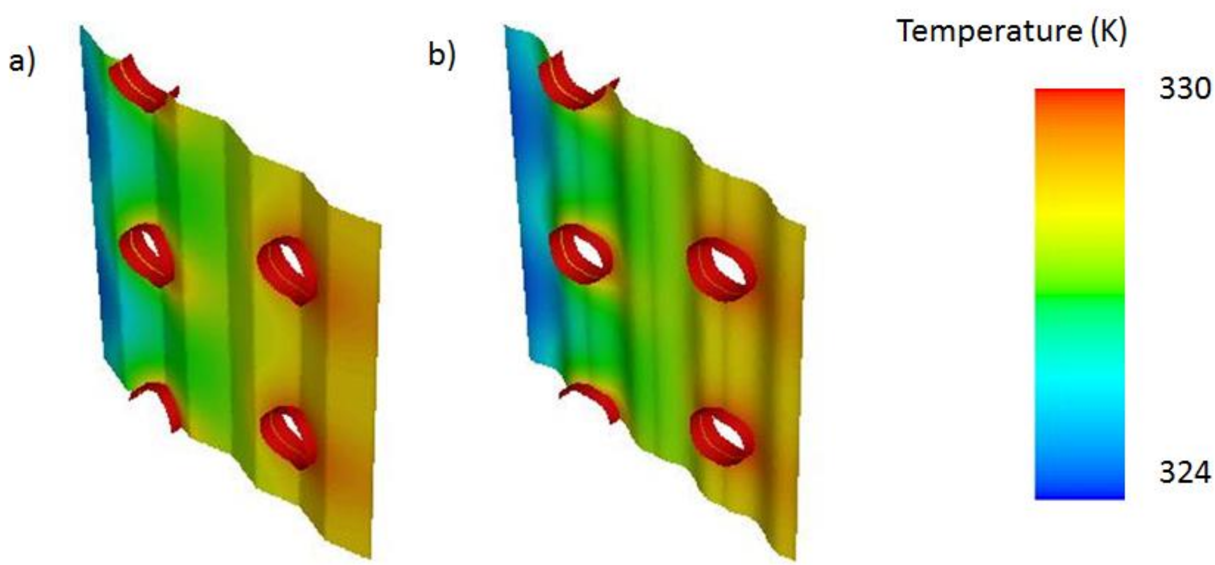


Figure 8. Fin temperature profile: a) baseline; b) optimum design.

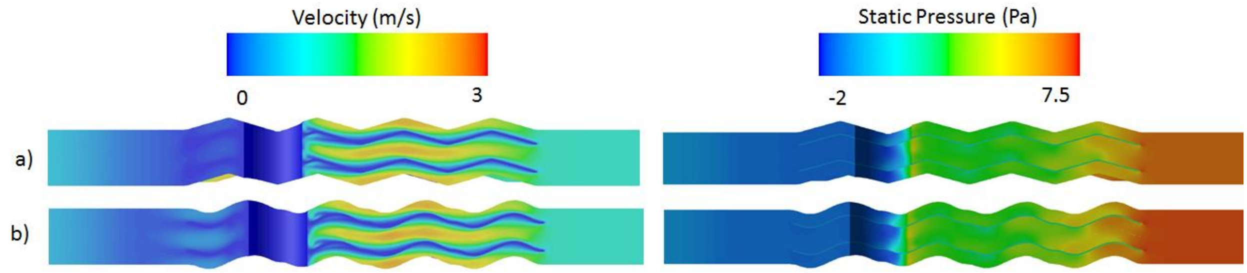


Figure 9. Velocity and pressure contours: a) baseline; b) optimum design.

From Figure 8 we can see that for the optimum design the fin temperatures are lower than the baseline, indicating that more heat is being transferred to the air stream. The pressure contours, however, show that the optimum design has higher pressure drop. The results here are absolutely inconclusive as to whether such optimum fin would outperform the conventional one under frosting or fouling conditions since the performances are very similar and they are compensating in terms of thermal-hydraulic characteristics.

4. CONCLUSIONS

This paper presented one of the first studies on fin profile optimization for fin and round tube HX's using diameters below 5.0mm. The optimization results suggest a potential improvement in thermal resistance both from heat transfer coefficient and surface area augmentation. The higher friction can have enhanced negative effects under severe operating conditions, i.e. frosting or fouling. The overall results suggest the design space studied should be extended in order to capture more designs that may potentially have lower pressure drop in addition to a higher heat transfer. As future work the simulations should include frosting and fouling effects as well. Additionally a more comprehensive analysis would be optimizing the whole surface including tube and fin dimensions. This paper serves as a starting point for a research opportunity with many possibilities to explore this type of surface.

NOMENCLATURE

A_c	Minimum free flow area	m^2	P_d	Wave amplitude	mm
A_{fin}	Fin surface area	m^2	P_l	Tube longitudinal pitch	-
A_{fr}	Frontal face area	m^2	P_t	Tube transverse pitch	-
A_o	Surface area	m^2	T	Temperature	K
C	Heat capacitance rate	W/K	u	Velocity	m/s
C_f	Friction factor	-	UA	Thermal conductance	W/K
c_p	Specific heat	J/kg.K	u_c	Maximum velocity	-
d	Depth	mm	X_f	Half wavelength	mm
D_h	Surface hydraulic diameter	mm	ΔP	Pressure drop	Pa
Do	Tube outer diameter	-			
e	Absolute relative difference	-			
F_p	Fin pitch	-	*Greek Letters*		
h	Heat transfer coefficient	$W/m^2.K$	δ_f	Fin thickness	mm
k	Air conductivity	W/m.K	ε	Effectiveness	-
N	Number of tube banks	-	η	Fin efficiency	-
NTU	Number of tranfer units	-	η_o	Fin effectiveness	-

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