Modelling Air-to-Air Plate-fin Heat Exchanger without Condensation

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SUMMARY

The existing heat exchanger models have limitations in simulating the performance of air-to-air plate-fin heat exchangers (PFHEs). For example, some models adopt empirical correlations which are not suitable for air-to-air PFHEs, while some use a constant effectiveness without considering impacts of changing airflow rate and temperature. In other cases, the models require detailed geometric data as inputs, which are usually difficult to access. To overcome these limitations, based on empirical correlations dedicated to PFHEs, we developed a new air-to-air PFHE model without condensation. This model considers the impact of change of airflow rate and temperature. It uses only nominal data that are known in the design phase and does not need geometric data. To evaluate the performance of the new model, it is implemented using Modelica. Case studies show that the new model could predict the results from experiments with a relative error less than 10% compared to the experimental data.

INTRODUCTION

The plate-fin heat exchanger (PFHE) is a type of compact heat exchanger that consists of a stack of alternate plates called parting sheets and fins brazed together as a block (Picon-Nunez, Polley et al. 1999, Sheik Ismail, Velraj et al. 2010). A study (Zhang Xiaosong 1998) shows that using air-to-air PFHEs in the heating, ventilation and air-conditioning (HVAC) system for heat recovery can lead to great energy saving effect, as the load of the fresh air handling units are reduced by 45% ~ 70%. Modelling is an important approach to study the performance of air-to-air PFHEs.

A review of existing air-to-air heat exchanger from the literature and mainstream simulation platforms shows that they have limitations in the modelling of air-to-air PFHEs.

Wetter (1999) presented a simple simulation model of an air-to-air plate heat exchanger. However, this model is designed for plate heat exchangers and calculates the convective heat transfer coefficient based on an empirical correlation with fixed exponent of velocity, which makes it not applicable for the PFHEs. Nakonieczny (2006) described a numerical model of the plate-fin air-to-air heat exchanger under unsteady flow conditions. In this model, geometric parameters of the heat exchanger are needed, which are usually difficult to access. The unsteady-flow equations in this model are discretized with semi-discrete finite-element method, which can lead to a longer computational time and may cause difficulties in achieving convergence. Rose, Nielsen et al. (2008) and Nielsen, Rose et al. (2009) presented a quasi-steady-state model and a dynamic model of a counter-flow air-to-air heat exchanger, respectively. In these two models, the effects of dehumidification and frost formation are taken into account and geometric data are needed in the calculation of the Reynolds number. Similarly, Liu, Rafati Nasr et al. (2016) developed a theoretical model to predict frosting limits for cross-flow air-to-air heat exchangers, which needs geometric data for the calculation of the heat transfer coefficient.

A review of other air-to-air heat exchanger models implemented on mainstream simulation platforms was conducted. In Modelica Buildings Library (Wetter, Zuo et al. 2014), the model HeatExchangers.ConstantEffectiveness can simulate air-to-air heat transfer, but it uses constant heat effectiveness ε without considering the impacts of the changing conditions on heat transfer. In Energy Plus (DoE 2016), there are three air-to-air heat exchanger simulation models: (a) Air-To-Air Sensible and Latent Effectiveness Heat Exchanger, (b) Air-To-Air Flat Plate Heat Exchanger, (c) Balanced Flow Desiccant Heat Exchanger. Model (a) models full heat exchanger, which is different from PFHE in structure and material. Model (b) adopts Michael Wetter' model (Wetter 1999) mentioned above. Model (c) models desiccant heat exchanger, which is also different from PFHE. In the Standard Component Library of TRNSYS 17 (Klein, Beckman et al. 2014), Type 5 and Type 91 can be used to model the air-to-air heat exchanger. The heat transfer effectiveness ε of Type 5 is calculated based on a fixed overall heat transfer coefficient UA, which also does not consider the impacts of the changing conditions on heat transfer. Type 91 uses a constant effectiveness. In the Standard Component Library of TRNSYS 18 (Klein, Beckman et al. 2017), no new air-to-air heat exchanger model is developed. In TESS Library 17 (Thornton, Bradley et al. 2014), Type 512, Type 650, Type 652, Type 657, Type 699, Type 761, Type 667 and Type 760 can be used to model the air-to-air heat exchanger. All of them use constant heat transfer effectiveness.

In this paper, we present a new model for air-to-air PFHEs, which overcomes the above-mentioned limitations of existing models. It does not need geometric data of the heat exchanger, but only nominal data in the calculation of the heat transfer coefficient. Only explicit equations are used in the model to avoid numerical discretization. In this way, short computational time and numerical stability are ensured. The impacts of the changing airflow rate and temperature are considered. In present stage of our work, we only focus on the modeling of air-to-air PFHEs without condensation. The effects of condensation will be considered in the future work.

The remainder of the paper is organized as follows. Firstly, a detailed deduction of mathematical equations for heat transfer

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is presented. Secondly, we introduce the implementation of the new air-to-air PFHE model with Modelica. Thirdly, case study to validate the new model is presented. Finally, concluding remarks are discussed.

MATHEMATICAL DESCRIPTION

Method of calculating $(\eta_f hA)_i$

By reviewing the literature (Manglik and Bergles 1995, Dong 2007, Sheik Ismail, Velraj et al. 2010), PFHEs have different fin types. The heat transfer correlations in most types of fins can be written as the Equation (1) or (2):

$$j = c_1 c_2 Re^m \tag{1}$$

$$Nu = C Re^n \tag{2}$$

where j is heat transfer factor; c_1 is a constant value which does not depend on the geometry of the heat exchanger; c_2 is a constant value which depends on the geometry of the heat exchanger; Re is Reynolds number; Nu is Nusselt number; C is a constant associated with c_1 and c_2 ; $n = m + 1$ and generally, the range of m lies between -1 and 0 and the range of n between 0 and 1.

Wetter (1999) provided a finned water-air-coil model without condensation, and in its air side, there is:

$$\frac{h_a}{h_{a,0}} = x_a(\vartheta_a)\left(\frac{\dot{m}_a}{\dot{m}_{a,0}}\right)^n \tag{3}$$

where h is convection heat transfer coefficient; $\dot{m}$ is mass flow rate; 0 is the parameters under the nominal condition; a is air side; x_a is the variation of the air properties as a function of the temperature; ϑ_a is air temperature.

In the air-to-air plate-fin heat exchanger, both of two sides are air. Thus, equation (3) can be written as:

$$\frac{h_i}{h_{i,0}} = x_i(\vartheta_i)\left(\frac{\dot{m}_i}{\dot{m}_{i,0}}\right)^n \tag{4}$$

where $i = 1,2$ means side 1 or side 2 of the heat exchanger.

Wetter (1999) mentioned that $x_i(\vartheta_i)$ can be expressed as:

$$x_i(\vartheta_i) = \frac{k_i}{k_{i,0}}\left(\frac{\mu_{i,0}}{\mu_i}\right)^n \tag{5}$$

where k is the thermal conductivity of dry air; μ is the dynamic viscosity of dry air. For the thermal conductivity, k, at a pressure of 1 bar can be approximated linearly by:

$$k = 2.453 \times 10^{-2} + 7.320 \times 10^{-5}\vartheta \tag{6}$$

$$[k] = W/(m \cdot K); [\vartheta] = ℃$$

For the dynamic viscosity, μ_a, the linear approximation for dry air at 1 bar is:

$$\mu = 1.706 \times 10^{-5} + 4.529 \times 10^{-8}\vartheta \tag{7}$$

$$[\mu] = Pa \cdot s; [\vartheta] = ℃$$

To simplify the equation (5), $x_i(\vartheta_i)$ is approximated by a first-order Taylor series expansion with respect to the variable ϑ_i, about the temperature $\vartheta_{i,0}$. A nominal value of 25℃ is selected for $\vartheta_{i,0}$. The approximating $x_i(\vartheta_i)$ can be expressed as:

$$x_i(\vartheta_i) = 1 + (2.7769 - 2.4895 \times n) \times 10^{-3} \times (\vartheta_i - \vartheta_{i,0}) \tag{8}$$

where the second-order terms will be neglected. *Figure 1* shows that the values of approximating $x_i(\vartheta_i)$ are closed to the real ones. Thus, we can use the approximating values to simplify the equation (5) in HVAC system. To avoid an iteration over the heat exchanger, the factor $x_i(\vartheta_i)$ that represents the air property variation is evaluated using the air inlet temperature instead of the mean air temperature. Thus, the equation ((8) can be rewritten as:

$$x_i(\vartheta_{i,in}) = 1 + (2.7769 - 2.4985 \times n) \times 10^{-3} \times (\vartheta_{i,in} - \vartheta_{i,in,0}) \tag{9}$$

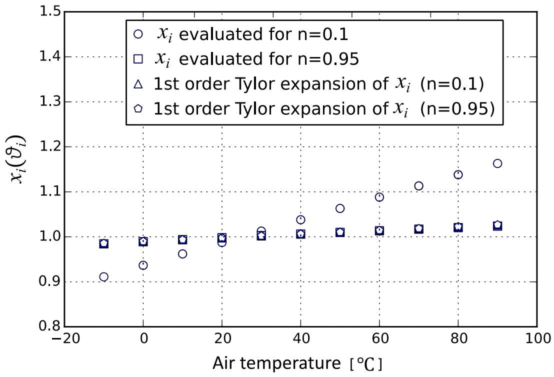

Figure 1 Relative variation of $x_i(\vartheta_i)$, with nominal temperature $\vartheta_{i,0} = 25℃$.

To consider the total fin efficiency and heat transfer area, we update the equation (4), and it can be expressed as:

$$(\eta_f hA)_i = x_i(\vartheta_{i,in})\left(\frac{\dot{m}_i}{\dot{m}_{i,0}}\right)^n (\eta_f hA)_{i,0} \tag{10}$$

where η_f is the total fin efficiency and is constant in this paper; A is heat transfer area.

Under non-nominal conditions, when neglecting the heat resistance of the material and the fouling on the surface, the overall heat transfer coefficient $(U_{avg}A)$ is calculated as(Tan 2013):

$$(U_{avg}A) = \frac{1}{\left(\frac{1}{\eta_f hA}\right)_1 + \left(\frac{1}{\eta_f hA}\right)_2} \tag{11}$$

When equation (11) is calculated under nominal condition and we define $r = \frac{(\eta_f hA)_{1,0}}{(\eta_f hA)_{2,0}}$, and then

$$(\eta_f hA)_{1,0} = (r+1)(U_{avg}A)_0 \tag{12}$$

and

$$(\eta_f hA)_{2,0} = \frac{r+1}{r}(U_{avg}A)_0 \tag{13}$$

Assuming $\dot{m}_1^* = \dot{m}_2^*$ and $\dot{\vartheta}_{1,in}^* = \dot{\vartheta}_{2,in}^*$, and thus, $(\eta_f hA)_1^* = (\eta_f hA)_2^*$, we get the equation (14) and (15) by using equation (10).

$$(\eta_f hA)_1^* = x_1^*(\vartheta_{1,in}^*)\left(\frac{\dot{m}_1^*}{\dot{m}_{1,0}}\right)^n (\eta_f hA)_{1,0} \tag{14}$$

$$(\eta_f hA)_2^* = x_2^*(\vartheta_{2,in}^*)\left(\frac{\dot{m}_2^*}{\dot{m}_{2,0}}\right)^n (\eta_f hA)_{2,0} \tag{15}$$

According to equation (14) and (15), we can calculate r:

$$r = \frac{x_2^*(\vartheta_{2,in}^*)}{x_1^*(\vartheta_{1,in}^*)}\left(\frac{\dot{m}_{1,0}}{\dot{m}_{2,0}}\right)^n \quad (16)$$

Because $\vartheta_{1,in}^*$ and $\vartheta_{2,in}^*$ have little impact on the value of $\frac{x_2^*(\vartheta_{2,in}^*)}{x_1^*(\vartheta_{1,in}^*)}$ in HVAC conditions, we set $\vartheta_{1,in}^* = \vartheta_{2,in}^* = 25°\text{C}$. This is in the median value of air temperature in an HVAC system. Thus, r can be expressed as:

$$r = \frac{x_2(25°\text{C})}{x_1(25°\text{C})}\left(\frac{\dot{m}_{1,0}}{\dot{m}_{2,0}}\right)^n \quad (17)$$

We use equation (17) to calculate r, and then use equation (12) and (13) to calculate $(\eta_f hA)_{i,0}$. After getting values of $x_i(\vartheta_{i,in})$ by using equation (9), we can get values of $(\eta_f hA)_i$ by using equation (10).

Heat transfer and outlet temperature

In this part, we need to calculate the values of heat transfer rate and outlet temperatures of two sides based on the effectiveness-NTU method (Shah and Sekulic 2003). The dimensionless heat transfer effectiveness, ε, is defined as:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (18)$$

where $\dot{Q}$ is the actual heat transfer; $\dot{Q}_{max}$ is the maximum possible heat transfer. We neglect the heat exchanger to environment and the model is no phase exchange. The heat balance of the fluid streams can be expressed as:

$$\dot{Q} = \dot{C}_1|\vartheta_{1,out} - \vartheta_{1,in}| = \dot{C}_2|\vartheta_{2,in} - \vartheta_{2,out}| \quad (19)$$

where $\dot{C} = \dot{m}c_p$. The maximum heat exchange can be written as:

$$\dot{Q}_{max} = \dot{C}_{min}|\vartheta_{2,in} - \vartheta_{1,in}| \quad (20)$$

where $\dot{C}_{min} = \min(\dot{C}_1, \dot{C}_2)$. According to equation (19) and (20), the equation (18) can be rewritten as:

$$\varepsilon = \frac{\dot{C}_1(\vartheta_{1,in} - \vartheta_{1,out})}{\dot{C}_{min}(\vartheta_{1,in} - \vartheta_{2,in})} \quad (21)$$

The effectiveness can also be expressed as a function of the number of heat transfer units (NTU), and the flow arrangement over the heat exchanger:

$$\varepsilon = f(NTU, flow\ arrangement) \quad (22)$$

To a plain fin or wavy fin, equation (22) can be expressed as equation (23); To an offset fin or louvered fin, equation (22) can be written as equation (24) (DoE 2016).

$$\varepsilon = 1 - exp\left(\frac{NTU^{0.22}}{C_r}[exp(-C_r NTU^{0.78}) - 1]\right) \quad (23)$$

$$\varepsilon = \left[\frac{1}{1 - \exp(-NTU)} + \frac{C_r}{1 - \exp(-C_r NTU)} - \frac{1}{NTU}\right]^{-1} \quad (24)$$

where $C_r = \frac{\dot{C}_{min}}{\dot{C}_{max}}$.

The dimensionless number of transfer units NTU is:

$$NTU = \frac{U_{avg}A}{\dot{C}_{min}} \quad (25)$$

$U_{avg}A$ can be calculated by equation (11). By using equation (10), $(\eta_f hA)_1$ and $(\eta_f hA)_2$ can be got and then equation (11) can be rewritten as:

$$(U_{avg}A) = \frac{(r+1)(U_{avg}A)_0}{\frac{1}{x_1}\left(\frac{\dot{m}_{1,0}}{\dot{m}_1}\right)^n + \frac{r}{x_2}\left(\frac{\dot{m}_{2,0}}{\dot{m}_2}\right)^n} \quad (26)$$

After the value of ε is gained, we will use equation (21) to calculate $\vartheta_{1,out}$:

$$\vartheta_{1,out} = \vartheta_{1,in} + \varepsilon\frac{\dot{C}_{min}}{\dot{C}_1}(\vartheta_{2,in} - \vartheta_{1,in}) \quad (27)$$

Then $\dot{Q}$ can be calculated by using equation (19). After that, $\vartheta_{2,out}$ can be got, and it can be written as:

$$\vartheta_{2,out} = \vartheta_{2,in} \pm \frac{\dot{Q}}{\dot{C}_2} \quad (28)$$

In equation (28), if $\vartheta_{1,in} > \vartheta_{1,out}$, then $\vartheta_{2,in} < \vartheta_{2,out}$, and "+" is used; if $\vartheta_{1,in} < \vartheta_{1,out}$, then $\vartheta_{2,in} > \vartheta_{2,out}$, and "–" is used.

CREATION OF MODELICA MODEL

Introduction of Modelica and Modelica Buildings Library

We selected Modelica as the modeling language. Modelica is an equation-based, object-oriented modeling language. It is called the next generation modeling language (Mattsson, Elmqvist et al. 1997) and a new paradigm for building energy modeling, simulation and optimization (Wetter, Bonvini et al. 2016). Compared to traditional building simulation programs, Modelica-based modeling and simulation have the following characteristics: efficient numerical solution, well management of complex large systems, simulation of dynamic effects, use of models beyond time domain simulation, use of models in conjunction with optimization algorithms, etc. (Wetter 2009). Based on Modelica, Lawrence Berkeley National Laboratory (LBNL) has developed a free open-source library called Modelica Buildings Library (Wetter, Zuo et al. 2014). This library supports rapid prototyping, as well as design and operation of building energy and control systems including HVAC system topologies (Wetter, Zuo et al. 2014, Wetter, Bonvini et al. 2015). In this library, existing packages include prototype models for fluid, heat transfer, media and room, etc., which offers great convenience for users to implement building energy system modeling and simulation. The proposed air-to-air PFHE model without dehumidification is implemented based on this library. The following sections show our model in Modelica.

Model of air-to-air PFHE

According to mathematical description, we build the model of air-to-air PFHE in Modelica. The model is shown in Figure 2. Over side 1, the fluid flows from Air inlet of side 1 in and from Air outlet of side 1 out and over side 2 from Air inlet of side 2 in and from Air outlet of side 2 out. Mass flow rate sensor1, Mass flow rate sensor 2, Temperature sensor 1 and Temperature sensor 2 measure the mass flow rates $\dot{m}_1$, $\dot{m}_2$ and the inlet temperatures $T_{1,in}, T_{2,in}$ over the two sides respectively and export the results to the hA (air-to-air) module and E_NTU calculator module. The hA (air-to-air) module is used to calculate the heat conductivity $(\eta_t hA)_1$ and $(\eta_t hA)_2$ over the two sides and exports the results to the E_NTU calculator module. The E_NTU calculator module is the core module of the effectiveness-NTU method and used to calculate UA, NTU, ε, $\dot{Q}_{max}$ under non-nominal conditions. The function of module Q_calculator is to solve the Q_1 and Q_2 over two sides of the heat exchanger, which are then imported into Static conservation equation 1 and Static conservation equation 2 modules respectively. These two modules are instances of taticTwoPortConservationEquation module in Modelica Buildings Library. This module implements a steady-state conservation equation for energy and mass fractions and calculates outlet parameters of the heat exchanger.

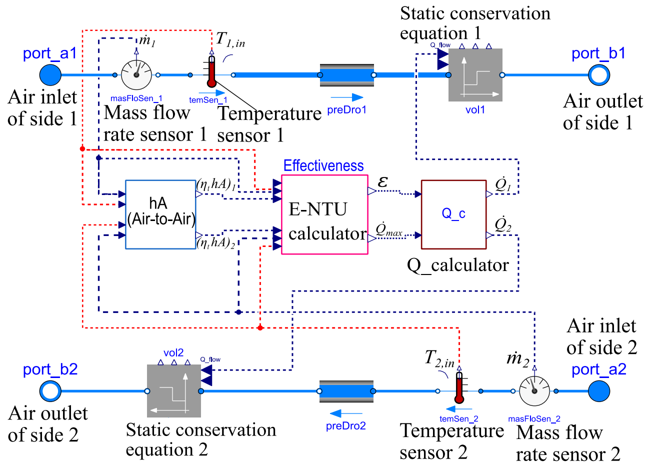

Figure 2 Model of air-to-air PFHE in Modelica.

CASE STUDY

Figure 3. shows the diagram of the case study model in Modelica. In this figure, the air-to-air PFHE is the new heat exchanger model. Sink_1 and Sink_2 represent two ideal heat sinks that are infinitely large. They set the system pressure level and simulate the ambient environment. Fan_1 and Fan_2 are ideal flow source models. The air mass flow rate over the two sides of the heat exchanger are given by the fixed value output blocks Flow rate setting of side 1 and Flow rate setting of side 2. The fluid inlet temperature over the two sides are given by the fixed value output blocks Temperature setting of side 1 and Temperature setting of side 2. $T_{1,out}$ and $T_{2,out}$ are outlet temperatures over the two sides given by the temperature sensors.

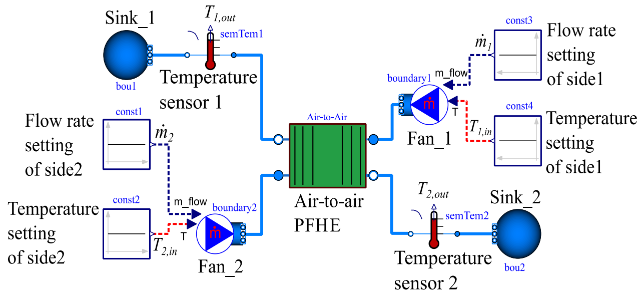

Figure 3 Diagram of the case study model in Modelica

Table 1. Experimental and simulating data of heat exchanger

Case	Fresh air						Exhaust air					
	$\dot{m}_1$	$\vartheta_{1,in}$	$\vartheta_{1,out}$		$\dot{Q}_1$		$\dot{m}_2$	$\vartheta_{2,in}$	$\vartheta_{2,out}$		$\dot{Q}_2$	
			Exp	Sim	Exp	Sim			Exp	Sim	Exp	Sim
	kg/s	℃	℃	℃	kW	kW	kg/s	℃	℃	℃	kW	kW
1	0.33	35.24	31.55	31.57	1.230	1.229	0.33	27.15	30.94	30.82	1.260	1.229
2	0.40	35.25	31.68	31.66	1.440	1.457	0.40	27.05	30.47	30.64	1.380	1.457
3	0.50	35.28	31.74	31.92	1.790	1.704	0.50	27.27	30.68	30.63	1.720	1.704
4	0.60	35.31	31.72	32.02	2.180	2.005	0.60	27.17	30.63	30.46	2.100	2.005
5	0.67	35.45	31.95	32.21	2.370	2.205	0.67	27.25	30.58	30.49	2.250	2.205
6	0.73	36.01	32.54	32.58	2.560	2.540	0.73	27.19	30.61	30.62	2.520	2.540
7	0.83	36.45	32.62	33.00	3.210	2.902	0.83	27.35	31.01	30.8	3.070	2.902

The experimental data listed in *Table 1* from the literature (Gao 2008) are used to do case study of the proposed air-to-air PFHE model. "Exp" means data from experiment.

We chose the exponent *m* of the heat transfer factor *j* from the literature (Dong 2007), which is $m = -0.3345$, $n = m + 1 = 0.6655$. We chose Case 6 as the nominal condition. After running the cases, the simulation results are summarized in *Table 1*. "Sim" means data from simulation. To compare the results from experiments, we need to calculate the relative differentials. The equation can be expressed as:

$$Diff = \frac{Value_{sim} - Value_{exp}}{Value_{exp}} \times 100\% \qquad (29)$$

where $Value_{sim}$ means data from simulation; $Value_{exp}$ means data from experiment. According to equation (29), the relative differentials are summarized in *Table 2*.

Table 2. Relative differentials of outlet temperatures and heat transfer rates

Case	Fresh air		Exhaust air	
	$\vartheta_{1,out}$	$\dot{Q}_1$	$\vartheta_{2,out}$	$\dot{Q}_2$
1	0.06%	-0.08%	-0.39%	-2.46%
2	-0.06%	1.18%	0.56%	5.58%
3	0.57%	-4.80%	-0.16%	-0.93%
4	0.95%	-8.03%	-0.56%	-4.52%
5	0.81%	-6.96%	-0.29%	-2.00%
6	0.12%	-0.78%	0.03%	0.79%
7	1.16%	-9.60%	-0.68%	-5.47%

From *Table 2*, outlet temperatures from simulation are closed to those got from experiments. When we do experiments, the situation of environment, measured errors and other reasons may impact the effects of heat transfer rate. Thus, the values of heat transfer rate in the model are higher than those in experiments. The highest value of differential can be -9.60% (Case 7). According to analysis, the new model is workable, and can be used in simulation of the HVAC system.

CONCLUSIONS

In the paper, a new model of air-to-air plate-fin heat exchanger is built. Then existing data from experiments are found from other researchers' works. We use these data to validate the model. The results show that the new model can simulate air-to-air PFHEs with reasonable error.

This new model considers the impact of the changing airflow rate and temperature and is capable of predicting part-load behaviour, requiring only nominal data, which are known in the design phase. It does not need geometric data as inputs. Furthermore, it does not require numerical discretization which is computationally more efficient than the models using the finite-element method.

This model can be only used in the situation without condensation, and the correlations must own the form, $j = c_1 c_2 Re^m$ or $Nu = CRe^n$ ($n = m + 1$ and n is between 0 and 1).

In the future, we will continue building a new model of air-to-air plate-fin heat exchanger that can be used in the situation with condensation.

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