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CoilDesigner: a general-purpose simulation and design tool for air-to-refrigerant heat exchangers

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Abstract

A simulation and design tool to improve effectiveness and efficiency in design, and analysis of air to refrigerant heat exchangers, CoilDesigner, is introduced. A network viewpoint was adopted to establish the general-purpose solver and allow for analysis of arbitrary tube circuitry and mal-distribution of fluid flow inside the tube circuits. A segment-by-segment approach within each tube was implemented, to account for two-dimensional non-uniformity of air distribution across the heat exchanger, and heterogeneous refrigerant flow patterns through a tube. Coupled heat exchangers with multiple fluids inside different subsets of tubes can be modeled and analyzed simultaneously. A further sub-dividing-segment model was developed in order to address the significant change of properties and heat transfer coefficients in the single-phase and two-phase regime when a segment experiences flow regime change. Object-oriented programming techniques were applied in developing the program to facilitate a modular, highly flexible and customizable design platform and in building a graphic user-friendly interface. A wide variety of working fluids and correlations of heat transfer and pressure drop are available at the user's choice. The model prediction with CoilDesigner was verified against experimentally determined data collected from a number of sources.

Keywords: Heat exchanger; Evaporator; Air cooler; Software; Calculation; Heat transfer

CoilDesigner: un outil complet permettant la simulation et la conception des échangeurs de chaleur air-frigorigène

Mots clés : Échangeur de chaleur ; Évaporateur ; Refroidisseur d'air ; Logiciel ; Calcul ; Transfert de chaleur

1. Introduction

Paradigms in which a computer model replaces the physical prototype in the design and optimization of energy-consuming equipment and systems, are becoming increasingly popular and gaining great momentum. It is showing more and more effect on reducing product design and development time, saving capital and operational cost, and increasing energy efficiency of the system.

Air-to-refrigerant heat exchangers, mostly plate-fin-tube coils and microchannel heat exchangers, used to transfer heat between air and fluid (refrigerant, water, water–glycol,

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Nomenclature

A	heat transfer area (m^2)
c	friction factor
c_p	specific heat ($J\ kg^{-1}\ K^{-1}$
D	tube diameter (m)
F	function name
g	local gravitational constant ($m\ s^{-2}$)
h	specific enthalpy ($J\ kg^{-1}$)
h	convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
h_d	convective mass transfer coefficient ($kg\ m^{-2}\ s^{-1}$)
h_{fg}	vaporization heat ($J\ kg^{-1}$)
I	index for tube, tube location, refrigerant flow direction
J	index for junction
JTA	junction-tube connectivity matrix
K	thermal conductivity ($W\ m^{-2}\ K^{-1}$)
L	length (m)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
N	number of tubes in each row, number of segments in each tube
NTU	number of transfer units
P	pressure
Q	heat transfer rate (W or Btu h^{-1})
R	heat transfer resistance (kW^{-1})
S_l	tube spacing in air flow direction (m)
S_t	tube spacing normal to air flow direction (m)
T	temperature (°C or °F)
x	horizontal coordinate of tube location (m)
y	vertical coordinate of tube location (m)
Greek symbols	
η	heat transfer effectiveness
η_{ms}	surface effectiveness under wet surface condition
η_s	surface effectiveness under dry surface condition
θ	angle (degrees)
ρ	density ($kg\ m^{-3}$)
ω	humidity ratio of air ($kg\ kg^{-1}$ dry air)
Subscript	
air	moist air
a	acceleration
bend	tube bends
C	fin-tube contact
dir	refrigerant flow direction in terms of entering page, or leaving page
dry	dry surface
F	fouling, friction
g	gravitation
I	index of tube, segment
in	inlet, inside
J	index of segment
k	index of segment
out	outlet, outside
ref	refrigerant
row	tube row
s	surface
seg	segment of tube
T	tube
total	total surface area of fin and tube
ver	tube in each row
w	wall
water	moisture

ammonia–water, or oil), are the major components of the air conditioning, heat pumping, and refrigeration systems. They play a vital role in the manufacturing cost and energy consumption of the systems. Due to the complexity in terms of geometry, tube arrangement, circuitry, non-uniformity of airflow, thermal and hydraulic phenomena in multi-phase flow, and variety of working fluids, it appears to be inefficient to accurately and rapidly predict the performance of these coils by analytical or graphic design approaches as described in many conventional heat exchanger and HVAC handbooks.

There are a number of mathematical models and simulation tools that have been developed for design and rating of heat exchangers. However, the use of these models and tools is restricted because they are tailored to very specific existing systems or component applications, or because their usage requires a considerable amount of time and effort due to lack of graphic user interface, or because their modeling of thermal and hydraulic phenomena is simplified.

Domanski [1,2] presented a public-domain software package EVAP-COND for simulating the performance of finned-tube evaporators and condensers, using a graphic user interface with the overall performance shown in a tabular form. No freedom in choice of correlations for heat transfer and pressure drop is available. The circuitry design is limited to typical evaporator or condenser configurations. This simulation tool was based on the mathematical model EVSIM with a tube-by-tube approach by the same author [3]. The non-uniform distribution of the refrigerant flow among individual circuits after splitting at a particular location in the coil is addressed. The air flow mal-distribution can only be addressed on the tube level (one-dimensional), whereas, in real life the air distribution is almost always non-uniform in two-dimensions.

Bensafi et al. [4], Vardhan et al. [5], Corberan et al. [6], Liang [7], Oliet [8], and Wang [9], developed various computational programs for plate-fin-tube coils. They elaborated on their own features in their models. Some literature detailed heat transfer and fluid flow calculations by discretizing the entire coil into 3D nodes or elements and using local air/fluid properties in solving the associated governing equations. Some made comparative studies of a number of correlations for both heat transfer and pressure drop on the refrigerant side, or of refrigerant flow circuitries to balance the increase of the heat transfer coefficient and pressure drop and therefore to improve the coil performance. Other literature made contributions on establishing sophisticated models on air moisture removal from the fin and tube surface, or comprising of more working fluids.

In this paper, CoilDesigner, a general-purpose simulation and design tool for air-to-refrigerant heat exchangers, is introduced. CoilDesigner distinguishes itself by providing the greatest generality and flexibility to date along with an array of customization features. Underlying the tool is a detailed mathematical model of the nature of thermal and hydraulic phenomena integrated with the most up to date and comprehensive libraries of working fluids and correlations for heat transfer and pressure drop for both refrigerant and air side. The highly user-friendly graphical interface facilitates ease of circuitry design, intuitive and versatile forms for data input and output, parametric analysis, interaction with spreadsheets, choice of fin types and refrigerants, addition of user defined correlations, etc. The modular component based programming approach adopted in CoilDesigner allows integration with other simulation and optimization tools. With these combined distinct features, CoilDesigner is expected to significantly increase the productivity and reduce the cost in design and rating of heat exchangers.

2. Modeling approach

A 'general' finned tube coil is shown in Fig. 1. 'General' here refers to arbitrary refrigerant flow circuiting, with no restrictions on the number and location of the inlet and outlet streams, and allowing for multiple working fluids in different subsets of the coil.

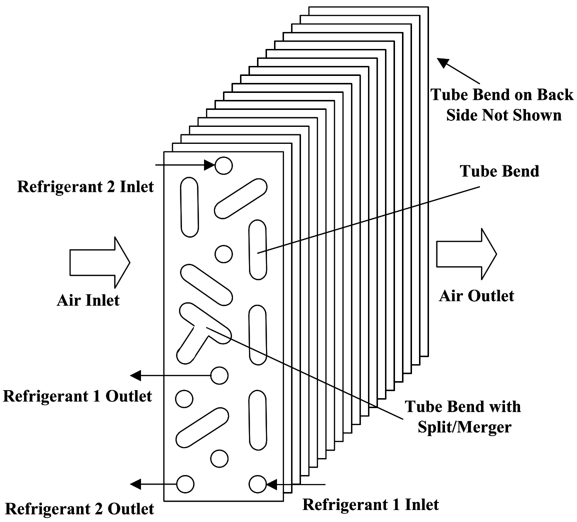

Fig. 1. A general fin-tube coil.

2.1. Junction-tube connectivity matrix

The term 'junction' is defined as the intersection where two or more tubes are joined together. In order to facilitate model implementation, a junction-tube connectivity matrix is defined and created to describe the location relationship between junctions and tubes. The relationship is as follows:

JTA[j,i] = 1: junction j is upstream and connected to tube i
JTA[j,i] = −1: junction j is downstream and connected to tube i
JTA[j,i] = 0: junction j is not connected to tube i

Fig. 2 shows a heat exchanger constructed of eight tubes with six junctions. The left side of this figure is the cross-sectional view of the coil. Each circle represents a tube. The crossed sign indicates that the refrigerant flows into the page, and the dotted sign indicates that the refrigerant flows out of the page. The solid line between the two circles indicates that the two tubes are connected at the frontal side, whereas, the dashed line means the two tubes are connected at the backside. The right side of the figure is an electric-network-like representation of the coil in terms of the tube connections. Each rectangle represents a tube. Each dot is a junction. The junction-tube connectivity matrix is shown in Table 1.

Based on information in the junction-tube connectivity matrix, the following can be determined:

(1) The refrigerant flow can be tracked from the inlet(s) of the coil to the outlet(s) of the coil to establish the energy balance relationship between the upward stream and downward stream of refrigerant.
(2) The direction of the refrigerant flow: an index i_{dir} of 1 or −1 is assigned to the tube where the refrigerant flows into the page or out of the page, respectively.
(3) When there are different working fluids (independent streams) in the same coil, the circuit(s) that contains the same working fluid can be distinguished.

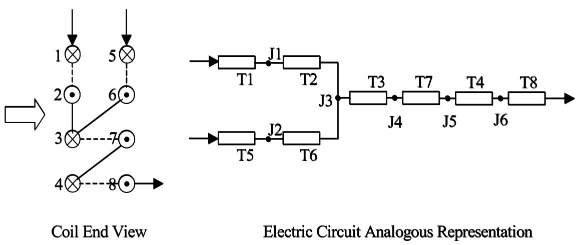

Fig. 2. Illustration of junctions and tubes in a coil.

Table 1
Junction-tube connectivity matrix

Junction	Tube							
	1	2	3	4	5	6	7	8
1	−1	1	0	0	0	0	0	0
2	0	0	0	0	−1	1	0	0
3	0	−1	1	0	0	−1	0	0
4	0	0	−1	0	0	0	1	0
5	0	0	0	1	0	0	−1	0
6	0	0	0	−1	0	0	0	1

The tubes are numbered in order of top to bottom in each row (normal to air flow), and left row to right row (in the direction of air flow), as shown in Fig. 2, to facilitate forming the 2D junction-tube connectivity matrix. The number of junctions depends on the circuiting design. Indexes i_{row} and i_{ver} identify respectively in which row and at which vertical location in that row the tube is situated. Using these two indexes, the predecessor–successor relationship of tubes in the airflow direction can be determined, and the length of the connecting bend between any two tubes can be calculated.

To account for non-uniform air distribution, as well as heterogeneous properties and heat transfer coefficients of the refrigerant, each finned tube is divided into N_{seg} segments. The segments are numbered from 1 to N_{seg} in the order in which the refrigerant flows through the tube.

Based on the tube location indexes i_{row} and i_{ver} and the refrigerant flow direction index i_{dir} of the tube, the predecessor and successor segments of the neighboring tubes can be identified, for the purpose of energy and mass conservation analysis on the air side.

2.2. Modeling assumptions

In developing the heat transfer model for the heat exchanger, the following assumptions are made:

(1) Each segment is treated as a discrete unit of heat transfer, without considering the conduction heat transfer through the fin plates between tubes.

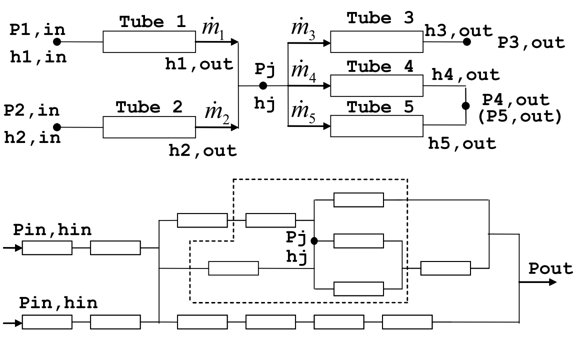

Fig. 3. Refrigerant side mass and energy flow in a tube network.

(2) When the air flow at the face of the heat exchanger is non-uniform (i.e. different air velocities at different segments of the tubes in the frontal face row), the air velocity at the segments in the air flow direction across the heat exchanger remains the same as that at the corresponding segment in the frontal face. The air side heat transfer coefficient for each segment is calculated based on the individual air velocity at that segment.

(3) When dehumidification occurs, the heat transfer resistance due to the water film on the surface of the tube and the fin, is either neglected or can be accounted for by adding a certain value of resistance to the fin-tube contact resistance.

2.3. Refrigerant side modeling

In order to simulate a heat exchanger without restriction on the tube connection and flow circuitry, an analogy of the coil to an electric circuit can be made. The finned tube in a coil is like the resistor in an electric circuit, the mass flow rate through a tube is analogous to the electric current, and the pressure is analogous to the electric potential. While the electric current via a resistor can be considered as a linear function of the potentials at the two ends of the resistors, the mass flow rate through a tube is a highly non-linear function and determined by several variables including inlet pressure, inlet enthalpy and outlet pressure of the refrigerant and the surrounding air condition, plus the dimensions of the tube and the fins. The inlet enthalpy of the refrigerant is determined by the heat transfer of upstream fluids.

Fig. 3 shows how the equations of mass and energy conservation are formulated at a particular junction. This particular tube network has been chosen to be more complex and general than one would find in a heat exchanger, to demonstrate the concept. The mass flow entering a junction j is equal to the mass flow leaving the junction *j*. The energy flow entering the junction is equal to the energy flow leaving the junction [10]. The enthalpy at the inlet of each tube downstream to the junction *j* is the mass flow weighted average enthalpy of the fluid mixed at the junction from the upstream tubes.

$$\sum_{\mathrm{JTA}[j][i]=-1} \dot{m}_i = \sum_{\mathrm{JTA}[j][i]=1} \dot{m}_i \tag{1}$$

where

$$\sum_{\mathrm{JTA}[j][i]=-1} \dot{m}_i h_{i,\mathrm{out}} = \sum_{\mathrm{JTA}[j][i]=1} \dot{m}_i h_{i,\mathrm{in}} \tag{2}$$

$$h_{i,\mathrm{in}} = \frac{\sum_{\mathrm{JTA}[j][i]=-1} \dot{m}_i h_i}{\sum_{\mathrm{JTA}[j][i]=-1} \dot{m}_i} \tag{3}$$

The subscripts j and i denote junction and tube, respectively.

The fluid pressure drop over the segment k of tube i can be expressed in a hydraulic equation,

$$P_{i,k,\mathrm{in}} - P_{i,k,\mathrm{out}} = \Delta P_\mathrm{f} + \Delta P_\mathrm{a} + \Delta P_\mathrm{g} \tag{4}$$

where ΔP_f is the friction term, and can be calculated in the form,

$$\Delta P_\mathrm{f} = c \frac{2 l_\mathrm{seg}}{0.5\pi(\rho_{i,k,\mathrm{in}} + \rho_{i,k,\mathrm{out}}) D_\mathrm{in}^3} \dot{m}_i^2 \tag{5}$$

ΔP_a is the acceleration term,

$$\Delta P_\mathrm{a} = \frac{16 \dot{m}_i^2}{\pi^2 D_\mathrm{in}^4} \left(\frac{1}{\rho_{i,k,\mathrm{out}}} - \frac{1}{\rho_{i,k,\mathrm{in}}} \right) \tag{6}$$

and ΔP_g is the gravitational term,

$$\Delta P_\mathrm{g} = 0.5(\rho_{i,k,\mathrm{in}} + \rho_{i,k,\mathrm{out}}) g l_\mathrm{seg} \sin\theta \tag{7}$$

Various correlations [11–14] and empirical equations exist for obtaining the frictional pressure drop in the form of Eq. (5), both for single phase and two phase flow, depending on the flow pattern, working fluids, tube type, heat and mass flux, and other operating conditions.

The tubes in the heat exchanger are connected to each other via 180° bends. The pressure drop in the tube bend is usually higher than that in a straight tube of the same length. The enhancing effect of the bend curvature is normally accounted for with a multiplier or an additional term. CoilDesigner as a general tool allowing arbitrary circuitry design is capable of calculating the length of an arbitrary tube bend:

$$l_\mathrm{bend} = \frac{\pi}{2} \sqrt{(x_1 - x_2)^2 + (y_1 - y_2)^2} \tag{8}$$

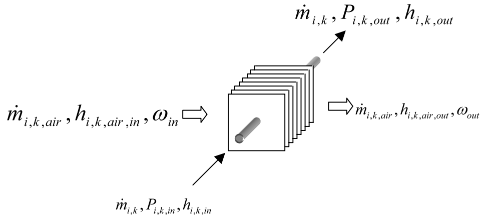

Fig. 4. Segment as a single cross-flow heat exchanger.

where,

$$x_1 = (i_{\mathrm{row},1} - 1) S_\mathrm{l} \tag{9}$$

$$x_2 = (i_{\mathrm{row},2} - 1) S_\mathrm{l} \tag{10}$$

For inline tube arrangement,

$$y_1 = (N_\mathrm{ver} - i_{\mathrm{ver},1}) S_\mathrm{t} \tag{11}$$

$$y_2 = (N_\mathrm{ver} - i_{\mathrm{ver},2}) S_\mathrm{t} \tag{12}$$

For staggered tube configuration, if the tube is at the odd column, y_1 and/or y_2 are calculated using Eqs. (11) and (12); if the tube is at the even column: for staggered-convergent configuration,

$$y_1 = (N_\mathrm{ver} - i_{\mathrm{ver},1} - 0.5) S_\mathrm{t} \tag{13}$$

$$y_2 = (N_\mathrm{ver} - i_{\mathrm{ver},2} - 0.5) S_\mathrm{t} \tag{14}$$

for staggered-divergent configuration,

$$y_1 = (N_\mathrm{ver} - i_{\mathrm{ver},1} + 0.5) S_\mathrm{t} \tag{15}$$

$$y_2 = (N_\mathrm{ver} - i_{\mathrm{ver},2} + 0.5) S_\mathrm{t} \tag{16}$$

Appropriate built-in correlations [15,16] are available for calculating pressure drop in the bends. User-defined correlations can also be used for bend pressure drop calculations.

2.4. Modeling of heat transfer between refrigerant and air

Each segment of a tube is treated as a single cross-flow heat exchanger as illustrated in Fig. 4. The air across the finned segment is assumed to be the unmixed fluid, and the refrigerant throughout the segment is the mixed fluid.

2.5. Dry surface condition

When the average wall/fin temperature of the tube segment is higher than the dew temperature of the air flowing across the segment, it is assumed that no water vapor condensation occurs and the segment operates under dry surface condition.

The equations of heat transfer between air and refrigerant, and energy balance are,

$$Q = \frac{\Delta T_\mathrm{m}}{R} = \frac{f(T_\mathrm{air,in}, T_\mathrm{air,out}, T_\mathrm{ref,in}, T_\mathrm{ref,out})}{R} \tag{17}$$

$$Q = \dot{m}_\mathrm{air}(h_\mathrm{air,in} - h_\mathrm{air,out}) \tag{18}$$

$$Q = \dot{m}(h_\mathrm{ref,out} - h_\mathrm{ref,in}) \tag{19}$$

where,

$$R = \frac{1}{h_\mathrm{ref} A_\mathrm{t,in}} + \frac{(D_\mathrm{out} - D_\mathrm{in})}{k(A_\mathrm{t,in} + A_\mathrm{t,out})} + \frac{R_\mathrm{c}}{A_\mathrm{t,out}} + \frac{R_\mathrm{f}}{A_\mathrm{t,out}} + \frac{1}{h_\mathrm{air} A_\mathrm{total} \eta_\mathrm{s}} \tag{20}$$

and under dry surface condition,

$$\dot{m}_{\rm air}(h_{\rm air,in} - h_{\rm air,out}) = \dot{m}_{\rm air} c_{\rm p,air}(T_{\rm air,in} - T_{\rm air,out}) \quad (21)$$

The ε-NTU method, based on the inlet conditions that are known, for cross-flow configuration with one fluid mixed and the other unmixed, is applied to calculate the heat transfer rate between the air and the refrigerant in an iteration-free way.

2.6. Wet surface condition

When the heat transfer surface is at a temperature below the dew point of the passing air stream, condensation of vapor occurs and introduces latent heat transfer in addition to the sensible heat transfer, between the moist air and the heat exchanger surface, which becomes wet in the process.

From a strict point of view, for a finite-length finned tube segment, it is possible that only a part of the outside surface is wetted in either the radial direction or axial direction or in both directions. Identification of surface area below or above the dew point both along the primary surface (tube) and the associated extended surface (fin) appears to be difficult due to the uncertainty affecting the temperature profile, and may be impractical in a general heat exchanger simulation program. In the current model, a segment is assumed to be either completely dry or wet, based on the mean tube/fin surface temperature $\bar{T}_{\rm dry,s}$ calculated under the dry surface condition assumption,

$$\bar{T}_{\rm dry,s} = \eta_{\rm s}(\bar{T}_{\rm w} - \bar{T}_{\rm air}) + \bar{T}_{\rm air} \quad (22)$$

The governing equations of heat and mass transfer, and energy balance over a wetted segment are:

$$Q = \dot{m}_{\rm air}(h_{\rm air,in} - h_{\rm air,out}) - \dot{m}_{\rm air}(\omega_{\rm in} - \omega_{\rm out}) h_{\rm water} \quad (23)$$

$$Q = \dot{m}(h_{\rm ref,out} - h_{\rm ref,in}) \quad (24)$$

$$Q = \frac{\bar{T}_{\rm w} - \bar{T}_{\rm ref}}{\frac{1}{h_{\rm ref}A_{\rm t,in}} + \frac{(D_{\rm out} - D_{\rm in})}{k(A_{\rm t,in} + A_{\rm t,out})} + \frac{R_{\rm c}}{A_{\rm t,out}} + \frac{R_{\rm f}}{A_{\rm t,out}}} \quad (25)$$

$$Q = h_{\rm air} A_{\rm total}(\bar{T}_{\rm air} - \bar{T}_{\rm s}) + h_{\rm d} A_{\rm total} h_{\rm fg}(\bar{\omega}_{\rm air} - \bar{\omega}_{\rm s}) \quad (26)$$

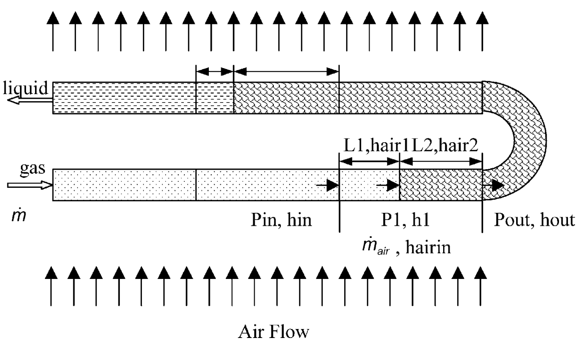

Fig. 5. Schematic diagram of sub-divided segment.

where, the mean surface temperature $\bar{T}_{\rm s}$ is given by

$$\bar{T}_{\rm s} = \eta_{\rm ms}(\bar{T}_{\rm w} - \bar{T}_{\rm air}) + \bar{T}_{\rm air} \quad (27)$$

In the current model, the surface effectiveness $\eta_{\rm ms}$ is calculated using McQuiston's method [17]. Hill's generalized ε-NTU heat flux equation for combined heat and mass transfer is applied when pure refrigerant inside the wetted segment experiences a phase change [18]. For single phase or two-phase nonazeotropic refrigerants, Ragazzi's ε-NTU equation is used in deriving the heat transfer rate of the segment [19].

The refrigerant side heat transfer coefficient $h_{\rm ref}$ in Eqs. (20) and (25) can be obtained by using built-in correlations [20–28], or by using empirical equations or values provided by the user. The correlations of the air side heat transfer coefficient $h_{\rm air}$ in Eqs. (20) and (26) include those from Kim et al. [29,30] and Wang et al. [31].

2.7. Air side mass, energy flow and pressure drop

During heat (and mass) transfer, the air side condition including enthalpy, temperature and humidity also change along the flow path. The mass, energy, and humidity conservation between the neighboring segments in the air flow direction are as follows.

For staggered tube arrangement,

$$\dot{m}_{{\rm air},k} = 0.5(\dot{m}_{{\rm air},i} + \dot{m}_{{\rm air},j}) \quad (28)$$

$$\dot{m}_{{\rm air},k} h_{{\rm air},k,{\rm in}} = 0.5(\dot{m}_{{\rm air},i} h_{{\rm air},i,{\rm out}} + \dot{m}_{{\rm air},j} h_{{\rm air},j,{\rm out}}) \quad (29)$$

$$\dot{m}_{{\rm air},k} \omega_{k,{\rm in}} = 0.5(\dot{m}_{{\rm air},i} \omega_{i,{\rm out}} + \dot{m}_{{\rm air},j} \omega_{j,{\rm out}}) \quad (30)$$

For in-line tube arrangement,

$$\dot{m}_{{\rm air},k} = \dot{m}_{{\rm air},i} \quad (31)$$

$$h_{{\rm air},k,{\rm in}} = h_{{\rm air},i,{\rm out}} \quad (32)$$

$$\omega_{k,{\rm in}} = \omega_{i,{\rm out}} \quad (33)$$

The pressure drop of the air flowing over the heat exchanger is calculated by using appropriate correlations [29–31] or empirical values, according to the tube configuration, fin pattern, and surface condition (dry or wet).

2.8. Sub-dividable segment in case of flow regime change within the segment

CoilDesigner as a general-purpose and flexible simulation and design tool allows for a segment-by-segment approach which can be reduced to a tube-by-tube approach (i.e. the tube is not divided, having only one segment). The tube-by-tube approach is justified especially when air distribution is uniform and/or a single-phase working fluid is flowing inside the tube.

To accommodate for this feature while maintaining the same modeling accuracy, a sub-dividing segment approach is implemented. Fig. 5 shows two tubes in which the refrigerant undergoes de-superheating, condensation and sub-cooling. One tube is located behind the other tube in the airflow direction. Each tube is divided into three segments.

The inlet of the refrigerant is gas (quality ≥ 1), and the ε-NTU equations for the single phase are used to calculate the heat transfer between the refrigerant and the air, along with the outlet refrigerant pressure/temperature/enthalpy, and the outlet air temperature/humidity. The outlet enthalpy of the refrigerant is checked. If it is less than the saturated vapor enthalpy corresponding to the outlet pressure, it means that there is condensation or even sub-cooling taking place somewhere within the segment, and the segment needs to be subdivided into at least two sub segments. An iterative scheme is used to identify the interface where saturated vapor starts to condense, or saturated liquid begins to subcool, and vice-versa for the case of an evaporator.

In view of the entire coil, the particular subdivided segment is treated as an integrated segment by passing its outlet refrigerant pressure/enthalpy to the next segment, and the outlet air temperature to the neighboring 'air-wise' downstream segment(s). The integrated outlet air enthalpy is the sub-segment-percentage averaged enthalpy. When there are three sub segments,

$$h_{\mathrm{air,out}} = \frac{l_1}{l_{\mathrm{seg}}} h_{\mathrm{air1}} + \frac{l_2}{l_{\mathrm{seg}}} h_{\mathrm{air2}} + \frac{l_3}{l_{\mathrm{seg}}} h_{\mathrm{air3}} \tag{34}$$

When there are two sub segments,

$$h_{\mathrm{air,out}} = \frac{l_1}{l_{\mathrm{seg}}} h_{\mathrm{air1}} + \frac{l_2}{l_{\mathrm{seg}}} h_{\mathrm{air2}} \tag{35}$$

3. Simulation studies

Simulation studies are conducted to show the capability of the design tool in predicting all aspects of heat exchanger performance under a wide variation of design and operating conditions. As one example, Fig. 6 shows the mass flow rate through each tube in a coil with arbitrary circuitry. Similar to an electric circuit, tubes connected in parallel decrease the flow resistance, while tubes in series increase the flow resistance, and the mass flow rate through each tube is varied accordingly.

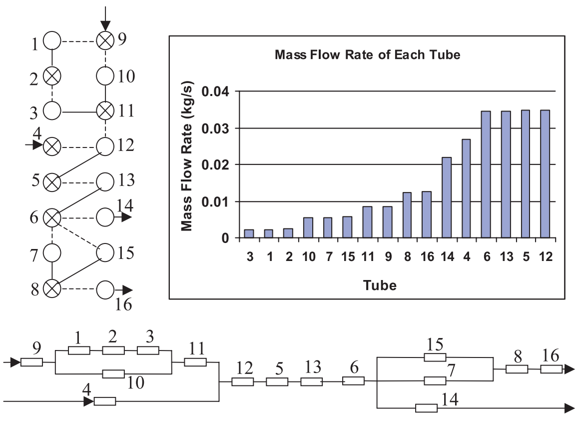

Fig. 6. Mass flow rate of each tube in coil with arbitrary circuitry.

4. Model validation

The performance results of coils predicted with the simulation tool are compared against experimental data collected from the open literature, experiments in laboratories, and other sources, for the purpose of model validation. The experimental data represents coils of diverse geometries, varying operating conditions, different working fluids, and includes microchannel heat exchangers.

4.1. Model agreement with experimental data in literature

McQuiston carried out an extensive set of experiments on plate-fin-tube coils in developing general air side heat, mass and friction coefficient correlations for both wet and dry surface conditions [32,33]. The model verification is conducted by comparing the coil capacity prediction against the test data reported by McQuiston [32].

The air side heat transfer coefficient in the simulation is calculated using the correlation developed by Kim et al. [29], and the water side heat transfer coefficient is computed with the Dittus–Boelter equation.

Fig. 7 shows the comparison of the predicted heat duty with the measured heat duty of the four coils under both dry and wet surface conditions. An overall agreement of 10% is found between the simulation results and the experimental data.

4.2. Comparison of predicted temperature with measured temperature on an intertwined coil

The test data on a condenser coil with two refrigerants in different circuits is used to verify the capability of the design

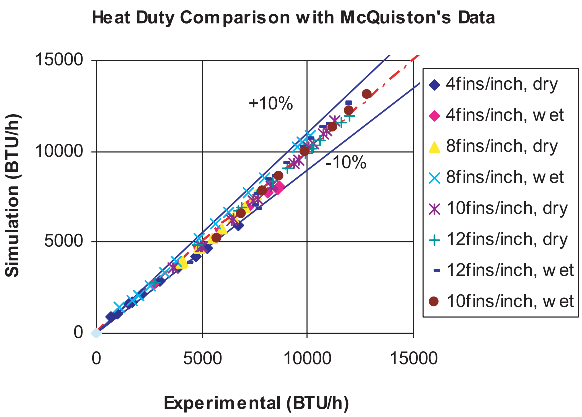

Fig. 7. Heat duty comparison with the experimental data of McQuiston's coils.

tool in handling multiple working fluids in the same coil. Fig. 8 shows the comparison of the simulated temperatures with the measured temperatures along the circuits of two refrigerants, respectively. Nearly all the predicted temperatures are within 1 °F of the measured temperature at each point.

5. Graphic user interface

Compared with many other heat exchanger simulation packages, object-oriented programming techniques are implemented in developing CoilDesigner to facilitate a highly flexible and customizable design platform and to build a graphic user-friendly interface. Features include:

(1) Interactive visual-based, and flexible modeling environment that is quick to learn and easy to use.
(2) Convenience in input, editing, and post processing of data with tabular and graphic representation.
(3) Parametric analysis and graphing to perform sensitivity study, enhance visual interpretation of output information and to closely examine any performance irregularities.

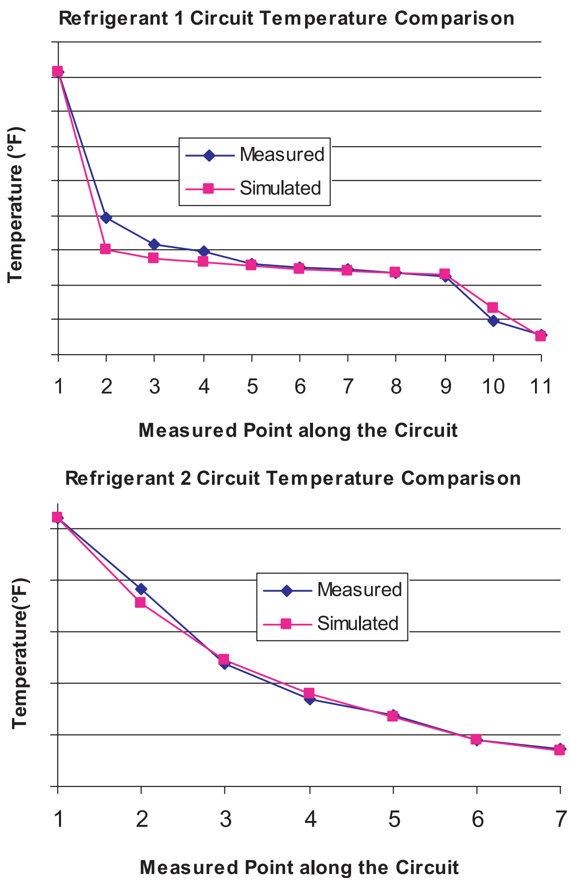

Fig. 8. Simulated vs. measured temperature profiles along circuits.

One of the prominent features of CoilDesigner is its convenience in designing tube circuitry by connecting tube ends in the GUI. Fig. 9 shows the main interface for fin-and-tube coil construction. The left hand view represents the front face area of the coil. Each cell corresponds to the two-dimensional set of air inlet properties including velocity, temperature and relative humidity. The number of cells in a row is equal to the number of segments in a tube. The right hand pane of Fig. 9 is the tube end view. A dotted line indicates a tube connection on the back end (away from the user), and a solid line indicates a tube connection on the front end (towards the user).

CoilDesigner allows the user to choose from any refrigerant that is available in the REFPROP 7.0 [34] property database. The user can also choose from a variety of void fraction models that are used in charge calculation. Another distinguishing feature of CoilDesigner is the option to determine and customize the values or correlations for heat transfer coefficients and pressure drops. Empirical values and public or proprietary correlations are all at the user's choice. A correction factor for using built-in correlations is also incorporated to account for heat transfer surface enhancement or operating condition customization. The references for such built-in correlations are included in the accompanying help file.

6. Conclusions

A comprehensive and flexible, general-purpose simulation and design tool for air-to-refrigerant heat exchangers has been developed. It is applicable to the design of condensers, evaporators, and heating and cooling coils under any operating conditions. Based on a network viewpoint and a flexible and versatile refrigerant circuitry, the tool can address mal-distribution of fluid flow through tubes, and interlaced heat exchangers with different working fluids inside respective subsets of tubes. With a segment-by-segment approach, the impact of two-dimensional non-uniformity of air distribution across the heat exchanger, and the local refrigerant behavior, on the heat exchanger performance can be studied. The freedom in choosing the values or correlations for heat transfer coefficient and pressure drop offers the possibility of accurate performance prediction in the designing or rating of heat exchangers constructed of varied fin and tube geometries, using diverse working fluids, and under a wide range of operating conditions. A highly efficient and intuitive graphical user interface is provided for ease and convenience in engineering use. The prediction results with CoilDesigner are compared with experimentally determined data collected from a number of sources. The simulation tool was shown to predict the heat transfer rate of a variety of coils under a wide range of operating conditions with good accuracy.

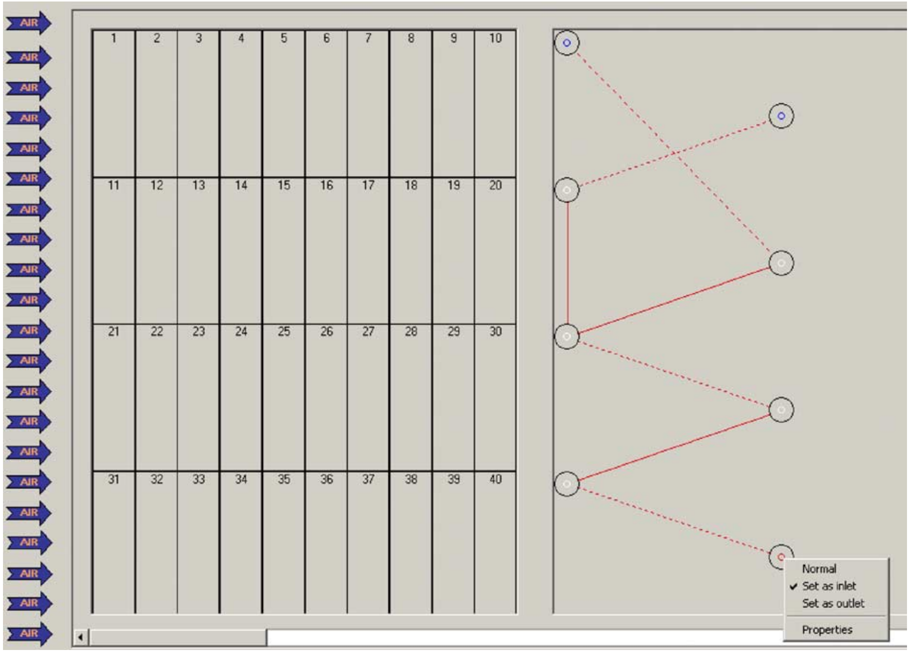

Fig. 9. The main interface for fin-and-tube coil construction.

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