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A simplified explicit model for determining the performance of a chilled water cooling coil

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ABSTRACT

A simplified explicit model for the chilled water cooling coil was presented which could determine the performance of even a partially-wet coil without any need for iterative calculations. A quadratic correlation was derived for evaluating the dry portion of a partially-wet coil from the inlet chilled water temperature based on performance data generated from a validated numerical model for a sample coil over a range of operating conditions. By applying the present explicit model to simulate the performance of the sample coil, it was found that the errors in the calculated coil total capacities did not exceed 2.2% when compared with those based on the numerical model. The percentage errors in the simulated coil latent capacities varied widely from 100% for a just-slightly-wet coil down to 0.26% for a fully-wet coil. Nevertheless, the present model was considered an efficient and yet accurate approach for determining the performance of a chilled water cooling coil.

Un modéle explicite simplifié pour la détermination de la performance d'un serpentin de refroidissement á eau refroidie

Mots clés : Serpentin á eau refroidie ; Conditionnement d'air ; Valeur globale de transfert de chaleur ; Différence de température moyenne logarithmique ; Différence d'enthalpie moyenne logarithmique

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Nomenclature

a	coefficient according to Eq. (59)
A_f	total area of fins (m^2)
A_i	inside surface area of tube (m^2)
A_o	outside surface area of tube (m^2)
A_{oe}	effective outside surface area of tube according to Eq. (20) (m^2)
A_{of}	outside surface area of tube including fins (m^2)
C	capacitance rate ($kW\ K^{-1}$)
C_d	condensing factor as defined in Eq. (7) ($^\circ C^{-1}$)
c_p	specific heat capacity at constant pressure ($kJ\ kg^{-1}\ K^{-1}$)
c_s	rate of change of the specific enthalpy with temperature along the saturated air line in the psychometric chart ($kJ\ kg^{-1}\ K^{-1}$)
dp	dew point temperature ($^\circ C$)
F_{dry}	dry portion of coil which varies from 0 to 1
h	specific enthalpy ($kJ\ kg^{-1}$)
h_s	fictitious specific enthalpy of saturated air ($kJ\ kg^{-1}$)
L	dimensionless length of a coil which varies from 0 to 1
m	mass flow rate ($kg\ s^{-1}$)
N	Number of discretization coil segments
PED	percentage error difference (%)
Q	total capacity of coil (kW)
Qlat	latent capacity of coil (kW)
Q_{ratio}	coil capacity ratio as defined in Eq. (58)
RMSD	root-mean-square difference of the dry portions calculated from the numerical and explicit models according to Eq. (60)
T	temperature ($^\circ C$)
UA	overall heat transfer value of a coil ($kW\ K^{-1}$)
UA_h	overall heat transfer value of a coil as defined in Eq. (30) ($kg\ s^{-1}$)

Symbols

α_i	heat transfer coefficient of inside surface of tube ($kW\ m^{-2}\ K^{-1}$)
α_o	sensible heat transfer coefficient of outside surface of tube ($kW\ m^{-2}\ K^{-1}$)
α_{ow}	global heat transfer coefficient of outside surface of wet tube with fins ($kW\ m^{-2}\ K^{-1}$)
β	parameter defined in Eq. (10)
χ	parameter defined in Eq. (A.5)
Δh_{fg}	specific latent heat of evaporation of water ($kJ\ kg^{-1}$)
Δh_i	specific enthalpy difference between the inlet air and the saturated air at the inlet chilled water temperature ($kJ\ kg^{-1}$)
ΔT_i	temperature difference between the inlet air and chilled water ($^\circ C$)
ϕ	dimensionless parameter according to Eq. (47)
γ	parameter defined in Eq. (23) ($kW\ K^{-1}$)
η_f	fin efficiency
η_{of}	efficiency of outside surface with fins
λ	parameter defined in Eq. (37) ($kg\ s^{-1}$)
ω	humidity ratio ($g\ kg^{-1}$ dry air)

Subscripts

a	air
da	dry air
dp	dew point condition of air
dry	dry coil
dw	interface between dry and wet coil
ew	chilled water
i	inlet
jfd	just-fully-dry
jfw	just-fully-wet
k	designation of coil segment
num	multi-node numerical model
o	outlet
present	present simplified explicit model
Q	total capacity of coil
Qlat	latent capacity of coil
ref	reference condition
sf	tube outside surface
wet	wet coil

1. Introduction

Air-conditioning systems are commonly found in most of the modern buildings particularly the high-rise commercial premises in which cooling is normally required throughout most of the year. High-capacity central chilled water plants are usually employed to provide cooling to such kind of buildings through chilled water cooling coils in air-handling units or fancoils. As the energy consumption by the air-conditioning systems accounts for a substantial proportion of the total energy demand in a non-industrial city, the proper modeling of the air-conditioning system is crucial for understanding the energy performance of the system under different operating conditions. Being one of the major components, the chilled water cooling coil governs the heat transfer effectiveness between the chilled water and the supply air. However, the occurrence of condensation on the coil surface complicates the modeling approach, as both sensible and latent heat transfers coexist in this circumstance. Moreover, the heat transfer coefficient on the outside surface of a wet coil varies with the air state and the surface temperature of the coil. The situation becomes even more complex if only part of the coil is wet. In this regard, the numerical approach (Benelmir and Mokraoui, 2012; Li et al., 2010; Zhou and Braun, 2007a, b; Wang and Hihara, 2003; Wang et al., 2007) was employed in which the entire coil is divided into numerous coil segments. The heat transfer for the whole coil is determined segment-by-segment in an iterative manner until convergence is met. The main merit of the numerical approach is that the variation of the heat transfer coefficients along the coil can be fully taken into account. However, the resultant computation time becomes very long which is particularly unfavorable when a long term dynamic simulation of the entire air-conditioning system is to be performed. To relieve the situation, generalized approaches were developed (ASHRAE, 2012; Wang et al.,