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Simplified thermodynamic modeling of chilled water coils based on bypass factors

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ABSTRACT

This paper aims towards the development of a simple mathematical model for chilled water cooling coil based on the bypass factor as a principal input parameter. The methodology involves a temperature-based discretization of the coil that precisely shows when the coil will be dry, wet or partially wet. This procedure not only depicts the actual cooling–dehumidification profile of the coil, but also allows calculation of sensible and latent coil loads by the summations of individual loads of discrete coil elements. Maximum deviations in the range of ±5% for sensible and total coil loads are obtained between majorities of the findings from the theoretical model and experimental data available in literature. A close compliance with actual results and ease of application make this mathematical model a good choice for energy simulation tools.

1. Introduction

With an increased awareness for the harmful environmental implications of over-usage of conventional energy resources and ever-increasing energy demand, it becomes imperative that the present energy resources be used in an efficient and sustainable manner. In building sector, air conditioning is a major energy-consuming exponent. Hence, economic and efficient consumption of energy for air conditioning applications is of foremost importance, if global energy conservation is to be realized.

A finned tube heat exchanger (or simply a coil) is one of the main mechanical components of any air conditioning equipment. It forms a thermodynamic link between the refrigerant and conditioned air that is necessary to create comfortable conditions inside the built environments. Therefore, accurate estimates of the heat transfer rates through these coils are necessary to optimize the operation of whole equipment.

Threlkeld has developed the earliest detailed model for cooling–dehumidification coils [1]. His model is based on the Log Mean Enthalpy Difference (LMED) method by assuming a linear relationship between the saturation enthalpy and temperatures of coolant and coil tube. Many researchers like Elmahdy [2–4] and Braun [5] took Threlkeld's model as a reference to propose new coil models. Elmahdy et al. have conducted detailed and extensive experimentation on two kinds of chilled water coils for validating Threlkeld's LMED model [3]. Braun, on the other hand, has developed a steady state ε-NTU model for a cooling–dehumidifying coil in counter flow configuration that is applicable for dry, wet and partially wet coil conditions [5].

Researchers and investigators have used the heat transfer rates obtained from earlier models to develop new methodologies. The standard method as per ASHRAE Equipment Handbook [6] and AHRI Standard 410 [7] consider overall dry surface heat transfer by the Log Mean Temperature Differences (LMTD) between coolant and air stream, whereas for wet surface heat transfer, the Log Mean Enthalpy Differences (LMED) between the surface condition and air stream, are considered. Wang and Hihara [8] have developed a cooling–dehumidification model for chilled water coils based on Equivalent Dry-bulb Temperatures (EDT) to calculate the heat and mass transfer rates and predict cooling modes of the coil. Pirompugd et al. [9] proposed new correlations to calculate heat and mass transfer rates based on extensive experimentation on fin-and-tube heat exchangers under cooling–dehumidification conditions. Xia et al. [10] developed a generalized LMED model, which is based on non-unity Lewis number for calculating the heat and mass transfer rates through an air-cooling coil under wet condition.

To simplify the analytical complexity of cooling–dehumidifying coils for air conditioning applications and to predict and evaluate their performance, many researchers have proposed various numerical methods. Mirth and Ramadayani [11] simplified a cooling coil as a counter flow heat exchanger to develop a numerical model that involves discretization of coil along air path and application of energy balance principles for each step till the end of

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Nomenclature

Symbols

A_{in}	area of inner side (water side) of chilled water coil (m^2)
A_{out}	area of outer side (air side) of chilled water coil (m^2)
BF	overall bypass factor of coil
C_{pa}	specific heat capacity of dry air = $1.006\,\mathrm{kJ\,kg^{-1}\,K^{-1}}$
C_{pm}	specific heat capacity of moist air $\sim 1.02\,\mathrm{kJ\,kg^{-1}\,K^{-1}}$
C_{pl}	specific heat capacity of liquid water = $4.186\,\mathrm{kJ\,kg^{-1}\,K^{-1}}$
C_{pv}	specific heat capacity of water vapor = $1.86\,\mathrm{kJ\,kg^{-1}\,K^{-1}}$
ΔC_{plv}	difference of specific heats of liquid and vapor: $2.326\,\mathrm{kJ\,kg^{-1}\,K^{-1}}$
h_a	specific enthalpy of air ($\mathrm{kJ\,kg^{-1}}$)
h_s	saturation enthalpy of air at effective surface temperature ($\mathrm{kJ\,kg^{-1}}$)
l_0	specific latent heat of vaporization at 273 K = $2501\,\mathrm{kJ\,kg^{-1}}$
$\dot{m}_a$	total mass flow rate of air ($\mathrm{kg\,s^{-1}}$)
$\dot{m}_v$	mass flow rate of water ($\mathrm{kg\,s^{-1}}$)
P_t	ambient pressure (kPa)
$\dot{Q}_t$	coil load capacity (kW)
q	specific enthalpy change across the coil ($\mathrm{kJ\,kg^{-1}}$)
T_a	temperature of air (°C)
T_s	effective surface temperature (ADP) of coil (°C)
T_w	temperature of chilled water (°C)
U_{in}	overall heat transfer coefficient on water side $\mathrm{kW\,m^{-2}\,K^{-1}}$
U_{out}	overall heat transfer coefficient on air side $\mathrm{kW\,m^{-2}\,K^{-1}}$

Greek Symbols

ω	Humidity ratio (kg moisture/kg dry air)

Subscripts

0	reference state 273.15 K
a	of air
avg	averaged
des	design (or desired)
dp	dew point
in (or 1)	coil inlet condition (on coil)
out	coil outlet condition (off coil)
latent	latent part of coil load
sen	sensible part of coil load
sen-dry	sensible part of load capacity for a dry coil
sen-wet	sensible part of load capacity for a wet coil
x	of a coil element

coil. Vardhan et al. [12] extended Mirth's work by discretizing the coil into nodes along coolant path and by applying iterative calculation for convergence of air properties at each node from coolant inlet to outlet. Mansour et al. [13] employed a row-by-row method for designing finned tube cooling coils and ascertaining its cooling–dehumidification profile.

In order to determine annual energy consumption by cooling coils, Morisot et al. [14], Lemort et al. [15] and Chillar et al. [16] have developed simple coil models for energy modeling applications. These models derive the required heat transfer resistances applicable at non-nominal operating conditions from available catalogue data at standard conditions. In recent years, dynamic modeling has become more popular in order to propose various controlling strategies for efficient functioning of cooling–dehumidifying coils. Among many of the works done on dynamic performance modeling, the work by Yiu et al. [17], Yao et al. [18] and Sekhar et al. [19] are mentioned here. Yiu and his colleagues have developed a simulation model for the dynamic performance of dry and wet cooling coils. Yao et al. studied the dynamic relationship between the coil's heat exchanges and various operating parameters by applying classical control theory. Sekhar's work is mainly concentrated on the development of optimized controlling strategies for enhanced dehumidification performance of oversized cooling coils for hot and humid climates. All the cooling coil models or methodologies mentioned here have their own pros and cons, and their application is entirely based on the objectives targeted.

Most of the thermodynamic modeling techniques for air conditioning coils thrusts more upon the accuracy of the of heat exchanger framework rather than on their simplification and application for load estimation and energy simulations. These methods often require detailed inputs of cooling coils and implement iterative procedures for convergence of solutions for coil loads and supply temperatures. However, most of control sequences for cooling equipment link supply temperatures or mass flow rates or both with the feedback from zone temperatures. Hence, in energy simulations the intended supply and controllable operating conditions are affixed based on zone conditions. These parameters are also determined and included in design's specifications based on the type of applications, climate and equipment.

In a similar way, the bypass factors (BF) of cooling coils at design load conditions are available via AHU manufacturer's catalogue or customized selection software depending on the design zone load ratios, application and type of equipment selected. The chilled water supply temperatures and the net water temperature difference across the coil are obtained from chiller selection and water distribution network details, respectively. Considering all these factors, it can be surmised that modeling cooling coils with computational complexities in conjugation with whole building energy simulation tools are not required.

On the other hand, the coil modeling methodology applied in many load estimation and energy simulation tools neither substantiate the air supply temperature properly based on the inputs provided for chiller plant, nor determine the chilled water flow rate at the correct delta *T* and realized coil apparatus dew point temperature (ADP). For example, these tools do not model the chiller and pumping energy consumptions accurately, when the coil is incorrectly selected at a different delta *T* than the plant. Literature study reaffirms that this is one of the major reasons for inefficient functioning of chillers [20]. In addition to this, many of these tools do not consider changes in BF with airside parameters (primarily the air mass flow rate) at part load conditions, which results in incorrect coil loads. Considering the fact that an air-handling unit runs on part loads more than 90% of operational time, this will lead to faulty estimation of annual cooling energies.

Hence, a relatively accurate but simplified methodology is required to model the cooling and dehumidification processes through a chilled water coil. A method is shown here that addresses the solution to this problem. The overall objectives of the present work are summarized as follows:

- Development of a simple chilled water cooling coil with bypass factor and either of chilled water delta *T* or required supply temperature as an additional input, instead of any detailed physical data for the coil.
- Representation of cooling–dehumidification mode (wet, dry or partially wet) of the coil at various operating conditions.
- Computation of sensible and latent loads of the coil by a discrete load summation or an 'element-by-element' method.
- Computation of coil loads at variable conditions by determining coil bypass characteristics at a reference performance point.