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Advancing evaporative rooftop packaged air conditioning: A new design and performance model development

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ABSTRACT

This paper presents a technological advancement in evaporative cooling rooftop air conditioning comprising a uniquely designed evaporative water cooler that includes a multi-stage hydronic unit and high thermal performance. In the new design, the water cooler is a counterflow air-to-air heat exchanger in which ambient air is pre-cooled in a dry path on one side of a heat transfer surface by water flowing on the other side of the surface. The water is then cooled by evaporation in a wet path by a secondary air stream flowing through the heat exchanger on the same side as the water but in the opposite direction. Outside air is cooled in the dry passages and then enters the wet passages at a lower wet bulb temperature than that of the outdoor air, potentially producing a lower sump water temperature compared to those produced by traditional evaporative condensers. We also developed a computer model to simulate the performance of the rooftop packaged unit. The model is based upon the Simulation Problem Analysis and Research Kernel (SPARK) simulation program and can be used to optimize component sizes and to perform an economic analysis. In addition, the model can be used for fault detection and diagnosis during operation. The simulation model was calibrated with experimental data obtained from the study and was then used for optimal sizing and performance tracking.

1. Introduction

According to the US Energy Information Agency (EIA), buildings account for 39% of the primary energy use in the US and use approximately 39 Quad in 2006. Space cooling and heating collectively account for one-third of the total primary energy use in buildings [1]. Rooftop packaged air conditioners and other unitary HVAC devices are used extensively for space cooling and heating in commercial buildings in the US. They are particularly common in small commercial buildings, such as retail stores in strip malls, school buildings, and office buildings [2]. Packaged air conditioning units account for 44% of all cooling equipment and are responsible for cooling approximately half of all commercial building spaces in the US. However, rooftop packaged units are often designed and installed poorly, and they are prone to faults and malfunctions during operation and control. Unlike large built-up HVAC systems, most of which are commonly managed by dedicated building operators who are responsible for their daily operation and maintenance, rooftop packaged units are typically installed in smaller buildings, where the owners tend to be more cost-sensitive and often capital constrained, i.e., have a tighter budget resulting in fewer resources for daily operation and lower levels of responsiveness to troubleshooting.

In typical rooftop packaged units, the direct expansion evaporator coils are usually not designed to have uncontrollable surface temperatures, which often cause unnecessary latent cooling. In addition, dust from the air can deposit onto the closely spaced fins of the coils; this dust causes condensed moisture droplets to accumulate due to latent cooling. With the help of elevated moisture or condensed water, the dust may bridge the narrow gaps between the fins, resulting in significantly lower efficiency in heat and mass transfer through the coils. While it is difficult to quantify the fraction of rooftop packaged units in the market operating with significantly lower efficiency or with degradation or faults, some studies in the literature have indicated that energy savings of 10–30% are possible from improving the operation of rooftop units [3]. Katipamula and Brambley (2004) estimated that the potential annual energy savings of improving the design and operation of rooftop packaged units on buildings in California ranged from 2 to 7 trillion BTUs ($0.58–2.1 \times 10^8$ kWh) [4]. Researches aiming at improving the performance of rooftop packaged air conditioning systems have been conducted widely. Active desiccants have been

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Nomenclature

A	heat exchanger surface area (m^2)
c	specific heat (W/kg K)
C	constant
d	thickness (m)
D	pump nominal diameter (cm)
D_v	diffusion coefficient (m^2/s)
evp	evaporative mode
h	convective heat transfer coefficient (W/m K)
heat	heating mode
h_{fg}	heat of vaporization of water (W/kg)
h_m	mass transfer coefficient (kg/s m^2)
k	thermal conductivity (W/m K)
K_m	mass transfer coefficient (Lewis number = 1) (m/s)
L	characteristic length (m)
m	mass flow rate (kg/s)
ΔM	mass change (kg/s)
n	pump nominal rotation speed (rpm)
p	pressure rise (meter of water)
Q	heat transfer rate (W)
Re	Reynolds number
RH	relative humidity
Sc	Schmidt number
Sh	Sherwood number
sing	single loop mode
T	temperature (°C)
U	Total conductance (W/m K)
V	air velocity (ft/s) (m/s)
w	humidity ratio (kg/kg)

Greek letters

ρ	air density (kg/m^3)
μ	absolute viscosity (kg/m.s)

Subscripts

dry	dry passage
in	inlet
lat	latent heat
out	outlet
sen	sensible heat
wet	wet passage
vent	ventilation

Acronyms

SPARK	Simulation Problem Analysis and Research Kernel
EIA	US Energy Information Agency
HVAC	heating, ventilation, and air conditioning
CEWC	counterflow evaporative water cooler

applied in conjunction with rooftop packaged equipment in the past to address increased ventilation air pre-conditioning concerns in restaurants or theaters. This approach did not find market acceptance for several reasons, including first cost, operational cost, and energy efficiency [5]. The research conducted by James and John [6] tackled this problem by designing a new active desiccant integration with the equipment. Yu et al. [7] studied the impact of physical air flow monitoring meter on the performance of rooftop packaged air conditioning since low SCFM directly impairs temperature distribution and causes poor IAQ.ASHRAE standard 62.1-2007 [8] specifies ventilation and circulation air flow rate based on the occupancy and floor area.

The objectives of this study were to advance the design of rooftop packaged units using evaporative cooling and to develop a computer model capable of simulating the performance of the newly designed rooftop unit. This model can then be used to optimize the new design and to provide an economic analysis.

2. Methodology

In this paper, we first present a technological advancement in evaporative cooling rooftop air conditioning comprising a unique, recently developed design for an evaporative water cooler, followed by the details of the development of a computer model to simulate the performance of the rooftop packaged unit. The model is based upon the SPARK [9] simulation program and can be used for optimal component sizing and economic analysis and potentially for fault detection and diagnosis during operation. The new simulation model will be calibrated with measured data from actual experiments and tests.

The new information produced, along with the new design, is expected to promote the wider adoption of evaporative cooling systems. The future implementation of this new efficient and reliable system in commercial buildings on even a modest scale can provide significant savings in electricity use due to the improved efficiency.

2.1. Technology advancement – new design

In response to the low operating efficiency that is often associated with conventional refrigerant-based rooftop packaged units, we designed and constructed a hydronic rooftop packaged unit with a significant efficiency improvement (Fig. 1). We also developed field measurements and performed computer modeling to quantify the working system's energy savings. This new system is an evaporative cooling unit constructed on the rooftop of a commercial building. The rooftop unit has five major hydronic components: a water cooler, a water-to-air coil, pumps (one for the condenser loop and one for the evaporator loop), and a gas water heater.

In the new design, the water cooler is a counterflow air-to-air heat exchanger in which ambient air is pre-cooled in a dry path on one side of the heat transfer surface by water flowing on the other side of the surface. Water is then cooled by evaporation in a wet path by a secondary air stream flowing through the heat exchanger on the same side as the water but in the opposite direction. Outside air is cooled in the dry paths and then enters the wet paths with a lower wet bulb temperature than that of the outdoor air, potentially producing lower sump water temperatures compared to those produced by evaporative condensers.

Compared to conventional refrigerant-based rooftop packaged units, this new evaporative cooling unit has several new features:

- The new unit includes a high-performance counterflow evaporative water cooler (CEWC), which also functions as a ventilation air pre-cooler, heat recovery device for exhaust air, and an evaporatively cooled condenser to reduce compressor energy consumption. The CEWC cools water, which is then used in the condenser to cool the refrigerant. The ventilation air is pre-cooled in the CEWC to near or below the set point.
- The rooftop unit has five operation modes, one for heating and four for cooling. The four cooling modes are the ventilation cooling (free cooling), evaporative cooling, single loop cooling,

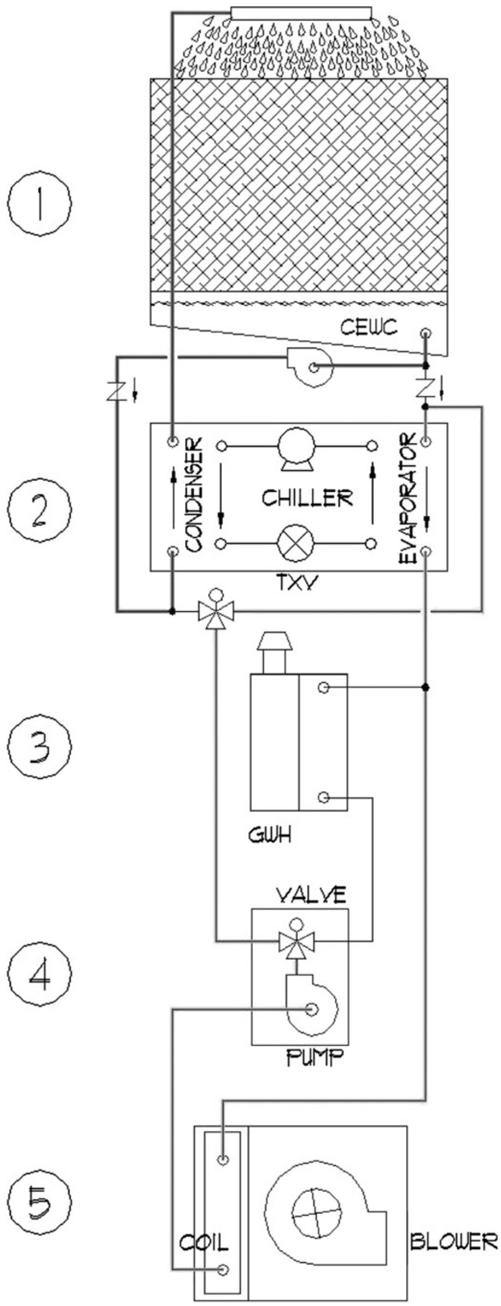

Fig. 1. Rooftop schematic diagram (single loop).

and split loop cooling modes. In the ventilation cooling mode, the outside air is introduced to the room directly when the outside temperature is less than the required supply air temperature. In the evaporative cooling mode, the chiller is turned off, and the water from the CEWC is used as chilled water by the coil. In the single loop mode, the chiller is turned on, and the water from the CEWC is further cooled by the evaporator before entering the coil. In the split loop mode, the unit runs as a traditional water-cooled rooftop unit. The CEWC functions as a cooling tower, and there are separate water loops for the evaporator and the condenser. As the cooling load increases with the outdoor air temperature, the rooftop unit switches from the ventilation cooling mode to the evaporative cooling mode, then to the single loop mode and finally to the split loop mode sequentially to meet the load requirements.

- The new design allows heat from the blower motor to be removed from the building's exhaust air stream, which prevents this heat from being added to the supply air stream during cooling.
- The new design uses a larger coil that allows the chilled water to have a higher temperature while delivering the same cooling capacity with a lower load on the compressor. A large coil is used to avoid unnecessary latent cooling and to reduce the latent cooling fraction. A pump and a motorized valve circulate water to the water-to-air coil and back to the chiller. A second pump circulates water from the CEWC to the condenser and to the pre-cooling coil for ventilation air. The new, improved unit is designed to potentially reduce HVAC electrical energy consumption and peak demand by more than 65% in dry climates and 50% in humid climates.
- The CEWC is a counterflow air-to-air heat exchanger in which ambient air is pre-cooled on one side of the heat transfer surface by water flowing on the other side of the surface (Fig. 2). The output of the CEWC is pre-treated fresh air for ventilation and chilled water for the water-to-air coil. Outside air is drawn down from the left side in a dry passage, and the

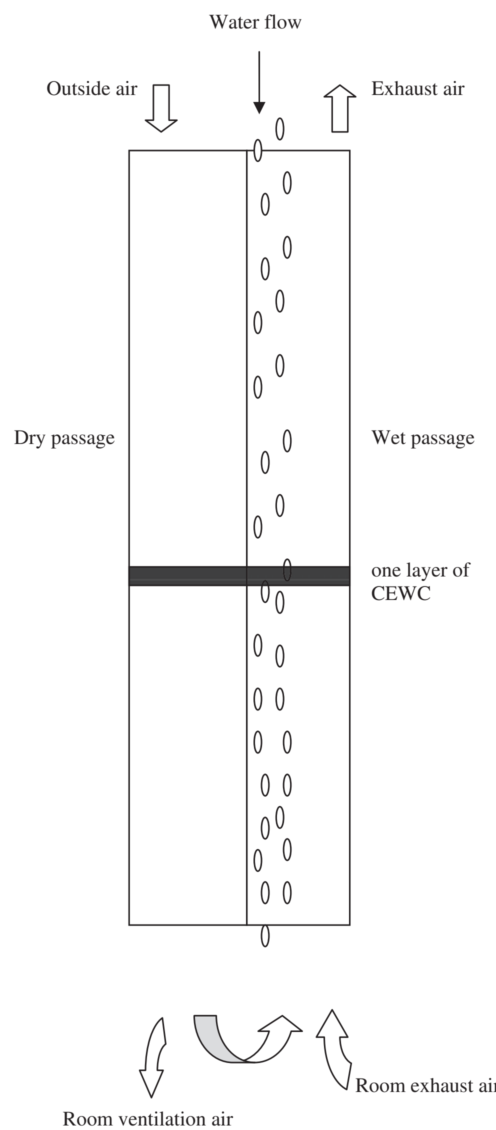

Fig. 2. CEWC diagram (one pair of dry/wet passages).

air is cooled by the water flow through a separation panel on the right side. Part of the air will be delivered to the room for ventilation. The majority of the air will mix with the exhaust air stream and make a U-turn at the bottom of the CEWC unit to go up through the wet passage on the right side. The water is evaporatively cooled by this mixed air stream flowing through the same side but in the opposite direction of the water flow. Sensible heat is transferred from the dry air to the surface and from the surface to the water. Both sensible and latent heat are transferred from the water to the secondary air.

2.2. Development of computer model for simulation

We developed a computer model to simulate the performance of the new rooftop unit. The model was used to study design variations in a range of climates. Each component was modeled using non-linear algebraic equations that describe either heat and mass transfer or fluid flow. The equation set was solved simultaneously using the Newton–Raphson method. Some model equations were based on standard thermal fluid analysis; others were derived from empirical measurements. After calibration, the model was used with building load data to predict the system's performance and operating costs under different load and ambient conditions. The model will also enable us to determine optimal component sizing and compare design alterations with conventional rooftop units.

The model of the CEWC was linked to the models of other components, including a compressor, an evaporator, a condenser, a water-to-air coil, three fans and two pumps, each of which was implemented as a simulation object. The models of the individual components were developed and tested separately; these component models were then interconnected to specify the model of the entire system. The different operating modes of the rooftop unit involved different combinations of the components, which were sometimes connected in different ways. The same component and subsystem models were used for the different configurations. As a result, the sizing of the main components of the rooftop unit could then be optimized by comparing the energy costs predicted by the model with the cost of different component sizes.

3. Simulation model

The following section presents the details of the model development and discusses how the model was used to verify the system's performance. This section includes the documentation of the performance model, including the structure of the rooftop unit model and a detailed description of the model at different levels.

3.1. Computer modeling of system performance

The simulation program SPARK (Simulation Problem Analysis and Research Kernel) was used to develop the performance model of the unit. SPARK is an object-oriented software system that can be used to perform simulations of physical systems modeled using sets of differential and algebraic equations [9]. 'Object-oriented' simulation means that the components and subsystems are modeled as objects that can be interconnected to specify the model of the entire system. Often the same component and subsystem models can be used in many different system models, reducing the cost of development.

The process of describing a problem to produce a SPARK model begins by breaking the problem down in an object-oriented way. This involves thinking about the system in terms of its components so that a SPARK object can represent each component. Then, a model is developed for each component that is not already available in a SPARK library. Because there may be several components of the same type, SPARK object models, i.e., equations or groups of equations, are defined in a generic manner, called classes. Classes serve as templates for creating any number of like objects that may be needed in a problem. The problem model is then completed by linking the objects together, thus defining how they interact, specifying data values that specialize the model to represent the actual problem to be solved and providing boundary values.

SPARK models have a hierarchical structure. The smallest programming element is a class consisting of an individual equation, called an atomic class. Atomic classes are saved as files with the extension "cc". A macro class consists of several atomic classes (and possibly other macro classes) combined together into a higher level unit. Macro classes are saved as .cm files. Problem models are similarly described using the atomic and macro classes and placed in a problem specification file. When SPARK processes the problem, the problem specification file is converted to a C++ program, which is compiled, linked and executed to solve the problem for a particular set of boundary conditions specified at the run-time.

3.2. Model structure

Because SPARK models have a hierarchical structure, three levels of models were developed for the new system:

1. individual component models (eight components including the CEWC, fan, pump, etc.);
2. operating mode models (five operation modes);
3. the model of the complete rooftop system.

These component models can be tested and calibrated separately against the experimental data or catalog data. The models for each operating mode link all the mechanical component models together in particular configurations. The model combines the operating mode level models together with the control strategy to select the appropriate operating mode. The structure of the rooftop unit model is shown in Fig. 3. Some of the component models consist of lower level sub-component macro classes and atomic classes. For example, in the chiller component model, there are the sub-component models of the condenser and the evaporator. In these sub-component models, the more fundamental atomic classes, such as the U value class, were used to describe individual physical processes. The rooftop unit has five major hydronic components: the CEWC, the chiller, the water-to-air coil, the pumps (one for the condenser loop and one for the evaporator loop), and the gas water heater. The unit has three fans: the CEWC fan, the ventilation fan, and the supply fan. In total, there are eight subsystems and components in the model.

In this paper, we use a bottom-up approach to explain how the model and its processes are structured in a step-by-step manner.

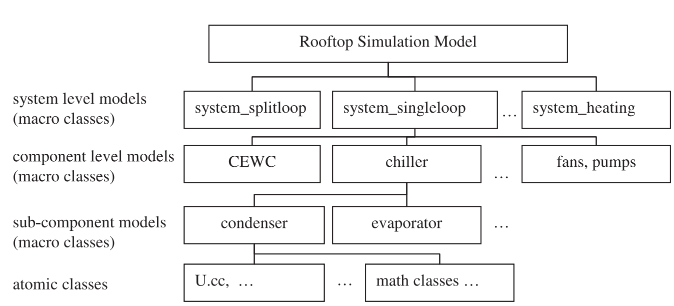

Fig. 3. The hierarchical structure of the rooftop unit model.

First, the component level models are described; then the operating mode models and finally the complete rooftop unit model are described.

3.2.1. CEWC model

The CEWC was divided into a significant number of discrete, fixed sections in the direction of the air flow. The driving potentials for heat and mass transfer were taken to be constant in each section, allowing the heat and mass transfer processes between the three fluid streams to be modeled separately but solved simultaneously. The CEWC was modeled by subdividing the heat exchanger into 20 layers along the direction of the fluid flow. In each layer, the heat and mass transfer equations were defined in six automatic classes. These six classes were combined in a macro class that describes all the physical processes occurring in that layer. The 20 layer objects were combined in a higher level macro class, which described the whole heat exchanger. The layers were coupled by defining the fluid input conditions of the downstream layer as the fluid output conditions of the adjacent upstream layer. The macro class that defined the whole CEWC unit consisted of the combination of the 20 heat exchange models and the air-mixing model, which determines the condition of the air in wet passages as a mixture of air from the dry passages and air from the room. The detailed equations and functions of each class are described below.

3.2.1.1. Heat and mass transfer coefficients. The conductance between the air-dry passage and the water flow is determined from [9]:

$$h_{\text{air}} = C_1 \cdot (0.99 + 0.21 V_{\text{air}}) \tag{1}$$

$$h_{\text{water}} = C_2 \cdot m_{\text{water}}^{0.8} \tag{2}$$

$$U = \frac{1}{\left(\frac{1}{h_{\text{air}}} + \frac{1}{h_{\text{water}}} + \frac{d}{k}\right)} \tag{3}$$

where V_{air} is the air velocity in the dry passage (m/s), m_{water} is the water mass flow rate (kg/s), k is the thermal conductivity of the plate media (W/m K), d is the thickness of the heat exchange surface (m), h_{water} is the convective heat transfer coefficient between water and the surface (W/m K), h_{air} is the convective heat transfer coefficient between air and the surface (W/m K), U is the total conductance (W/m K), and C_1 and C_2 are two constants determined from the CEWC test results.

The heat and mass transfer between the air in the wet passages and the water stream are modeled using the following equations:

$$Re = \frac{\rho_{\text{air}} \cdot V_{\text{air}} \cdot L}{\mu} \tag{4}$$

$$Sc = \frac{\mu}{\rho_{\text{air}} \cdot D_V} \tag{5}$$

$$Sh = 0.664 \cdot Sc^{0.333} \cdot Re^{0.5} \tag{6}$$

$$h_m = \frac{Sh \cdot D_V}{L} \tag{7}$$

$$K_m = h_m \cdot \rho_{\text{air}} \tag{8}$$

where D_v is the diffusion coefficient (mass diffusivity) (m^2/s); L is the characteristic length (m); μ is the absolute viscosity (kg/m s); Re is the Reynolds number, which is dimensionless; Sc is the Schmidt number, which is dimensionless; Sh is the Sherwood number, which is dimensionless; h_m is the mass transfer coefficient (kg/s m^2); K_m is the mass transfer coefficient (Lewis number = 1) (m/s); and ρ_{air} is the air density (kg/m^3).

3.2.1.2. Heat and mass transfer in one layer. In the dry passages, the governing equations of the heat transfer are:

$$Q_{\text{Sen,dry}} = UA\left(\frac{T_{\text{water,in}} + T_{\text{water,out}}}{2} - \frac{T_{\text{air,dry,in}} + T_{\text{air,dry,out}}}{2}\right) \tag{9}$$

$$Q_{\text{total,dry}} = Q_{\text{sen,dry}} \tag{10}$$

where $Q_{\text{sen,dry}}$ is the sensible heat transfer on the dry passage (W), $Q_{\text{total,dry}}$ is the total heat transfer on the dry passage over one layer (W), $T_{\text{water,in}}$ is the water temperature at the inlet of one layer (°C), $T_{\text{water,out}}$ is the water temperature at the outlet of one layer (°C), $T_{\text{air,dry,in}}$ is the dry passage air temperature at the inlet of one layer (°C), $T_{\text{air,dry,out}}$ is the dry passage air temperature at the outlet of one layer (°C), and A is the heat exchanger surface area.

In wet air passage, the governing equations are (ASHRAE, 2009):

$$Q_{\text{Sen,wet}} = h_{\text{sen}}A\left(\frac{T_{\text{water,in}} + T_{\text{water,out}}}{2} - \frac{T_{\text{air,wet,in}} + T_{\text{air,wet,out}}}{2}\right) \tag{11}$$

$$h_{\text{sen}} = K_m c_p \tag{12}$$

$$Q_{\text{Lat,wet}} = K_m A\left(w_i - \frac{w_{\text{air,in}} + w_{\text{air,out}}}{2}\right) h_{\text{fg}} \tag{13}$$

$$Q_{\text{total,wet}} = Q_{\text{sen,dry}} + Q_{\text{lat,dry}} \tag{14}$$

where $Q_{\text{sen,wet}}$ is the sensible heat transfer on the wet passage over one layer (W), $Q_{\text{lat,wet}}$ is the latent heat transfer on the wet passage over one layer (W), $Q_{\text{total,wet}}$ is the total heat transfer on the dry passage over one layer (W), h_{sen} is the sensible heat transfer coefficient (W/m^2), $T_{\text{air,wet,in}}$ is the wet passage air temperature at the inlet of one layer (°C), $T_{\text{air,wet,out}}$ is the wet passage air temperature at the outlet of one layer (°C), $w_{\text{air,in}}$ is the humidity ratio of the air in the wet passage at the inlet of one layer (kg/kg), $w_{\text{air,out}}$ is the humidity ratio for the air in the wet passage at the outlet of one layer (kg/kg), w_i is the air–water interface humidity ratio (saturated air with water temperature) (kg/kg), h_{fg} is the heat of vaporization of water (W/kg), and c_p is the specific heat of the air at constant pressure (W/kg K).

The energy balances for the dry side air, wet side air and water streams are:

$$Q_{\text{air,dry}} = m_{\text{air}} \cdot c_p (T_{\text{air,in}} - T_{\text{air,out}}) \tag{15}$$

$$Q_{\text{air,wet}} = m_{\text{air}} \cdot c_p (T_{\text{air,wet,in}} - T_{\text{air,wet,out}}) \tag{16}$$

$$Q_{\text{water}} = m_{\text{water}} \cdot c_{\text{water}} (T_{\text{water,in}} - T_{\text{water,out}}) \tag{17}$$

where $Q_{\text{air,dry}}$ is the internal heat change of air on the dry passage over one layer (W), $Q_{\text{air,wet}}$ is the internal heat change of air on the wet passage over one layer (W), Q_{water} is the internal heat change of water flow over one layer (W), c_{water} is the water specific (W/kg K), $m_{\text{water,in}}$ is the water mass flow at the inlet of one layer (kg/s), and m_{air} is the air mass flow rate through one layer (kg/s).

The water mass balances for the water and moist air streams are:

$$\Delta M_{\text{water}} = m_{\text{water,out}} - m_{\text{water,in}} \tag{18}$$

$$\Delta M_{\text{water,air}} = m_{\text{air}}\left(w_{\text{air,in}} - w_{\text{air,out}}\right) \tag{19}$$

where $m_{\text{water,out}}$ is the water mass flow at the outlet of one layer (kg/s), ΔM_{water} is the mass change of the water flow over one layer (due to evaporation) (kg/s), and $\Delta M_{\text{water,air}}$ is the water mass change of the air flow (wet passage) over one layer (due to evaporation) (kg/s).

3.2.2. Other component models

The water-to-air coil was modeled as a counterflow heat exchanger using the standard NTU method [10]. We used a manufacturer-specific chiller model to describe its performance. The chiller model consists of a compressor model, an evaporator model and a condenser model. The chiller cooling capacity and power consumption were modeled by curve fitting the catalog data. The water temperature change at the condenser and evaporator were simulated using standard heat balance functions [11]. The unit has two pumps: one pump for the condenser loop (when the system runs in the split loop mode) and one pump for the main loop. Separate pump models have been built for the two pumps in case one pump is modified in the future. The pump model is a fifth order polynomial curve fitting [12] as follows:

$$m_0 = \rho \cdot n \cdot D^3 \tag{20}$$

$$p_0 = \rho \cdot n^2 \cdot D^2 \tag{21}$$

$$m_1 = \frac{m}{m_0} \tag{22}$$

$$p_1 = a_0 + a_1 \cdot m_1 + a_2 \cdot m_1^2 + a_3 \cdot m_1^3 + a_4 m_1^4 \tag{23}$$

$$p = p_0 \cdot p_1 \tag{24}$$

where n is the pump nominal rotation speed, which is 3250 for the pump provided (rpm); D is the pump nominal diameter, which is 5 cm; m_0 is the water flow under nominal conditions; p_0 is the pressure increase across the pump under nominal condition (m of water); m_1 is the dimensionless flow (kg/s); p_1 is the dimensionless pressure; m is the water mass flow rate (kg/s); p is the pressure rise across the pump (meter of water); and ρ is the water density, 62.4 (1000 kg/m^3).

The model has four hydraulic system loops: one for the heating mode, one for the single loop mode, one for the condenser loop in the split loop mode, and one for the evaporator loop in the split loop mode. We conducted a series of tests to map the relationship between the system pressure drop in each loop and the flow rate. The relationship between the mass flow rate and the pressure drop was modeled by second order polynomial curve fitting. The three fans in the rooftop unit, i.e., the supply fan, the CEWC fan and the ventilation fan, are all variable speed fans. In the fan model, the rotation speed was calculated from the airflow rate and the pressure increase across the fan. The supply fan was simulated by curve fitting the catalog data.

3.3. Operation modes

SPARK has no “if and then” functions in the macro class high level. Therefore, in the system class, all the configuration data of each rooftop unit component were passed to the ports of each component class directly. The other ports of the component models were linked with each other to produce the operating mode level models. The rooftop unit has five running modes; therefore, the model has five operating mode level objects. Regardless of which operation conditions the rooftop unit is operating under, the simulation engine runs the simulation for all of the conditions and chooses which output data to report through a gate. The governing equations below show how the components were linked together to form these operating mode level models.

3.3.1. Split loop mode

The rooftop unit runs like a traditional water-cooled rooftop unit. The unit runs with two separate water loops: one for the condenser and one for the evaporator. In the air stream, the outside air is pre-cooled by the CEWC and mixed with the return air from the room. This air mixture is then passed through the water-to-air coil and delivered into the room.

3.3.2. Single loop mode

The air flow is the same as in the split loop mode, but there is only one water loop. The sump water from the CEWC is further cooled by the evaporator before entering the water-to-air coil. The water leaving the water-to-air coil is sent to the condenser, and the water leaving the condenser is sent to the CEWC.

3.3.3. Evaporative cooling mode

The class of the evaporative cooling mode is the same as in the single loop mode. All the port links are the same except the chiller is turned off. The chiller power consumption, the cooling capacity, and the heat exchange of the condenser are changed to zero by a switch in the chiller model.

3.3.4. Ventilation-only mode

There is no water loop in the ventilation-only mode. The outside air is linked directly to the coil and the air distribution system.

3.3.5. Heating mode

The heating mode is the same as the single loop mode, except the chiller is changed into a water source heat pump.

The flow chart of the whole model of the rooftop unit is shown in Fig. 4, and the decision making flow chart is shown in Fig. 5. The model needs three kinds of input data: the building load, the thermal property data, and the rooftop configuration parameters. The thermal property data and the configuration data are passed into all five modes of the rooftop system directly. The load demand is first filtered and then passed to the right system level models. For example, the filter transfers a faked load to the heating mode when it is running under the cooling mode and a faked cooling load to the cooling modes when it is running under the heating mode. This ensures numerical stability because the cooling mode models cannot simulate heating conditions. The results from the faked inputs were discarded later. This process is not computationally efficient because of the limitation in SPARK modeling. After the simulation, the results of all five modes are passed into an output class to decide which mode the rooftop is running in, and the output class chooses the right outputs to report.

3.4. Model calibration

To calibrate the CEWC model, the heat and mass transfer coefficients were determined by comparing the performance of the CEWC predicted by the model with the performance measured in the laboratory. Two parameters of the CEWC were calibrated: the U value (C_1 and C_2) between the air flow in the dry passage and the water flow and the K_m value (C_3) between the air flow in the wet passage and the water flow. The U value and the K_m value in the model are based on the empirical formulas in the ASHARE

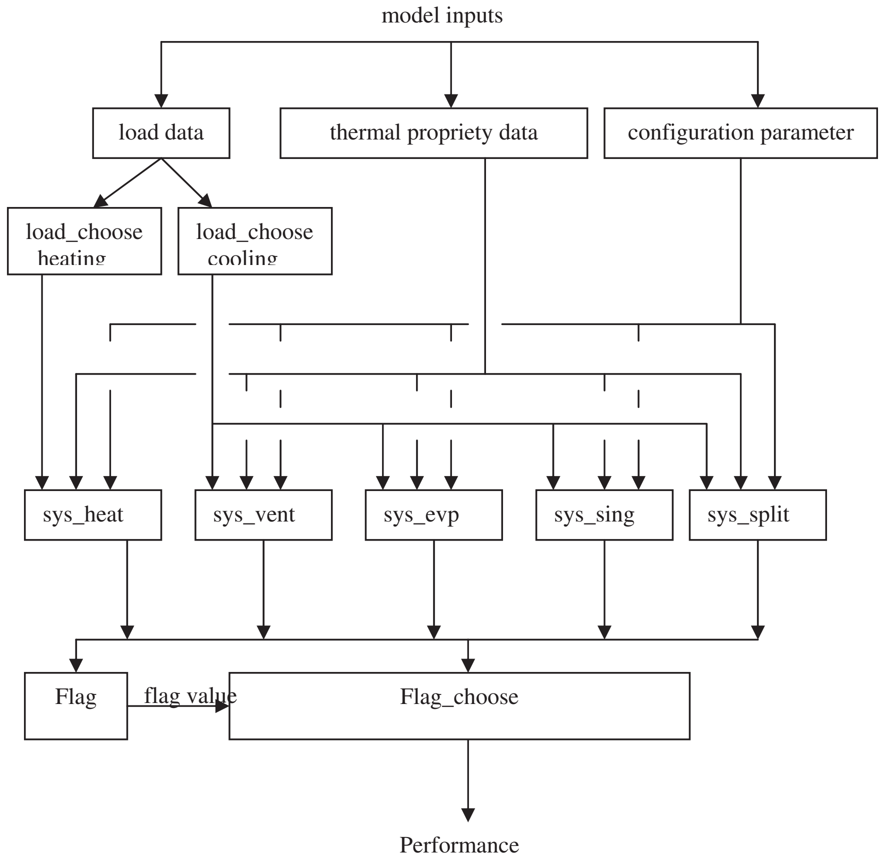

Fig. 4. Model data flow chart.

handbook. These formulas can only be used to predict the real U and K_m within an order of magnitude, and more accurate predictions of these values should be obtained from the actual calibration. The CEWC performance data collected from the bench scale test were used to calibrate the CEWC model. The CEWC was constructed in such a way that its exit air in the dry passage was directed back to the wet passage and made a U-turn at the bottom of the CEWC.

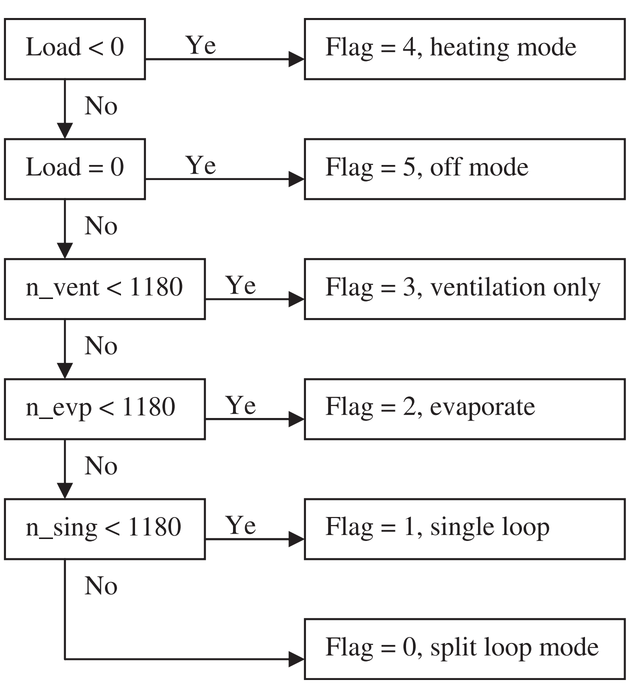

Fig. 5. Model decision making flow chart.

The dataset from the experiment is listed in Table 1. Seven tests were conducted at different times during a single day. The water inlet temperature was adjusted by changing the power of a water heater unit. The water flow through the CEWC varied from 0.51 to 3.8 GPM (0.13–0.98 L/min). The inlet air temperature was relatively steady, and the relative humidity ranged from 16.05% to 21.03%. In the wet passage, the water outlet temperature and air conditions at the bottom of the CEWC (i.e., sump temperature and RH (Relative Humidity)) were measured. In the dry passage, a more detailed air temperature measurement along the vertical passage was conducted. The measurements were taken at 0, 5, and 10 inches away from the centerline of the dry passage and at heights of 0, 2, 7, 11, 15, 21, and 26 inches from the bottom.

Before model calibration, the reliability and accuracy of the performance data were checked. The energy balance of the air stream and the water stream was examined. Considering the CEWC as a closed system, the heat gain from the air stream should be equal to the heat loss of the water stream. For Test 1 (at 15:00:00), the heat gain in the air side was 62% higher than the heat loss from the water side, which indicates the presence of poor measurements during this test. For all tests except the first one, the difference between the heat gains in the air stream and the heat loss of the water stream was within 20%. To avoid the interlinked effects of the two parameters (K_m and U), the two parameters were calibrated separately. Dataset 5 (17:49:00) was used to calibrate the U value because its relatively higher water flow rate meant that the water temperature was stable. After the U value was decided, the K_m value was adjusted until the model agreed with the data as closely as possible.

Table 1
Measured outlet air and water temperatures for the CEWC calibration.

Time		Dry air temperature (°C)	Water temperature (°C)	Wet air temperature (°C)	Wet air relative humidity %
Test 1	Measured	23	21.8	21.7	92%
15:00:00	Modeled	22.7	21.8	23.5	78%
	Relative Deviation	1.30%	0.00%	8.29%	15.22%
Test 2	Measured	27.1	26.7	26.3	93%
16:19:00	Modeled	27.9	26.5	29.9	71%
	Relative Deviation	2.95%	0.75%	13.69%	23.66%
Test 3	Measured	27.4	27.1	26.2	94%
16:46:00	Modeled	28	27.2	29	73%
	Relative Deviation	2.19%	0.37%	10.69%	22.34%
Test 4	Measured	27.4	27.6	26.1	95%
17:00:00	Modeled	28.1	27.6	28.9	74%
	Relative Deviation	2.55%	0.00%	10.73%	22.11%
Test 5	Measured	27.8	28	26.6	96%
17:49:00	Modeled	28.4	27.9	29	75%
	Relative Deviation	2.16%	0.36%	9.02%	21.88%
Test 6	Measured	23.3	26.6	27.2	94%
18:26:00	Modeled	27.6	25.5	31	69%
	Relative Deviation	18.45%	4.14%	13.97%	26.60%
Test 7	Measured	27.4	28.8	28.8	96%
18:54:00	Modeled	29.8	28.5	32	76%
	Relative Deviation	8.76%	1.04%	11.11%	20.83%

After calibration, the C_1 and C_2 parameters in equations (1) and (2) were set to 5 and 2500, respectively. The model-predicted performance and the measured performance are compared in Table 1. Although there were some discrepancies in the wet air exit conditions, the sump water and the air temperatures were very close to each other.

The failure to predict the wet outlet temperature was due to inaccurate measurement and unevenly distributed water flow in the water path. It was very difficult to measure the air condition on the wet side due to the high humidity. The exit wet air condition was very close to the saturation line, and the air enthalpy increased dramatically as the air RH increased. The water temperature and the air temperature on the dry side were considered to be more accurate measurements, and the calibration should weigh more heavily on these data. The model was a one-dimension model. However, in the test data, there was a big temperature difference along the plate at the same height. In test 5, the temperature difference was as high as 4.4 °C.

4. System optimization and analysis

This section presents the results from the simulation model runs and illustrates how the model can be used for size optimization and economic analysis. The model was useful not only in the design phase for selecting and sizing the rooftop unit components but also in the operation phase after the unit is built. In addition, the simulation model predictions are compared with the real measured data obtained from operating the rooftop unit. We conducted four simulation runs to illustrate how to use the model for component sizing and cost analysis. The first run was intended to demonstrate how the rooftop model responds to different outdoor conditions. The second run was used to show how the model can be implemented for full-year running cost analysis. The third run demonstrated how the model can be used for optimal sizing. The fourth run determined how much energy the new unit saves compared to conventional units.

Run 1: different outdoor conditions.

We ran the rooftop computer model against a series of load demand and outside air conditions to evaluate how the model responded. The outside air temperature was changed from −1.1–37.8 °C, while the load demand was changed from −2.9–19.0 kW. The response of the model under each condition is shown in Table 2. The model has many parameters, but the table only shows the most important ones. The unit successfully switched the running mode from heating to ventilation-only, then to evaporative cooling, then to the single loop mode, and finally to the split loop mode automatically when the load demands and the outdoor air temperatures were increasing. The air temperature, the

Table 2
Unit response to different outdoor air temperatures and building thermal loads.

Outdoor and indoor air conditions		System Responses				
Outdoor air temperature (°C)	Building thermal load (kW)	HVAC operation model	CH_kW kW	T_cewc_vent_air (°C)	T_air_CC_in (°C)	T_air_sup_out (°C)
−1.1	−2.9	Heating	0	−1.1	2.9	37.7
10.0	−0.3	Heating	0	10.0	10.0	37.7
15.6	0.0	Off	0	N/A	N/A	N/A
21.1	4.4	Vent only	0	21.1	21.1	21.1
27.8	5.9	Evp	0	19.7	23.9	16.6
32.2	8.8	Evp	0	21.7	25.9	20.5
32.2	13.2	Sing loop	3.02	22.7	25.8	14.5
34.4	14.7	Sing loop	3.06	23.7	26.2	17.1
36.7	16.1	Split loop	3.10	25.8	26.5	11.7
37.8	19.0	Split loop	3.11	26.1	26.6	14.3

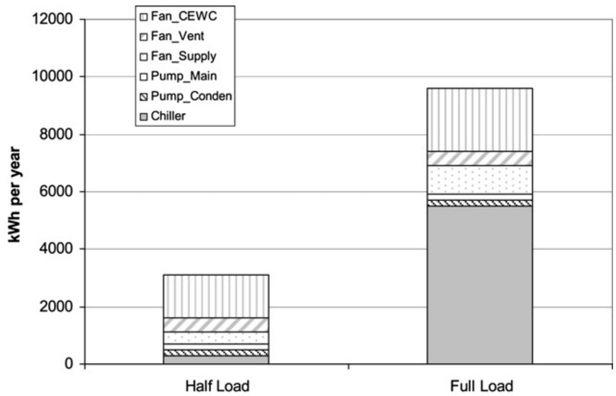

Fig. 6. Rooftop unit annual energy consumption.

leaving temperature of the CEWC, the entering temperature of the water-to-air coil, and the supply air temperature at different points in the unit are listed in the table. The air temperatures changed in each mode to meet the load requirement.

Run 2: energy analysis and unit sizing.

We ran the model against the full-year load profile of a small office building. This small office was a T-24 building about 300 m^2 in size located in Sacramento, CA. The hourly load profile was generated by DOE-2.1. The maximum cooling load was 15 kW. The maximum heating load was 30 kW. The annual electricity consumption was determined by adding the hourly energy consumption for a whole year. Fig. 6 shows the breakdown of the annual electricity usage by the unit, including the electricity use of different components. The biggest electricity consumer was the chiller, which accounted for approximately 50% of the total energy use by the whole system, while the CEWC fans collectively accounted for the remaining half of the energy use. From the initial run, we found that the unit might be undersized for this building because the unit was running under the evaporative cooling mode for a very short time period. To test this hypothesis, we reduced the heating/cooling load at each hour by half and repeated the model run. In this half-load scenario, the unit would operate in the evaporative cooling mode more often than under the full load, even though the outside air temperature remained the same. Fig. 6 also shows the results from the second model run.

Run 3: component sizing.

We ran the model to demonstrate the relationship between the CEWC size and the energy consumptions of the chiller, fans, and pumps. In general, a larger CEWC heat exchange area will reduce the sump air and water temperature and thus reduce the condensing temperature of the chiller and the supply air temperature. A lower condensing temperature reduces chiller power consumption, and a lower air supply temperature reduces the air mass flow rate and, thus, the supply fan power consumption.

Given a certain cooling load14.6 kW and outdoor air temperature 32.2 °C, we ran the model with different CEWC heat exchanger areas, ranging from 30 to 66 m^2. The modeling results are listed in Table 3. Little change was observed in the chiller energy consumption, while the electricity consumption of the supply fan was reduced by half when the simulation used the larger CEWC heat exchanger area. The model simulation can be used to evaluate the sizes of various CEWC components.

Run 4: Energy use comparisons.

We simulated small office, retail and school building types in Sacramento and Irvine, California and in Denver, Colorado. These three cities are in dry climate areas. The evaporative cooling mode was used more frequently than the single loop mode. When a load was satisfied with evaporative cooling, the compressor, which consumed the most power of any system component, was not used. The modeling results are shown in Table 4. The savings from the unit with the CEWC were significant: the unit reduced the compressor electricity consumption by more than 50% in all of these buildings. The magnitude of savings was close to the designed target of a 60% reduction in power consumption.

Table 3
The CEWC size and power consumption.

Temp. outside air °C	Area_CEWC_Wet water m^2	Power_chiller W	Temp. supply air °C	Power_supply fan Bhp
32.2	30	3026	16.7	0.82
32.2	34	3014	16.3	0.70
32.2	38	3004	15.9	0.62
32.2	39	3002	15.8	0.60
32.2	40	3000	15.7	0.58
32.2	41	2998	15.6	0.57
32.2	42	2996	15.6	0.56
32.2	46	2989	15.3	0.51
32.2	50	2983	15.0	0.47
32.2	54	2977	14.8	0.44
32.2	58	2972	14.6	0.42
32.2	62	2968	14.4	0.40
32.2	66	2964	14.3	0.38

Table 4
Energy usage of the newly designed unit compared to a conventional unit.

Location	Building type	Annual compressor electricity (kWh) conventional unit	Annual compressor electricity (kWh) new design	Energy saving rate(%)
Sacramento	Retail	10,529	4479	57.46%
	Small office	9887	3669	62.89%
	School	6437	2653	58.79%
Irvine	Retail	12,871	5798	54.95%
	Small office	11,913	5088	57.29%
	School	8015	3694	53.91%
Denver	Retail	7956	3427	56.93%
	Small office	9469	3649	61.46%
	School	4128	1657	59.86%

5. Conclusions

The objective of this study was to design, develop and simulate a hydronic rooftop package unit that provides a significant improvement in energy efficiency over a conventional rooftop unit. The simulation model built in the study was used to optimize the design, size, and configuration of the rooftop unit. We have found that the new rooftop unit can achieve a 60% reduction in overall annual energy use when compared with a conventional unit. The key advancement of the rooftop unit is the CEWC unit that can cool water down to within 2% of the web bulb air temperature.

A SPARK simulation model was developed. Modeling the CEWC was a challenge because no previous model exists for this design. We adopted an object-oriented modeling approach using SPARK to construct the model, and we divided the CEWC unit into 20 sections and simulated the unit section-by-section. The system model was constructed in a hierarchal way so that each component at the bottom level could be reused in configuring other systems. Through calibration, we found that the simulation model could reasonably simulate the CEWC and the entire rooftop unit.

We recommend that the model be used for component sizing and economic analysis in the design phase, as well as for fault detection and diagnosis during operation. Ideally, if the control system can measure all of the key inputs to the simulation model, a virtual system can run in parallel to the real mechanical system. Comparing the performance of the virtual system and real system will determine whether the system operates as expected. Performance discrepancies can be used for fault diagnosis and intervention at the component model level.

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