Simulation on Counter Flow Heat Exchanger for Indirect Evaporative Cooling Applications

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Abstract— We present simulation result on a evaporative air conditioner which was design based on a counter-flow closed-loop configuration consisting of separated working channels and product channels. The evaporative air conditioner is able to cool air to temperature below room temperature, it means below 27°C.To investigate the performance of the evaporative air cooler under a variety of conditions, the Eulerian-Lagrangian computational fluid dynamics(CFD)model was adopted. The dimension of the air flow passages, and product to working air flow performance were analyzed. Simulation results have indicated that the evaporative air conditioner is able to achieve a higher wet-bulb and dew-point effectiveness with lower air velocity, smaller channel height, larger length to height ratio, and lower product to working air flow ratio.

Key words: Indirect Evaporative Cooling, Heat Exchanger, Heat and Mass Transfer, Numerical Simulation, Air Conditioning

I. INTRODUCTION

Evaporative cooling, a potential alternative to the mechanical vapor compression system, has gained growing attention for use in air conditioning. Evaporative cooling is cost saving and environmentally-friendly method because the working fluids are water and air. Two basic types of the conventional evaporative cooling system for air conditioning are direct evaporative cooling system and indirect evaporative cooling system. In a direct evaporative cooling system, the air has direct contact with water so that the product air cooled and humidified simultaneously due to water evaporation. In an indirect evaporative cooling system, which usually uses some types of heat exchanger, the air can be cooled without absolute humidity change since the product air is kept separate from the evaporation process[1,2].

Compared with the mechanical vapor compression system which currently dominates the air conditioning market, the evaporative cooling system has following advantages [3-7]: (i) extensive saving on energy and cost; (ii) reduction in peak power demand; (iii) no CFCs usages; (iv) reduction in pollution emission; and (v) easy integration with build-up system. However, conventional evaporative cooling has several limitations: (1) increase in air humidity in direct evaporative cooling system resulting in air conditioning that are uncomfortable for humans; (2) low cooling effectiveness of indirect cooling system (around 40-60%) [8]; and (3) temperature of the eventual cooled air is limited by wet-bulb temperature.

In order to overcome these drawbacks of conventional evaporative cooling systems, researchers have investigated possible methods to modify the configuration of the cooler so that the outlet air temperature can be reduced to below its wet-bulb temperature and approaching its dew-point temperature[9,10]. Fig. 1 shows a schematic of a counter flow indirect evaporative cooler. The heat exchanger comprise wet and dry channel. Both the product air and working flow into these channels. The working air is diverted into wet channel, in wet channels, the pre-cooled working air absorb heat from the product air due to water evaporation and is finally exhausted to atmosphere. Concurrently, the product air is cooled along the dry channels by transferring its heat to adjacent wet channels.

Several studies have been conducted on dew-point cooling system. The basic idea of this type of evaporative cooling system is to branch part of the product air, which is pre-cooled, before it is finally delivered [11].

A. Nomenclature

A - area [m^2]
C_p - specific heat of water [J/kg K]
h - heat transfer coefficient [W/m^2 K]
h_m - mass transfer coefficient [m/s]
h_{fg} - latent heat of water evaporation [J/kg]
H - height of channel [m]
k_s - thermal conductivity [W/m K]
L - length of channel [m]
m - mass [kg]
M_p - molecular weight of water
Nu - Nusselt number
P - pressure
Pr - Prandtl number
Re - Reynold number
Sc - Schmidt number
Sh - Sherwood number
t - time [s]
T - temperature
u - velocity
w - humidity ratio
ϵ - efficiency
ρ - air density [kg/m^3]
μ - dynamic viscosity coefficient [Pa s]

Subscript
a - air
e - exhaust
in - inlet
out - outlet
db - dew-point
wb - wet-bulb
sat - saturated

Simulation results indicated that cooling effectiveness of cooler was largely depends on air velocity, dimension of air passage and working to intake air ratio. Performance of the cooler under different inlet air condition and key results have indicated that the range of achievable wet bulb effectiveness 80-90% and the dew-point effectiveness 58-80% [12, 13]. The cooler in the commercial application, with average wet bulb effectiveness 80% was used to pre-cool the fresh air supplied to an existing conventional mechanical vapor compression refrigeration air

conditioning system of building. For the cooler in residential application, the average wet bulb effectiveness was 90%.

It is noteworthy that work done on the evaporative cooling system is still at its infancy stage. To date, most theoretical and experimental work is based on the idea that the working air in wet channel is part of product air which is pre-cooled. Existing studies have yet to adequately study the helpful means of completely separating the working air from the product air or even consider using the room return air as working air.

This work is presenting a formalized and general model to enable the study of new design. The key objectives of this work which differ from existing literature are as follows: (i) we introduce an improved evaporative design to provide air cooling; (ii) developing a dew-point evaporative cooling design that completely separate the working air from the product air.

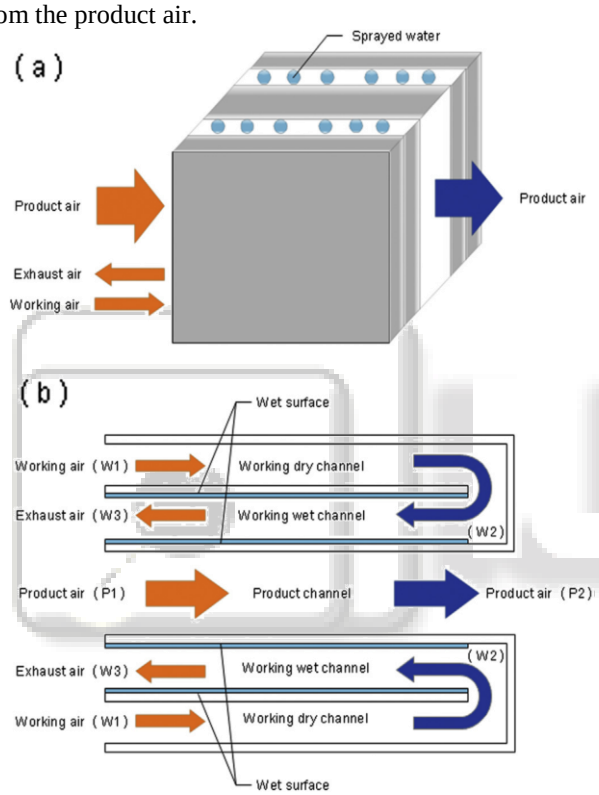

Fig. 1: Schematic of the novel dew point evaporative air cooler. (a) One-unit channel pair (b) plain view.

II. CONCEPT OF A DEW-POINT EVAPORATIVE COOLING

Fig. 1(a) is a schematic of a one-unit channel pair of the novel dew-point evaporative air cooler, and Fig. 1(b) is plain view of the cooler. A one unit channel pair of the cooler consists of a product channel and two adjacent working channels. The cooler comprise a number of these channel pairs that are stacked together. The inner surface of the wet channel is maintained under wet condition by spraying water in wet channel. The working channel has a closed-loop configuration. The air can be cooled by losing sensible heat due to water evaporation in direct or indirect contact with water droplets.

The air cooling process in the evaporative cooler is described as follows. The working air (W1) flows working dry channel. At the end of working dry channel, the working air (W2) is re-circulated into working wet channel and is subsequently exhausted to atmosphere (W3). Therefore, the working air is pre-cooled before entering the working wet channel. The air at the turning point (W2) has low wet bulb temperature and a high cooling potential. The product air (P1) is cooled along the product channel by losing heat to adjacent working wet channels, where heat is absorbed by vaporizing the water, and is finally delivered to the space (P2). This design is capable of cooling the product air without increasing the humidity of the supply air.

In this design, the working air and product the product air are entirely separated. Since the room return air usually has a low wet bulb temperature than ambient air or a fraction of the product air instead of using ambient air or a fraction of the product air[14].

III. COMPUTATIONAL METHOD

We present details related to the computational method in the following sections. The geometry of the proposed evaporative cooler and the appropriate mesh generation are first investigated. It is followed by the systematic layout of the key governing equation, boundary condition and initial conditions. Therefore, the various simulation conditions to demonstrate the performance of evaporative cooler are highlighted.

A. Geometry and Mesh Generation

The pre set dimension of the evaporative cooler are shown in Table 1. The pre set specification for the cooler were determined based on literature [15-20]. The channel length of the evaporative cooler was 1m and channel height was 20mm. Based on recommended range, the channel length, working channel height and product channel height were selected to be 1m, 20mm, 20mm, respectively.

Dimension	Value	Unit
Channel length	1	m
Channel width	0.1	m
Product channel height	0.2	m
Working channel height	0.2	m
Plate thickness	2	m

Table 1: Geometry and Mesh Generation

The influence of dimension of computation domain is illustrated in Fig. 2. The structured mesh was generated in Gambit for the computational domain. In this trail computational case, the dimension of the cooler was under pre set condition and the inlet air was the ambient air of 40°C and humidity ratio of 10 g/kg.

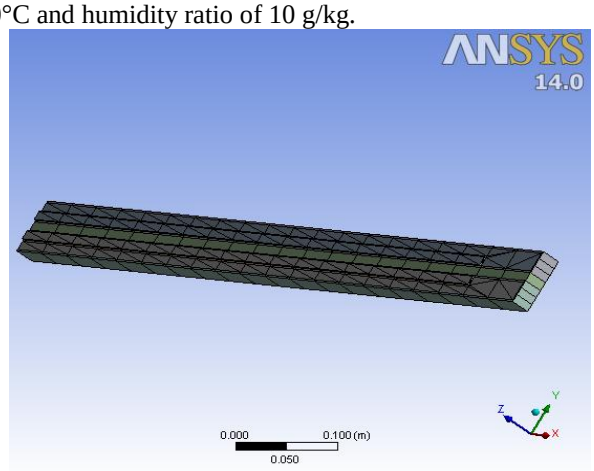

Fig. 2: Schematic of the computational mesh generation of evaporative cooler.

B. CFD Modeling

The heat and mass transfer process in the evaporative cooler was numerically solved using the software ANSYS FLUENT 14.0. The numerical model employed an Eularian-Lagrangian approach. The main air flow was treated as a continuum by solving Navier-Stokes equation (Eularian approach) and the water droplets ware solved as a discrete phase using the Lagrangian approach [21].

In order to simplify model formulation, the following assumption were made: (1) the outer surface was insulated so that no heat was transferred to the surrounding; (2) the channel width was small compared with channel height so that problem could reduce to 2-D; (3) water droplets were evenly distributed on the entire surface of wet channel, and (4) the air flow was steady and incompressible.

1) CFD Equation

- The flux of droplet vapor into the gas phase is related to the gradient of water vapor concentration between water droplet surface and moist air:

$$N_p = h_m\left(\frac{P_{sat}(T_p)}{RT_p} - X_p\frac{P_a}{RT_{db.a}}\right)$$

- Mass of water droplet is determined as:

$$m_p(t+\Delta t) = m_p(t) - N_p A_p M_p \Delta t$$

- Heat transfer process between water droplet and the surrounding air consists of both convective and evaporative heat transfer.

$$m_p c_p \frac{dT_p}{dt} = hA_p(T_{db.a} - T_p) + \frac{dm_p}{dt}h_{fg}$$

2) Coupling between Continuous and Discrete phase

The mass and heat transfer from the discrete phase to the continuous phase is computed by examining the change in mass and thermal energy of water droplets in the control volume. The mass exchange becomes a source of mass in the continuous phase continuity equation and a source of water vapor in the transport equation. The heat transfer to the continuous phase become a source of energy in the continuous phase energy balance in subsequent calculation the continuous phase [21].

3) Boundary Condition

The inlets of both product channel and working channel and working channel were set as velocity inlet defined by:

$$u = u_{in},\ \ v=0,\ \ T = T_{in},\ \ w = w_{in}$$

The outlets of both product channel and working channel were set as pressure outlet defined as

$$\frac{du}{dx} = 0, \frac{dv}{dy} = 0, \frac{dT}{dx} = 0, \frac{dw}{dx} = 0, P=0$$

Since the geometry of a one-unit channel pair could be considered symmetric, the centre line of product channel was set as symmetry boundary condition to reduce the extent of the computational model:

$$\frac{du}{dy} = 0, \frac{dv}{dy} = 0, \frac{dT}{dy} = 0, \frac{dw}{dy} = 0, \frac{dP}{dy} = 0$$

The outer wall was assumed to be adiabatic:

$$u = v = 0, \frac{dT}{dy} = \frac{dw}{dy} = 0$$

4) Simulation Condition

In plane plate cooler, the Reynolds number was generally low so that the air flow was treated as fully developed laminar flow. The k-ε model was adopted. The governing equation was discretized with the finite volume method. The pressure and velocity were coupled using the SIMPLE (semi-implicit method for pressure linked equations) algorithm. The advection term in the momentum, energy and mass transport equation were discretized using the second order upwind.

Discrete phase model (DPM) mass source in the volume monitor was used to record the solution history. The solution was considered to be conserved when the DPM mass source in the fluid domain was constant and the monitors were flat.

The performance of the dew-point evaporative air cooler could be evaluated by the wet bulb and dew point effectiveness, which was mathematically expressed as:

$$\varepsilon_{wb} = \frac{(T_{db.in} - T_{db.out})}{T_{db.in} - T_{wb.in})}$$

$$\varepsilon_{dew} = \frac{(T_{db.in} - T_{db.out})}{(T_{db.in} - T_{dew.in})}$$

IV. MATHEMATICAL MODEL

A. The Mass Conservation Equation

The equation for conservation of mass, or continuity equation, can be written as follows:

$$\frac{\partial\rho}{\partial T} + \nabla.(\rho\vec{\vartheta}) = S_m$$

General form of the mass conservation equation and is valid for incompressible as well as compressible flows. The source S_m is the mass added to the continuous phase from the dispersed second phase (e.g., due to vaporization of liquid droplets) and any user-defined sources. For 2D axisymmetric geometries, the continuity equation is given by

$$\frac{\partial\rho}{\partial T} + \frac{\partial}{\partial x}(\rho x) + \frac{\partial}{\partial r}(\rho\vartheta_r) + \frac{\rho\vartheta_r}{r} = S_m$$

B. Momentum Conservation Equations

Conservation of momentum in an inertial (non-accelerating) reference frame is described

$$\frac{\partial}{\partial T}(\rho\vec{\vartheta}) + .(\rho\vec{\vartheta}\,\vec{\vartheta}) = -\nabla p + \nabla\bar{\tau} + \rho\vec{g} + \vec{F}$$

Where P is the static pressure, $\bar{\tau}$ is the stress tensor (described below), and $\rho\vec{g}$ and the gravitational body force and external body forces (e.g., that arise from interaction with the dispersed phase), respectively. $\vec{F}$ also contains other model-dependent source terms such as porous-media and user-defined sources.

The stress tensor is given by

$$\bar{\tau} = \mu\left[\left(\nabla\vec{\vartheta} + \nabla\vec{\vartheta}^T\right) - \frac{2}{3}\nabla\vec{\vartheta}\vec{I}\right]$$

where μ is the molecular viscosity, I is the unit tensor, and the second term on the right hand side is the effect of volume dilation.

C. Single Phase Flow

For an arbitrary scalar $\emptyset_k$, ANSYS FLUENT solves the equation

$$\frac{\partial\rho\emptyset_k}{\partial t} + \frac{\partial}{\partial x_i}\left(\rho u_i\emptyset_k - \Gamma_k\frac{\partial\emptyset_k}{\partial x_i}\right) = S_{\emptyset_k}$$

where Γ_k and $S_{\emptyset_k}$ are the diffusion coefficient and source term supplied by you for each of the N scalar equations. Note that Γ_k is defined as a tensor in the case of anisotropic diffusivity. The diffusion term is thus $\nabla.(\Gamma_k.\emptyset_k)$.

For the steady-state case, ANSYS FLUENT will solve one of the three following equations, depending on the method used to compute the convective flux:

– If convective flux is not to be computed, ANSYS FLUENT will solve the equation

$$\frac{\partial}{\partial x_i}\left(\Gamma_k \frac{\partial \emptyset_k}{\partial x_i}\right) = S_{\emptyset_k}$$

where Γ_k and $S_{\emptyset_k}$ are the diffusion coefficient and source term supplied by you for each of the N scalar equations.

– If convective flux is to be computed with mass flow rate, ANSYS FLUENT will solve the equation

$$\frac{\partial}{\partial x_i}\left(\rho u_i \emptyset_k - \Gamma_k \frac{\partial \emptyset_k}{\partial x_i}\right) = S_{\emptyset_k}$$

– It is also possible to specify a user-defined function to be used in the computation of convective flux. In this case, the user-defined mass flux is assumed to be of the form

$$F = \int_s \rho \vec{u}\, d\vec{S}$$

Where $d\vec{S}$ is the face vector area.

D. Energy Conservation Equation

Conservation of energy is described by

$$\frac{\partial}{\partial t}(\rho E) + \nabla.\left(\vec{v}(\rho E + P)\right) = -\nabla\left(\sum_j h_j J_j\right) + S_h$$

E. Realizable k- ε Model

The realizable k- ε model differs from the standard k- ε model in two important ways:

The realizable k- ε model contains an alternative formulation for the turbulent viscosity.

A modified transport equation for the dissipation rate ε has been derived from an exact equation for the transport of the mean-square vorticity fluctuation.

F. Transport Equations for the Realizable k- ε Model

The modeled transport equations for k and ε in the realizable k- ε model are

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_m + S_k$$

and

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \sigma}{\partial x_j}\right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\vartheta \varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k} C_{3\varepsilon} G_b S_\varepsilon$$

where

$$C_1 = max\left[0.43, \frac{\eta}{\eta+5}\right], \eta = S\frac{k}{\varepsilon}$$

G. Modeling the Turbulent Viscosity

As in other k- ε models, the eddy viscosity is computed from

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

The difference between the realizable k- ε model and the standard and RNG k- ε models is that C_μ is no longer constant.

V. Results and Discussion

Employing the validated model, we first investigate the performance of the cooler with plain channels. Reynolds number in these studies range from 375 to 1250. Therefore, the air flow in cooler is laminar flow. Wet bulb and dew point effectiveness are employed to evaluate the performance of cooler under varying conditions. Therefore, the outlet air temperature is one of the key results concerned in this study.

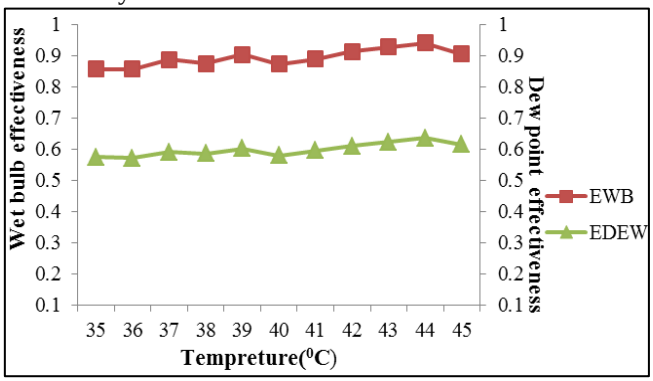

Fig. 3: Wet bulb and Dew point effectiveness for different inlet air conditions

Fig. 3 shows the wet bulb effectiveness and dew point effectiveness of cooler under different inlet air temperature and humidity conditions. In this test the inlet air temperature was varied from 35 to 45°C and humidity ratio was varied from 10 g/kg to 11 g/kg. The inlet velocity was kept as 1 m/s.

Simulation results shows that the cooler is able to cool temperature below room temperature since the wet bulb effectiveness of the cooler is approximately 90 to 95% as shown in Fig. 3 and dew point effectiveness 50 to 65%.

When the humidity ratio of inlet air is low, there is a greater driving force for mass transfer due to large vapor pressure difference between the air and the water interference. As a result the working air has greater capacity to absorb the moisture from water evaporation and the large sensible heat can be transferred from the dry channel to wet channel. Therefore higher wet bulb effectiveness is obtained when the inlet air humidity is low. On the other hand, as the temperature difference between the inlet dry bulb and dew point temperature is much greater at low humidity ratio. Therefore, the dew point effectiveness may be decreased in lower humidity ratio condition.

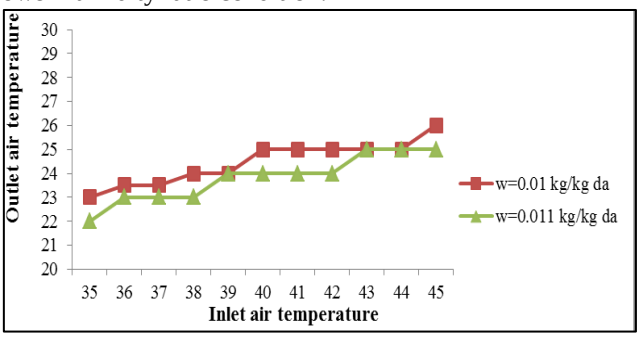

Fig. 4: Outlet air temperature under varying specific humidity condition.

Further illustrates the influence of inlet air temperature and humidity conditions on the outlet air temperature. In general, the simulation results shows that lower outlet air temperature can be achieved by reducing the inlet air humidity ratio and temperatures. This can be attributed to the fact that a lower inlet air humidity ratio provides a greater driving force for mass transfer so that the evaporative heat transfer can be enhanced. In addition lower initial air temperature is obtained by reducing the inlet air temperature.

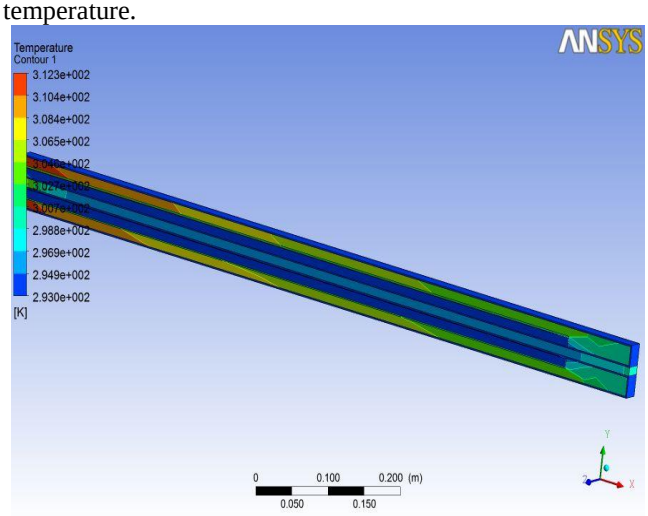

Fig. 5: Temperature profiles of air in dry working channel, wet working channel and product channel.

Temperature profiles of air in the channels are shown in Fig. 5. It can be seen that the air temperature decreases along the air flow direction in both working dry channel and product channel. At a certain point of all channels, the air in working wet channel achieves lowest temperature and therefore the heat transferred from dry channels to adjacent working wet channel due to water evaporation in working wet channel.

VI. CONCLUSION

The evaporative air cooler was able to cool air to the temperature below room temperature and approaching to dew point temperature. Operating under variant inlet conditions (air temperature and humidity ratio), simulation results showed that the cooler could achieve wet bulb effectiveness of up to 98%, and dew point effectiveness of up to 66%. In cooler, the working air and the product air were entirely separated. It was suggested that the length of the channel passage should be at least 50 times of height of working channel.

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