Energy xxx (2010) 1–7

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

A new approach to analysis and optimization of evaporative cooling system I: Theory

Qun Chen [a,b,*], Kangding Yang [a], Moran Wang [c], Ning Pan [b], Zeng-Yuan Guo [a]

[a] Key Laboratory for Thermal Science and Power Engineering of Ministry of Education, Department of Engineering Mechanics, Tsinghua University, Beijing 100084, China
[b] Biological and Agricultural Engineering Department, University of California, Davis, CA 95616, USA
[c] Earth and Environmental Sciences Division, Los Alamos National Laboratory, Los Alamos, NM 87545, USA

ARTICLE INFO

Article history:
Received 25 November 2009
Received in revised form
18 February 2010
Accepted 21 February 2010
Available online xxx

Keywords:
Evaporative cooling
Energy conservation
Moisture entransy and dissipation
Thermal resistance
Temperature potential

ABSTRACT

Using the analogy between heat and mass transfer processes, the recently developed entransy theory is extended in this paper to tackle the coupled heat and mass transfer processes so as to analyze and optimize the performance of evaporative cooling systems. We first introduce a few new concepts including the moisture entransy, moisture entransy dissipation, and the thermal resistance in terms of the moisture entransy dissipation. Thereinafter, the moisture entransy is employed to describe the endothermic ability of a moist air. The moisture entransy dissipation on the other hand is used to measure the loss of the endothermic ability, i.e. the irreversibility, in the coupled heat and mass transfer processes – this total loss is shown to consist of three parts: (1) the sensible heat entransy dissipation, (2) the latent heat entransy dissipation, and (3) the entransy dissipation induced by a *temperature potential*. Finally the new thermal resistance, defined as the moisture entransy dissipation rate divided by the squared refrigerating effect output rate, is recommended as an index to effectively reflect the performance of the evaporative cooling system. In the end, two typical evaporative cooling processes are analyzed to illustrate the applications of the proposed concepts.

1. Introduction

With the rising living standard, air-condition system has become more popular and even a necessity in life to create comfortable environment, and consumed a large amount of energy at the same time. Given the ever increasing urgency in energy conservation [1–3], a more efficient air conditioning technology is obviously highly desirable. Due to its low cost and potential high efficiency, evaporative cooling [4] has been viewed as an attractive option, when compared to other existing ones as vapor compression, absorption/adsorption and thermoelectric refrigeration systems, for both regions with dry and hot climate [5] and more temperate climates [6]. Still in order to improve the performance of the evaporative cooling units, extensive research has been conducted in analyzing the influences of such factors as moist air velocity, temperature and humidity [7,8], water velocity and temperature [9], longitudinal heat conduction [10], heat and mass exchanging materials properties [11], and geometries [12] on the efficiency of various traditional and novel evaporative coolers, including plate/tube type indirect evaporative cooler [7,13], compact-plate cross flow indirect evaporative cooler [10], and semi-indirect/dew-point evaporative cooler [14,15]. Several active or passive evaporative cooling systems have been developed, using wind tower [16], intermittent evaporative roof cooling [17–19], automatic wind-tracking evaporative cooling [20] and two-stage indirect/direct evaporative cooling [21,22]. The general underlining principle in all these is to maximize the refrigerating effect in an evaporative cooling system at an inlet moist air of given endothermic ability, i.e. to minimize the endothermic ability dissipation caused by the "inherent heat and mass transfer resistance" during a coupled heat and mass transfer in the system. However, by carefully examining the physical mechanisms involved, there are still some physical quantities missing both in estimating the endothermic ability of the moist air, and in measuring the endothermic ability dissipation during the process.

Take heat transfer as an example. Considering the need of an effective measure for the heat transfer ability of a system, a physical quantity termed entransy was introduced by Guo et al. [23,24]. The entransy dissipation was then used to reflect the loss of heat transfer ability during the process. Furthermore, they developed the minimum thermal resistance law based on the entransy

* Corresponding author. Key Laboratory for Thermal Science and Power Engineering of Ministry of Education, Department of Engineering Mechanics, Tsinghua University, Beijing 100084, China. Fax: +86 10 62781610.
E-mail address: chenqun@tsinghua.edu.cn (Q. Chen).

0360-5442/$ – see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.energy.2010.02.037

Nomenclature

A	area, m^2
c_p	constant pressure specific heat, $J\,kg^{-1}\,K^{-1}$
D	mass diffusivity, $m^2\,s^{-1}$
G	entransy, JK
$\dot{G}$	entransy flow rate, $J\,K\,s^{-1}$
g_s	sensible heat entransy per unit mass, $J\,K\,kg^{-1}$
g_l	latent heat entransy per unit mass, $J\,K\,kg^{-1}$
g_a	total entransy of moist air per unit mass, $J\,K\,kg^{-1}$
h	specific enthalpy, $J\,kg^{-1}$
$\vec{n}$	outward normal unit vector
P	pressure, Pa
$\dot{Q}$	heat flow rate, W
R_h	thermal resistance, $k\,W^{-1}$
T	temperature, K
t	time, s
U	internal energy, J; velocity, $m\,s^{-1}$
V	volume flow rate, $m^3\,s^{-1}$; Volume, m^3
ρ	density, $kg\,m^{-3}$
λ	thermal conductivity, $W\,m^{-1}\,K^{-1}$
ϕ_h	local entransy dissipation rate, $W\,Km^{-3}$
$\dot{\Phi}_h$	total entransy dissipation rate, WK
ω	mass friction, $kg\,kg^{-1}$
γ	evaporation latent heat, $J\,kg^{-1}$

Subscripts

a	air
c	cold fluid
dp	dew-piont
h	hot fluid
in	inlet
out	outlet
w	water
0	ambient

dissipation for heat transfer optimization. That is, the minimum thermal resistance based on the entransy dissipation leads to the maximum heat transfer efficiency. Based on the entransy theory, three heat transfer mechanisms, i.e. heat conduction [23], convective heat transfer [25–27] and thermal radiation [28], have been analyzed, leading to some heat transfer enhancement apparatuses, e.g. alternating elliptical axis tubes [29] and discrete doubled inclined ribs tubes [30]. Meanwhile, due to the analogy between heat and mass transfer, the entransy theory has recently been extended to mass transfer analysis and optimization by introducing the concepts of mass entransy and mass entransy dissipation [31].

In this paper, we extend the entransy theory to evaporative cooling analysis, treated as a coupled heat and mass transfer problem. The concepts of moisture (moist air) entransy and moisture (moist air) entransy dissipation will be introduced to, correspondingly, represent the endothermic ability of moist air, and measure the irreversibility of evaporative cooling processes due to endothermic ability loss. Based on the *moisture entransy dissipation*, a new expression of thermal resistance is defined to effectively reflect the performance of evaporative cooling systems.

2. Entransy, entransy dissipation and thermal resistance in heat transfer

In a pure heat transfer process, there has been no such concept as the transfer efficiency, or it has been inconceivable to develop an optimization scheme because the thermal energy is always conserved. The system entropy is only useful when mechanical work has been exchanged with the external surroundings. In order to reconcile this predicament, Guo et al. [23] introduced a physical quantity, termed *entransy*, to describe the heat transfer ability of an object based on the analogy between heat conduction and electric conduction. The definition of entransy is:

$$G = \frac{1}{2}UT \tag{1}$$

where temperature, T, is an intensive quantity, and internal energy of the object, U, is an extensive quantity. Because both temperature and internal energy are state functions, so is the entransy. Accompanying with the heat transfer, entransy is also transferred. However, different from the thermal energy conservation, *entransy is not conserved*, but dissipated during a heat transfer process, and this can be readily demonstrated. For a transient heat conduction process without any heat source, the thermal energy conservation equation can be expressed as

$$\rho c_p \frac{\partial T}{\partial t} = \nabla\cdot(\lambda \nabla T) \tag{2}$$

where ρ, c_p, λ are density, constant pressure specific heat and thermal conductivity, respectively. Multiplying both sides by temperature T gives the compositions of the total system entransy for the heat conduction:

$$\rho c_p T\frac{\partial T}{\partial t} = \nabla\cdot(\lambda T\nabla T) - \lambda|\nabla T|^2 \tag{3}$$

The left term in Eq. (3) can be viewed as the time variation of the entransy stored per unit volume which consists of two items shown in the right: the first represents the entransy transferred from one (or part of the) object to the another (part), while the second term is *the local entransy dissipation*, during the heat conduction. This is analogous to electric energy dissipation during an electric conduction process or mechanical energy dissipation during fluid flow process. Since both electric energy and mechanical energy dissipations are irreversibility measure of their respective process, entransy dissipation ϕ_h is hence a measure of the irreversibility in heat transfer

$$\phi_E = \lambda|\nabla T|^2 \tag{4}$$

Likewise for a steady-state *convective heat transfer* with no heat source, the thermal energy conservation equation is in the form

$$\rho c_p U_f\cdot\nabla T = \nabla\cdot(\lambda\nabla T) \tag{5}$$

where U_f is the velocity vector of the fluid. Again multiplying both sides by temperature T, an entransy expression for the convective heat transfer is derived as [23]:

$$U_f\cdot\nabla\left(\frac{\rho c_p T^2}{2}\right) = \nabla\cdot(\lambda T\nabla T) - \lambda|\nabla T|^2 \tag{6}$$

The left term in Eq. (6) is the entransy transferred associated with the fluid particles motion, still made of on the right side the first term describing the entransy diffusion within the fluid due to temperature gradient, and the local entransy dissipation, in the same form as in heat conduction. That is, convective heat transfer is essentially a *heat conduction* process accompanied by fluid flow, and the irreversibility in the convective heat transfer is actually caused by the heat diffusion.

By integrating Eq. (6) over the entire domain, transforming the volume integral to the surface integral on the domain boundary according to Gauss's Law, and ignoring the heat diffusion in the fluid flow direction at both inlets and outlets, we obtained the following equation:

$$\left(\frac{1}{2}\rho\dot{V}c_pT^2\right)_{\text{out}}-\left(\frac{1}{2}\rho\dot{V}c_pT^2\right)_{\text{in}} = \iint_{\Gamma} \vec{n}\cdot\lambda T\nabla T\mathrm{d}A - \iiint_{\Omega} \lambda|\nabla T|^2\mathrm{d}V \tag{7}$$

On the left, the two terms describe the entransy flowing into and out of the domain, respectively. On the right, the first term is the entransy flow rate induced by heat transfer through the domain boundary, while the second term is the *total entransy dissipation rate* (TEDR) during the process.

$$\dot{\Phi}_{\text{h}} = \iiint_{\Omega} \lambda|\nabla T|^2\mathrm{d}V \tag{8}$$

For heat transfer between cold and hot fluids in a heat exchanger, there is no heat transferred to/from the ambient through the boundary, so the first term on the right hand of Eq. (7) vanishes. Then Eq. (7) is reduced into [32]:

$$\begin{aligned}\dot{\Phi}_{\text{h}} &= (\dot{G}_{\text{h,in}}-\dot{G}_{\text{h,out}})+(\dot{G}_{\text{c,in}}-\dot{G}_{\text{c,out}})\\ &= \left[\left(\tfrac{1}{2}\dot{m}_{\text{h}}c_{\text{p,h}}T_{\text{h}}^2\right)_{\text{in}}+\left(\tfrac{1}{2}\dot{m}_{\text{c}}c_{\text{p,c}}T_{\text{c}}^2\right)_{\text{in}}\right]-\left[\left(\tfrac{1}{2}\dot{m}_{\text{h}}c_{\text{p,h}}T_{\text{h}}^2\right)_{\text{out}}\right.\\ &\quad\left.+\left(\frac{1}{2}\dot{m}_{\text{c}}c_{\text{p,c}}T_{\text{c}}^2\right)_{\text{out}}\right]\end{aligned} \tag{9}$$

where $\dot{G}_{\text{i,in}}$, $\dot{G}_{\text{i,out}}$, $\dot{m}_{\text{i}}$, $c_{\text{p,i}}$ and T_{i} are the entransy inflow and outflow, mass flow rate, constant pressure specific heat and temperature of the fluid i, with the subscript i = h (hot) = c (cold), respectively.

Equation (9) shows that the *TEDR* in a heat exchanger equals to the total entransy flowing into the heat exchanger minus that out of the heat exchanger. Furthermore, in electricity the electrical resistance is defined as the electrical dissipation rate divided by the electric current squared. Assuming an analogy between heat and electrical transfers, the thermal resistance associated with the entransy dissipation can be defined, likewise, as the *TEDR* divided by the total heat flow rate squared:

$$R_{\text{h}} = \frac{\dot{\Phi}_{\text{h}}}{\dot{Q}^2} = \frac{(\dot{G}_{\text{h,in}}-\dot{G}_{\text{h,out}})+(\dot{G}_{\text{c,in}}-\dot{G}_{\text{c,out}})}{\dot{Q}^2} \tag{10}$$

Compared to the existing definition of the thermal resistance as the reciprocal of thermal conductance, i.e. the reciprocal of the production of total heat transfer coefficient and surface area, the thermal resistance in Equation (10) based on the entransy dissipation has the following advantages:

1. It is independent of the heat exchanger type or fluid flow direction. Thus, this resistance is only determined by the effectiveness of the heat exchanger and the heat capacity flow ratio, and can be used to analyze and compare the capability between different heat exchangers. Whereas the existing thermal resistance depends on both heat exchanger type and fluid flow direction, thus difficult for comparison between distinctive cases.
2. The new thermal resistance can also be used to analyze heat exchangers with complex configurations, e.g. multiple flow of more than two fluids. Whereas the existing definition does not work well due to the potentially indefinite specific temperature difference and heat flux, and is hence only suitable for analyzing simple heat exchangers, e.g. double-flow types.
3. Most important, the thermal resistance based on entransy dissipation connects the effectiveness of heat exchanger directly to the irreversibility of heat transfer process, and consequently, offers a novel way to optimize the heat exchanger performance.

3. Moisture entransy, moisture entransy dissipation and thermal resistance for evaporative cooling

3.1. Physical model

Consider a coupled heat and mass transfer case between moist air and liquid water as shown in Fig. 1, where domain *acge* represents the moist air region including the main moist air region *abfe* and a thin air layer *bcgf* right on the surface of the liquid water. Both the temperature and partial pressure of the moist air in this thin layer correspond to those of the liquid water in domain *cdhg*. In addition this coupled heat and mass transfer process is assumed at steady-state, where all the physical parameters perpendicular to the fluid flow direction remain unchanged during the process.

Note the physics taking place at the different regions in the system often involves distinctive mechanisms, and thus have to be dealt with separately as shown below.

3.2. Definition of moisture entransy and its dissipation in an evaporative cooling system

3.2.1. In the moist air region

In the moist air region *acge*, the conservation equations of both sensible heat and water vapor of the moist air are:

$$\rho_{\text{a}}U_{\text{a}}\cdot\nabla(c_{\text{p,a}}T_{\text{a}}) = \nabla\cdot(\lambda_{\text{a}}\nabla T_{\text{a}}) \tag{11}$$

$$\rho_{\text{a}}U_{\text{a}}\cdot\nabla\omega_{\text{a}} = \nabla\cdot(\rho_{\text{a}}D_{\text{a}}\nabla\omega_{\text{a}}) \tag{12}$$

where ρ_{a}, $c_{\text{p,a}}$, λ_{a}, U_{a} and T_{a} are the density, constant pressure specific heat, thermal conductivity, velocity vector and temperature of moist air, respectively, while D_{a} is the mass diffusivity of water vapor in air, and ω_{a} is the mass fraction of water vapor in moist air. Due to the usually negligible small mass friction of water vapor compared to that of dry air as the dominant constituent in moist air, the mass fraction of water vapor can be as approximated as the weight ratio of the water vapor to dry air, i.e. ω_{a} also is the absolute humidity of the moist air. Eqs. (11) and (12) indicate the evaporation cooling process is a coupled heat and mass transfer problem. Multiplying both sides in Eq. (12) by the evaporation latent heat of water gives the latent heat conservation equation:

$$\rho_{\text{a}}U_{\text{a}}\cdot\nabla(\gamma\omega_{\text{a}}) = \nabla\cdot(\rho_{\text{a}}D_{\text{a}}\nabla\gamma\omega_{\text{a}}) \tag{13}$$

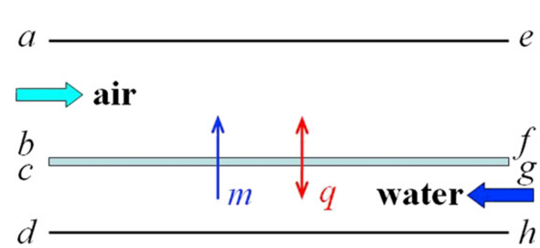

Fig. 1. Schematic of coupled heat and mass transfer during direct evaporative cooling process.

where γ is the evaporation latent heat of water, which for simplicity is assumed unchanged during the evaporative cooling process. Then adding both sides of Eqs. (11) and (13) yields the enthalpy conservation equation during such a direct evaporative cooling process:

$$\rho_a U_a \cdot \nabla\left(c_{p,a}T_a + \gamma w_a\right) = \nabla\cdot(\lambda_a \nabla T_a) + \nabla\cdot(\rho_a D_a \nabla \gamma \omega_a) \tag{14}$$

where $c_{p,a}T_a + \gamma w_a = h_a$, is the enthalpy of the moist air. Integrating Eq. (14) over the flowing domain *acge* of the moist air, transforming the volume integral to the surface integral on the domain boundary according to Gauss's Law, and neglecting the tiny sensible and latent heat diffusion along the moist air flow direction at both inlets and outlets, Equation (14) can be transformed into:

$$\begin{aligned}&\left[\rho_a V_a (c_{p,a}T_a + \gamma w_a)\right]_{eg} - \left[\rho_a V_a (c_{p,a}T_a + \gamma w_a)\right]_{ac} \\ &= \iint_{cg} \vec{n}_{cg}\cdot(\lambda_a \nabla T_a + \rho_a D_a \nabla\gamma\omega_a)\mathrm{d}A\end{aligned} \tag{15}$$

where $\vec{n}_{cg}$ represents the external normal vector at the boundary cg of domain *acge*, while V_a is the mass flow rate of the moist air. Eq. (15) connects the net enthalpy change in the moist air to the total heat flow rate between the moist air and water. Therein, the sensible heat is transferred in the corresponding temperature field driven by the sensible heat itself. On the other hand the temperature difference between moist air and ambient, $(T_a - T_0)$, is an intensive quantity, and the internal energy difference $c_{p,a}(T_a - T_0)$, is an extensive quantity. Furthermore, since the product of an extensive quantity and an intensive quantity results in the energy of that extensive quantity [33–36], Eq. (11) can be rewritten by multiplying both sides by the temperature difference $(T_a - T_0)$ to offer the conservation equation of the "energy" of the sensible heat, i.e. of the sensible heat entransy:

$$\begin{aligned}\rho_a U_a \cdot \nabla\left[\frac{1}{2}c_{p,a}(T_a - T_0)^2\right] &= \nabla\cdot[\lambda_a (T_a - T_0)\nabla(T_a - T_0)] \\ &\quad - \lambda_a |\nabla(T_a - T_0)|^2\end{aligned} \tag{16}$$

where $1/2c_{p,a}(T_a - T_0)^2 = g_s$ is the sensible heat entransy per unit mass with the ambient temperature T_0 as a benchmark. That is, when the temperature T_a of a substance is higher than the ambient temperature, then g_s represents the exothermic ability of this substance to the ambient, and conversely, the endothermic ability of this substance from the ambient. Corresponding to Eq. (6), Eq. (16) equals the sensible heat entransy in the air flow to the sensible heat entransy diffusion within the moist air, minus the sensible heat entransy dissipation during the process.

Unlike the sensible heat transfer, the latent heat, actually existing in the water vapor, is transferred driven by its concentration gradient, and hence induces a concentration field, rather than a temperature field. Thus, the temperature field in which the latent heat is transferred is nonetheless not caused by the latent heat transfer, but by the sensible heat transfer. In this case, the difference between the equivalent latent heat of the moist air and that of the saturated moist air, $\gamma\ (\omega_a - \omega_0)$, is an extensive quantity, and the temperature difference, $(T_a - T_0)$, is an intensive quantity. This is analogous to the gravitational field where mg is an extensive quantity and the height h is an intensive quantity for an object of mass m. Likewise, multiplying both sides of Eq. (13) by the temperature difference $(T_a - T_0)$ results in the expression of the latent heat entransy:

$$\begin{aligned}&\rho_a U_a \cdot (T_a - T_0)\nabla\gamma(\omega_a - \omega_0) \\ &= \nabla\cdot[\rho_a D_a (T_a - T_0)\nabla\gamma(\omega_a - \omega_0)] - \rho_a D_a \nabla(T_a - T_0)\cdot\nabla\gamma(\omega_a - \omega_0)\end{aligned} \tag{17a}$$

Assuming a linear relationship between humidity ω_a and its corresponding dew-point temperature $T_{a,dp}$, Eq. (17a) turns into:

$$\begin{aligned}&\rho_a U_a \cdot \nabla\tfrac{1}{2}\gamma(\omega_a - \omega_0)\left(T_{a,dp} - T_0\right) = \nabla\cdot[\rho_a D_a (T_a - T_0)\nabla\gamma(\omega_a - \omega_0)] \\ &-\rho_a D_a \nabla(T_a - T_0)\cdot\nabla\gamma(\omega_a - \omega_0) - \left(T_a - T_{a,dp}\right)\nabla\cdot[\rho_a D_a \nabla\gamma(\omega_a - \omega_0)]\end{aligned} \tag{17b}$$

where $1/2\gamma(\omega_a - \omega_0)(T_{a,dp} - T_0) = g_l$ is the latent heat entransy per unit mass of the moist air, measuring the endothermic ability of the moist air from the initial to the saturated state at the ambient temperature. Eq. (17b) indicates that the latent heat entransy transferred via the fluid particles motion consists of three parts, i.e. the latent heat entransy diffusion induced by water vapor diffusion, the latent heat entransy "dissipation" (either positive or negative) associated with not only the water vapor concentration field but also the temperature field, and finally the dissipation of endothermic ability of the moist air due to the difference of the dry-bulb temperature of the moist air over its dew-point temperature at the same hygroscopic capacity: this last term is analogous to that, for the same increment in mass, the variation of gravitational potential energy differs when the heights are different. This dry-bulb temperature over the dew-point temperature can be termed as the "*temperature potential*". For instance, even when the absolute humidity of moist air per unit mass changes from state ω_1 to ω_2, its endothermic ability remains zero in an isothermal process at ambient temperature T_0, whereas this endothermic ability equals to $1/2\gamma[\omega_1(T_{1,dp} - 2T_0) - \omega_2(T_{2,dp} - 2T_0)]$ for moisture sorption taken place on the saturation line. Conversely the endothermic ability of moist air increases as the temperature decreases, provided that the hygroscopic capacity remains the same. In other words, the performance of an evaporative cooling system can be improved by reducing the temperature potential of the moist air as much as possible so as to enhance the endothermic ability from the humidity difference. Adding both sides of equations (16) and (17b), respectively, yields:

$$\begin{aligned}&\rho_a U_a \cdot \nabla\tfrac{1}{2}\left[c_{p,a}(T_a - T_0)^2 + \gamma(\omega_a - \omega_0)\left(T_{a,dp} - T_0\right)\right] \\ &= \nabla\cdot[\lambda_a (T_a - T_0)\nabla(T_a - T_0) + \rho_a D_a (T_a - T_0)\nabla\gamma(\omega_a - \omega_0)] \\ &\quad - \left[\lambda_a |\nabla(T_a - T_0)|^2 + \rho_a D_a \nabla(T_a - T_0)\cdot\nabla\gamma(\omega_a - \omega_0)\right. \\ &\quad \left. + \left(T_a - T_{a,dp}\right)\nabla\cdot\rho_a D_a \nabla\gamma(\omega_a - \omega_0)\right]\end{aligned} \tag{18}$$

where $1/2[c_{p,a}(T_a - T_0)^2 + \gamma(\omega_a - \omega_0)(T_{a,dp} - T_0)] = g_a$ is the moisture entransy per unit mass, or the endothermic ability of the moist air. Integrating Eq. (18) over the moist air region *acge* and neglecting mass diffusion in the flow direction at inlets and outlets gives:

$$\begin{aligned}&\left\{\tfrac{1}{2}\rho_a V_a [c_{p,a}(T_a - T_0)^2 + \gamma(\omega_a - \omega_0)(T_{a,dp} - T_0)]\right\}_{eg} \\ &- \left\{\tfrac{1}{2}\rho_a V_a [c_{p,a}(T_a - T_0)^2 + \gamma(\omega_a - \omega_0)(T_{a,dp} - T_0)]\right\}_{ac} \\ &= \iint_{cg} \vec{n}_{cg}\cdot[\lambda_a (T_a - T_0)\nabla(T_a - T_0) + \rho_a D_a (T_a - T_0)\nabla\gamma(\omega_a - \omega_0)]\mathrm{d}A \\ &\quad - \iiint_{acge}\left[\lambda_a |\nabla(T_a - T_0)|^2 + \rho_a D_a \nabla(T_a - T_0)\cdot\nabla\gamma(\omega_a - \omega_0)\right. \\ &\quad \left. + \left(T_a - T_{a,dp}\right)\nabla\cdot\rho_a D_a \nabla\gamma(\omega_a - \omega_0)\right]\mathrm{d}V\end{aligned} \tag{19}$$

That is, the net entransy change in the moist air in the evaporative cooling domain equals to the total moisture entransy transfer at the air-water interface, and the total moisture entransy "dissipation", during the coupled heat and mass transfer process.

3.2.2. In the liquid water region

In domain *cdhg*, the thermal energy and entransy conservation equations of liquid water are:

Please cite this article in press as: Chen Q, et al., A new approach to analysis and optimization of evaporative cooling system I: Theory, Energy (2010), doi:10.1016/j.energy.2010.02.037

$$\rho_w U_w \cdot \nabla (c_{p,w} T_w) = \nabla \cdot (\lambda_w \nabla T_w) \tag{20}$$

$$\rho_w U_w \cdot \nabla \left[\frac{1}{2} c_{p,w} (T_w - T_0)^2\right] = \nabla \cdot [\lambda_w (T_w - T_0) \nabla (T_w - T_0)] - \lambda_w |\nabla (T_w - T_0)|^2 \tag{21}$$

where ρ_w, $c_{p,w}$, λ_w, U_w are the density, constant pressure specific heat, thermal conductivity, velocity vector and temperature of moist air, respectively. Integrating Eqs. (20) and (21) over the flow domain *cdhg*, respectively, and using the Gauss' Law yields:

$$(\rho_w V_w c_{p,w} T_E)_{gh} - (\rho_w V_w c_{p,w} T_w)_{cd} = \iint_{cg} \vec{n}'_{cg} \cdot \lambda_w \nabla T_w \mathrm{d}A \tag{22}$$

$$\left[\tfrac{1}{2}\rho_w V_w c_{p,w} (T_w - T_0)^2\right]_{gh} - \left[\tfrac{1}{2}\rho_w V_w c_{p,w} (T_w - T_0)^2\right]_{cd} = \iint_{cg} \vec{n}'_{cg} \cdot [\lambda_w (T_w - T_0) \nabla (T_w - T_0)] \mathrm{d}A - \iiint_{cdhg} \left(\lambda_w |\nabla (T_w - T_0)|^2\right) \mathrm{d}V \tag{23}$$

where $\vec{n}'_{cg}$ denotes the external normal direction of the boundary cg of domain *cdhg*, and V_w is the volume flow rate of the liquid water.

At the air–water interface cg between moist air region *acge* and liquid water region *cdhg*, the heat, i.e. sensible plus latent, transferred into moist air equals to the total heat out from the liquid water:

$$-\iint_{cg} \vec{n}_{cg} \cdot (\lambda_a \nabla T_a + \rho_a D_a \nabla \gamma \omega_a) \mathrm{d}A = \iint_{cg} \vec{n}'_{cg} \cdot \lambda_w \nabla T_w \mathrm{d}A \tag{24}$$

Combining Eqs. (14), (22) and (24) results in the total heat conservation equation for the direct evaporative cooling process:

$$(\rho_a V_a h_a)_{ac} - (\rho_a V_a h_a)_{eg} = (\rho_w V_w c_{p,w} T_w)_{cd} - (\rho_w V_w c_{p,w} T_w)_{gh} \tag{25}$$

It shows that the enthalpy increment in the moist air equals to the internal energy decrement in the liquid water. As the temperature at the air-water interface is continuous, i.e. $(T_a)_{cg} = (T_w)_{cg}$, the total entransy absorbed by moist air is equal to that released by liquid water.

$$-\iint_{cg} \vec{n}_{cg} \cdot [\lambda_a (T_a - T_0) \nabla (T_a - T_0) + \rho_a D_a (T_a - T_0) \nabla \gamma (\omega_a - \omega_0)] \mathrm{d}A = \iint_{cg} \vec{n}'_{cg} \cdot [\lambda_w (T_w - T_0) \nabla (T_w - T_0)] \mathrm{d}A \tag{26}$$

Then combining Eqs. (19), (23) and (26) leads to the total entransy conservation equation for the direct evaporative cooling process:

$$(\dot{G}_{a,in} - \dot{G}_{a,out}) + (\dot{G}_{w,in} - \dot{G}_{w,out}) = \dot{\Phi}_h = \iiint_{cdhg} \left(\lambda_w |\nabla (T_w - T_0)|^2\right) \mathrm{d}V + \iiint_{acge} \begin{bmatrix} \lambda_a |\nabla (T_a - T_0)|^2 + \rho_a D_a \nabla (T_a - T_0) \cdot \nabla \gamma (\omega_a - \omega_0) \\ + \rho_a D_a \left(T_a - T_{a,dp}\right) \nabla \cdot \nabla \gamma (\omega_a - \omega_0) \end{bmatrix} \mathrm{d}V \tag{27}$$

where $\dot{G}_{a,in}$ and $\dot{G}_{a,out}$ are the entransy inflow and outflow of moist air, respectively, $\dot{G}_a = 1/2 \rho_a V_a [c_{p,a} (T_a - T_0)^2 + \gamma (\omega_a - \omega_0)(T_{a,dp} - T_0)]$, while $\dot{G}_{w,in}$ and $\dot{G}_{w,out}$ are the entransy inflow and outflow of liquid water, respectively, $\dot{G}_w = 1/2 \rho_w V_w c_{p,w} (T_w - T_0)^2$. Equation (27) shows that the total entransy dissipation for moist air and water in direct evaporative cooling system is equal to the difference between entransy inflow and outflow of the two fluids.

Similar to the definition of thermal resistance based on entransy dissipation for heat exchanger, from the expression of moisture entransy dissipation as shown in Eq. (27), the thermal resistance based on moisture entransy dissipation (TRMED) for the evaporative cooling process is defined as the entransy dissipation rate divided by the total heat flow rate square:

$$R_h = \frac{\dot{\Phi}_h}{\dot{Q}^2} = \frac{(\dot{G}_{a,in} - \dot{G}_{a,out}) + (\dot{G}_{w,in} - \dot{G}_{w,out})}{\dot{Q}^2} \tag{28}$$

4. Analysis of two typical direct evaporative cooling processes

4.1. Direct evaporative cooling processes on an iso-enthalpic line

When the wet-bulb temperature of a moist air inflow is equal to the temperature of water inflow, both the moisture enthalpy and the water temperature will remain unchanged during the whole direct evaporative cooling process, i.e.,

$$(\rho_a V_a h_a)_{ac} - (\rho_a V_a h_a)_{eg} = 0 \tag{29}$$

$$(\rho_w V_w c_{p,w} T_w)_{cd} - (\rho_w V_w c_{p,w} T_w)_{gh} = 0 \tag{30}$$

At the air–water interface cg in Fig. 1, the latent heat flowing into the moist air equals to the sensible heat out of it, and the total heat released by water vanishes.

$$\iint_{cg} \vec{n}_{cg} \cdot (\lambda_a \nabla T_a) \mathrm{d}A + \iint_{cg} \vec{n}_{cg} \cdot (\rho_a D_a \nabla \gamma \omega_a) \mathrm{d}A = 0 \tag{31}$$

$$\iint_{cg} \vec{n}'_{cg} \cdot \lambda_w \nabla T_w \mathrm{d}A = 0 \tag{32}$$

Meanwhile, the temperature field of moist air and the concentration field of water vapor meet the following relationship:

$$\lambda_a \nabla (T_a - T_0) = -\rho_a D_a \gamma \nabla (\omega_a - \omega_0) \tag{33}$$

Therefore,

$$\nabla \cdot [\lambda_a (T_a - T_0) \nabla (T_a - T_0)] + \nabla \cdot [\rho_a D_a (T_a - T_0) \nabla \gamma (\omega_a - \omega_0)] = 0 \tag{34}$$

$$\lambda_a |\nabla (T_a - T_0)|^2 + \rho_a D_a \gamma \nabla (T_a - T_0) \cdot \nabla (\omega_a - \omega_0) = 0 \tag{35}$$

Substituting Eqs. (30), (32)–(35) into Eq. (27) gives:

$$\iiint_{cdhg} \left(\lambda_w |\nabla (T_w - T_0)|^2\right) \mathrm{d}V = 0 \tag{36a}$$

$$\iiint_{acge} \left[\lambda_a |\nabla (T_a - T_0)|^2 + \rho_a D_a \nabla (T_a - T_0) \cdot \nabla \gamma (\omega_a - \omega_0)\right] \mathrm{d}V = 0 \tag{36b}$$

$$\dot{\Phi}_h = \dot{G}_{a,in} - \dot{G}_{a,out} = \iiint_{acge} \rho_a D_a \left(T_a - T_{a,dp}\right) \nabla \cdot \nabla \gamma (\omega_a - \omega_0) \mathrm{d}V \tag{36c}$$

Please cite this article in press as: Chen Q, et al., A new approach to analysis and optimization of evaporative cooling system I: Theory, Energy (2010), doi:10.1016/j.energy.2010.02.037

The equations show that when the wet-bulb temperature of moist air inflow equals to the temperature of the water inflow, the total entransy dissipation for both sensible and latent heat transfer is equal to zero during the direct evaporative cooling process in Eqs. (36a) and (36b), and the entransy is dissipated only when the temperature potential is positive in Eq. (36c). It means that regardless of the area of heat and mass transfer surface, there exists entransy dissipation as long as there is a positive temperature potential.

4.2. Direct evaporative cooling processes on the saturation line

Once the inlet moist air is saturated, it will maintain the saturated state throughout the whole evaporative cooling process. In this case, assuming that the temperature and the absolute humidity of moist air satisfy the linear relationship:

$$w_{\mathrm{a,sat}} = aT_{\mathrm{a,sat}} + b \tag{37}$$

where a and b are constant variables, substituting Eq. (37) into Eq. (27) will give

$$\begin{aligned}\dot{\Phi}_{\mathrm{h}} &= (\dot{G}_{\mathrm{a,in}} - \dot{G}_{\mathrm{a,out}}) + (\dot{G}_{\mathrm{w,in}} - \dot{G}_{\mathrm{w,out}}) \\ &= \iiint_{cdhg} \left(\lambda_{\mathrm{w}} |\nabla (T_{\mathrm{w}} - T_0)|^2 \right) \mathrm{d}V + \iiint_{acge} \left[(\lambda_{\mathrm{a}} + \rho_{\mathrm{a}} D_{\mathrm{a}} \gamma a) |\nabla (T_{\mathrm{a}} - T_0)|^2 \right] \mathrm{d}V\end{aligned} \tag{38}$$

Eq. (38) shows that for evaporative cooling process on saturation line, the entransy is dissipated only through the coupled heat and mass transfer process with finite temperature and mass differences. That is, decreasing the temperature and mass differences by increasing the heat and mass transfer surface will reduce the entransy dissipation.

5. Conclusions

The concepts of moisture entransy, moisture entransy dissipation and the associated thermal resistance are introduced to study the coupled heat and mass transfer in evaporative cooling process. The moisture entransy, composed of both sensible and latent heat entransy, can be employed to describe the endothermic ability of a moist air. The moisture entransy dissipation on the other hand measures the irreversibility of evaporative cooling process as an endothermic ability loss, and consists of the sensible and latent heat entransy dissipation, and the entransy dissipation driven by the difference of a *temperature potential*. The thermal resistance defined as the total entransy dissipation divided by the refrigerating effect output squared can then be used as an index to evaluate the performance of the evaporative cooling system.

For an iso-enthalpic case where wet-bulb temperature of the moist air equals to the water temperature, the moisture entransy dissipation through both the sensible and latent heat transfer vanishes, and the entransy dissipation is only induced by its temperature potential. Conversely, once the inlet moist air is saturated, i.e. the direct evaporative cooling process is on the saturation line, the entransy is dissipated only by the coupled heat and mass transfer process between a finite temperature and concentration difference, while increasing the heat and mass transfer surface will reduce the entransy dissipation and hence improve the performance of the evaporative cooling process.

Acknowledgements

The present work is supported by the National Key Fundamental R&D Program of China (Grant No. G2007CB206901) and the Postdoctoral Scientific Fund of China (Grant No. 20080440027).

References

[1] Heglund NC, Willems PA, Penta M, Cavagna GA. Energy-saving gait mechanics with head-supported loads. Nature 1995;375(6526):52–4.
[2] Hoffert MI, Caldeira K, Benford G, Criswell DR, Green C, Herzog H, et al. Advanced technology paths to global climate stability: energy for a greenhouse planet. Science 2002;298(5595):981–7.
[3] Thauer RK, Kaster AK, Seedorf H, Buckel W, Hedderich R. Methanogenic archaea: ecologically relevant differences in energy conservation. Nature Reviews Microbiology 2008;6(8):579–91.
[4] Watt JR. Evaporative air conditioning handbook. New York: Chapman & Hall; 1986.
[5] Al-Jamal K. Greenhouse cooling in hot countries. Energy 1994;19 (11):1187–92.
[6] Costelloe B, Finn D. Indirect evaporative cooling potential in air–water systems in temperate climates. Energy and Buildings 2003;35 (6):573–91.
[7] Tulsidasani TR, Sawhney RL, Singh SP, Sodha MS. Recent research on an indirect evaporative cooler (IEC) 1. Optimization of the cop. International Journal of Energy Research 1997;21(12):1099–108.
[8] Muangnoi T, Asvapoositkul W, Wongwises S. Effects of inlet relative humidity and inlet temperature on the performance of counterflow wet cooling tower based on exergy analysis. Energy Conversion and Management 2008;49 (10):2795–800.
[9] Lemouari M, Boumaza M, Kaabi A. Experimental analysis of heat and mass transfer phenomena in a direct contact evaporative cooling tower. Energy Conversion and Management 2009;50(6):1610–7.
[10] Hettiarachchi HDM, Golubovic M, Worek WM. The effect of longitudinal heat conduction in cross flow indirect evaporative air coolers. Applied Thermal Engineering 2007;27(11–12):1841–8.
[11] Zhao X, Liu SL, Riffat SB. Comparative study of heat and mass exchanging materials for indirect evaporative cooling systems. Building and Environment 2008;43(11):1902–11.
[12] Sureshkumar R, Kale SR, Dhar PL. Heat and mass transfer processes between a water spray and ambient air – I. Experimental data. Applied Thermal Engineering 2008;28(5–6):349–60.
[13] Stoitchkov NJ, Dimitrov GI. Effectiveness of crossflow plate heat exchanger for indirect evaporative cooling. International Journal of Refrigeration-Revue Internationale Du Froid 1998;21(6):463–71.
[14] Martin RH. Characterization of a semi-indirect evaporative cooler. Applied Thermal Engineering 2009;29(10):2113–7.
[15] Zhao XD, Yang S, Duan ZY, Riffat SB. Feasibility study of a novel dew point air conditioning system for China building application. Building and Environment 2009;44(9):1990–9.
[16] Saffari H, Hosseinnia SM. Two-phase Euler-Lagrange CFD simulation of evaporative cooling in a wind tower. Energy and Buildings 2009;41 (9):991–1000.
[17] Alturki AM, Zaki GM. Energy saving through intermittent evaporative roof cooling. Energy and Buildings 1991;17(1):35–42.
[18] Wang XC, Niu JL, van Paassen AHC. Raising evaporative cooling potentials using combined cooled ceiling and MPCM slurry storage. Energy and Buildings 2008;40(9):1691–8.
[19] Kruger EL, Pearlmutter D. The effect of urban evaporation on building energy demand in an arid environment. Energy and Buildings 2008;40 (11):2090–8.
[20] Qiu GQ, Riffat SB. Novel design and modelling of an evaporative cooling system for buildings. International Journal of Energy Research 2006;30 (12):985–99.
[21] Jain D. Development and testing of two-stage evaporative cooler. Building and Environment 2007;42(7):2549–54.
[22] Al-Juwayhel F, El-Dessouky H, Ettouney H, Al-Qattan M. Experimental evaluation of one, two, and three stage evaporative cooling systems. Heat Transfer Engineering 2004;25(6):72–86.
[23] Guo ZY, Zhu HY, Liang XG. Entransy – a physical quantity describing heat transfer ability. International Journal of Heat and Mass Transfer 2007;50 (13–14):2545–56.
[24] Guo ZY, Cheng XG, Xia ZZ. Least dissipation principle of heat transport potential capacity and its application in heat conduction optimization. Chinese Science Bulletin 2003;48(4):406–10.
[25] Chen Q, Ren J, Meng JA. Field synergy equation for turbulent heat transfer and its application. International Journal of Heat and Mass Transfer 2007;50 (25–26):5334–9.
[26] Chen Q, Ren JX. Generalized thermal resistance for convective heat transfer and its relation to entransy dissipation. Chinese Science Bulletin 2008;53 (23):3753–61.
[27] Chen Q, Wang M, Pan N, Guo ZY. Optimization principles for convective heat transfer. Energy 2009;34(9):1199–206.
[28] Wu J, Liang XG. Application of entransy dissipation extremum principle in radiative heat transfer optimization. Science in China Series E-Technological Sciences 2008;51(8):1306–14.
[29] Meng JA, Liang XG, Chen ZJ, Li ZX. Experimental study on convective heat transfer in alternating elliptical axis tubes. Experimental Thermal and Fluid Science 2005;29(4):457–65.

Please cite this article in press as: Chen Q, et al., A new approach to analysis and optimization of evaporative cooling system I: Theory, Energy (2010), doi:10.1016/j.energy.2010.02.037

[30] Li XW, Meng JA, Guo ZY. Turbulent flow and heat transfer in discrete double inclined ribs tube. International Journal of Heat and Mass Transfer 2009;52(3–4):962–70.
[31] Chen Q, Ren JX, Guo ZY. The extremum principle of mass entransy dissipation and its application to decontamination ventilation designs in space station cabins. Chinese Science Bulletin 2009;54(16):2862–70.
[32] Liu XB, Meng J, Guo ZY. Entropy generation extremum and entransy dissipation extremum for heat exchanger optimization. Chinese Science Bulletin 2009;54(6):943–7.
[33] Onsager L. Reciprocal relations in irreversible processes. II. Physical Review 1931;38(12):2265–79.
[34] Onsager L. Reciprocal relations in irreversible processes. I. Physical Review 1931;37(4):405–26.
[35] Onsager L, Machlup S. Fluctuations and irreversible processes. Physical Review 1953;91(6):1505–12.
[36] Prigogine I. Introduction to thermodynamics of irreversible processes. 3rd ed. New York: Wiley; 1967.