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Analysis of Evaporative Cooling System and Recovery of Potable Water as by Product

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Abstract. Evaporative cooling is the most applied cooling technology in the world. In this paper a water recovery (by condensation) system by radiative cooling of humid-air (as output of evaporative cooling systems) is designed. The radiative cooling method is a method which utilizes the cold sky, which has low effective temperature, as a heat sink. A mathematical model is proposed to describe the thermal performance of the condenser. The results of the mathematical model predict condensation rates within the range 3 to 6-7 liter/m^2-night. Also, the effects of different design parameters on the proposed condenser performance are investigated. The water thus obtained is pure distilled water and can be used as potable water, thus the installation of such water condensation system with evaporative cooling applications may prove a quality improving strategy for evaporative cooling systems. It will be beneficial for the human beings by two ways, first for thermal comfort and second getting potable water as by product of the cooling system.

Keywords: Evaporative cooling, Distillation, condensation, potable water.

1. Introduction

Evaporative cooling operates using induced processes of heat and mass transfer, where water and air are the working fluids. It consists, specifically, water evaporation, induced by the passage of an air flow, thus decreasing the air temperature. In evaporative cooling systems an air stream with high humidity ratio is released in the ambient without any treatment. Evaporative cooling are mainly of two types namely direct and indirect evaporative cooling systems as shown in fig.1. In this paper an approach to recover water from such humid air is presented. In practical the sky has low effective temperature and coldness of sky can be used as a heat sink for the condensation of water vapor from humid air, which process is known as radiative cooling. At night the radiation losses from a surface exposed to the atmosphere, at lower temperature, can be used to lower the temperature of the surface and then the cold surface is used to condensate water vapor in the humid air. If the radiation losses from the surface exceed the absorbed radiation then only radiative cooling of a given surface may be achieved. For enhancing the radiation losses to the sky very good black body emitters are used, under clean and clear weather conditions. This improves the atmospheric transmissivity to long wavelength (infrared) radiation. On the other hand, for decreasing the absorbed radiation the cooling system should operate with no imposed heat flux as well as no convective heat gain from the ambient. It is possible to reduce convective heat gains by both low ambient temperature and low wind speed. Also, construction of a stationary air gap or an evacuated gap over the black body emitter, heat gain by the surface can be reduced. This may be achieved by using a polyethylene film cover in place of the traditional glass cover. As polyethylene has the property of very high transmissivity for long wavelength radiation and very low transmissivity for short wavelength radiation which has the major contribution to the incident solar radiation. As a result, polyethylene film reduces the thermal convective gain from the ambient and does not prevent the long wave thermal radiation losses to the sky. Also, the diffuse and direct short wavelength radiation emitted from the sun during daytime or reflected from the moon during night is prevented by polyethylene film. If the upper surface of the polyethylene cover is coated with a highly reflective material it improves the performance of the system as it reduces the heat gain. In the literature, it has been studied that the relative humidity of exhaust air in case of indirect evaporative cooling is quite high

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(more than 90% in certain conditions)[1] and for direct evaporative cooling system also the relative humidity of supply as well as outlet air from conditioned space is very high, which is also a drawback of direct evaporative cooling system. So it seems beneficial to recover water from such high humid air streams. To study the phenomenon of radiative cooling and the sky temperature the considerable research has been carried out [2-11]. Many of these works are done to predict the effective sky temperature [2-5]. Others [6-11] present experimental or theoretical investigations of the radiative cooling systems. However, and to the authors knowledge, most of the available work in the literature utilizes the radiative cooling method in lowering the temperature of a fluid but not for recovery of water, as condensate, from the output air of evaporative cooling systems. The aim of the present study is to investigate the capability of a radiative cooling system in condensing the water vapor from the humid air which comes out from evaporative cooling systems and the humidity ratio of such air is very high. The design of condenser is proposed and analyzed to distil pure water from the water-vapor present in the humid air. A mathematical model is proposed to describe the thermal performance of the condenser system. This model is solved analytically, and as a result, the condensation mass flow rate is predicted. The effects of different design parameters on the condenser performance are analyzed. To operate the radiative cooling systems in daylight the diffusive or direct solar radiation should prevented from entering the cooling system. To achieve this goal, the system should install in the shadow and a cover of a polyethylene film should be used effectively. To incorporate such water recovery system with evaporative cooling systems will be beneficial for human beings in both the respect, i.e. thermal comfort as well as water recovery.

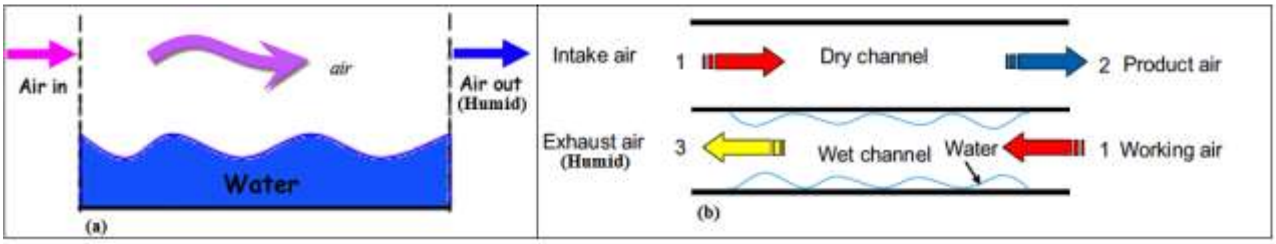


Fig. 1: Schematic element of (a)direct evaporative and (b)indirect evaporative cooling

2. Analysis

Referring to Fig. 2, the main functional part of the radiative condenser consists of black body emitters which might be vertical plates, vertical cylinders or spheres made of highly conductive materials (e.g. copper) and well painted so that the surface behaves as a black surface. These emitters are contained within a space which is covered with a polyethylene film in place of traditional glass cover as stated above. Humid air is injected to enter the gap between the transparent film and the surface of the emitter. The humidity of the injected air may be increased by forcing the air to penetrate through a wet porous material which is wetted from a source of impure water, which needed to be distilled. This is done to increase the humidity of air and to recover water from waste water. Due to the exchange of thermal radiation with the cold effective sky temperature, and prevention of direct solar radiation to enter, the blackbody emitters attain temperature lower than the water vapour saturation temperature. As a result, water vapor contained in the humid air condensates.The thermal energy balance on the emitter can be written as:

$$mc\frac{\partial T_s}{\partial t} = -\varepsilon\sigma A\left(T_s^4 - T_{sky}^4\right) - UA(T_s - T_\infty) + \dot{m}h'_{fg} \tag{1}$$

Where both T_{sky} and T_∞ may be time dependent, and this is the reason why it is introduced the dynamic effect into equation (1). Equation (1) is solved for T_s with the notation that the condensation mass flow rate $\dot{m}$ is a function of T_s as will be seen later. In terms of the obtained emitter surface temperature, the condensation mass flow rate is given as:

$$\dot{m} = \frac{h_L A(T_{sat} - T_s)}{h'_{fg}} \tag{2}$$

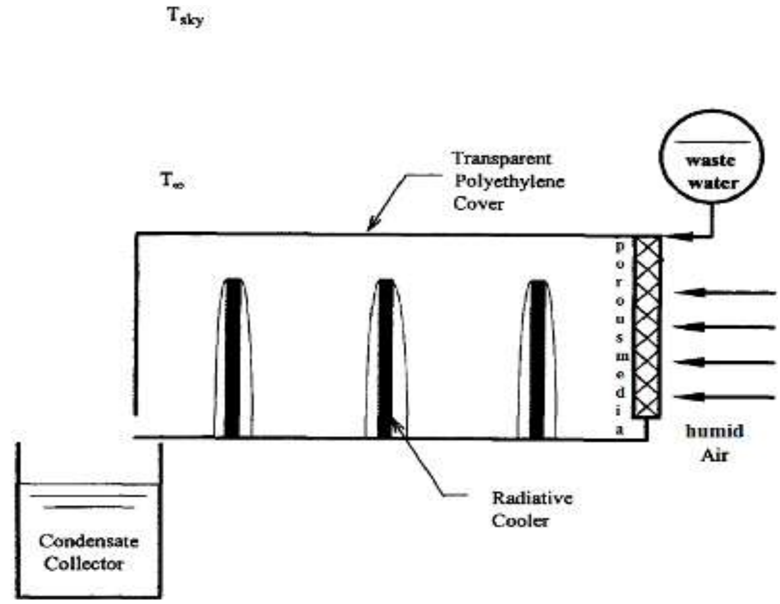


Fig. 2: schematic diagram

Although it is desirable to achieve dropwise condensation in practical applications, it is often difficult to maintain this condition. For this reason and because the convection coefficients for film condensation are smaller than those for dropwise case, condenser design calculations are often based on the assumption of film condensation. The average laminar convection coefficient on a vertical plate is given as [12, 13]:

$$h_L = 0.943\left[\frac{g\rho_1(p_1-\rho_v)k_1^3 h'_{fg}}{\mu_l(T_{sat}-T_s)L}\right]^{1/4} \tag{3}$$

where h'_{fg} is the modified latent heat which is modified to include the thermal advection effect as:

$$h'_{fg}=h_{fg}+0.68c_{p,l}(T_{sat}-T_s) \tag{4}$$

In using this equation all liquid properties should be evaluated at the condensation film temperature $T_f = [(T_{sat}+T_s)/2]$ and h_{fg} should be evaluated at T_{sat}. In terms of equation (2) and (3), equation (1) is rewritten as:

$$mc\frac{\partial T_s}{\partial T} = -\varepsilon\sigma A(T_s^4 - T_{sky}^4)\text{-UA}(T_s - T_\infty)+0.943\left[\frac{g\rho_1(p_1-\rho_v)k_1^3 h'_{fg}}{\mu_l(T_{sat}-T_s)L}\right]^{1/4}\text{A}(T_{sat}-T_s) \tag{5}$$

Equation (4) has to be solved for T_s numerically. However, an approximate analytical solution for T_s may be achieved if equation (5), under steady state conditions, is approximated as:

$$T_s^4+N_1T_s - N_2 \tag{6}$$

Where $N_1=\alpha+\frac{U}{\varepsilon\sigma}$, $N_2=T_{sky}^4+\frac{UT_\infty}{\varepsilon\sigma}+\alpha T_{sat}$, $\alpha = 0.943\left[\frac{g\rho_1(p_1-\rho_v)k_1^3 h'_{fg}}{\mu_l(T_{sat}-T_s)L}\right]^{1/4}$

The last term in equation (5) is approximated to be proportional to $(T_{sat}-T_s)$ instead of $[(T_{sat}-T_s)]^{3/4}$. Equation (6) assumes four different roots for T_s. These roots are given as:

$$T_{s,1}=\frac{F_1}{2}-\text{-}\frac{1}{2}\sqrt{F_1^2-2(y_1+F_2)}$$
$$T_{s,2}=\frac{F_1}{2}+\frac{1}{2}\sqrt{F_1^2-2(y_1+F_2)}$$
$$T_{s,3}=-\frac{F_1}{2}-\frac{1}{2}\sqrt{F_1^2-2(y_1+F_2)}$$
$$T_{s,4}=-\frac{F_1}{2}+\frac{1}{2}\sqrt{F_1^2-2(y_1+F_2)} \tag{7}$$

Where $F_1=\sqrt{y_1}$, $F_2=\sqrt{y_1^2+4N_2}$

and y_1 is the real root of the following cubic equation:

$y^3 + 4N_2$y-N_1^2 (8)

It is clear that the discriminant of equation (8), which is defined as D =$(4N_2/3)^3$+$N_1^2/4$, is positive. As a result, equation (8) assumes one real positive root which is given as:

$$y_1=[\ 0.5N_1^2 + \sqrt{\frac{64}{27}N_2^3+\frac{1}{4}N_1^4}]^{1/3} + [-\frac{1}{2}N_1^2 + \sqrt{\frac{64}{27}N_2^3+\frac{1}{4}N_1^4}]^{1/3} \quad (9)$$

Since both F_1 and F_2 are positive, and $F_1 \leq F_2$ it is clear that the first two roots of equation (6) are complex and the third one is a negative real root. As a result, the only acceptable root is the forth one which gives:

$$T_s= - \frac{F_1}{2} + \frac{1}{2}\sqrt{F_1^2 - 2(y_1 + F_2)} \quad (10)$$

Other empirical formula must be used for h_L if the condensation does not form a laminar film. As an example, for laminar wavy condensation, Kutateladze [14] recommends a correlation of the form:

$$h_L \frac{\left(\frac{v_l^2}{g}\right)^{1/3}}{k_1} = \frac{Re_\delta}{1.08Re_\delta^{1.22}-5.2}\ 30 \leq Re_\delta \leq 1800 \quad (11)$$

and for turbulent condensation, Labuntsov [15] recommends:

$$h_L \frac{\left(\frac{v_1^2}{g}\right)^{1/3}}{k_1} = \frac{Re_\delta}{1.08 + 58Pr^{-0.5}(R_\delta^{0.75}-253)}\ Re_\delta \leq 1800 \quad (12)$$

Where

$$Re_\delta = \frac{4\dot{m}}{b\mu_1}$$

and b is the liquid film width which is equal to the plate width. For dropwise condensation,

h_L is given for steam condensation on well-promoted copper surfaces as [16] :

h_L=255,510(W/m^2k) 00°C<T_{sat}

h_L=51,104+2044(W/m^2k) 22°C<T_{sat}<100°C (13)

The expressions listed above for h_L may also be used for condensation on the outer surface of a vertical cylinder of radius R, if R >> δ. Also, although all the preceding equations strictly apply only to saturated vapours, they can also be used with reasonable accuracy for condensation of superheated vapours [13].

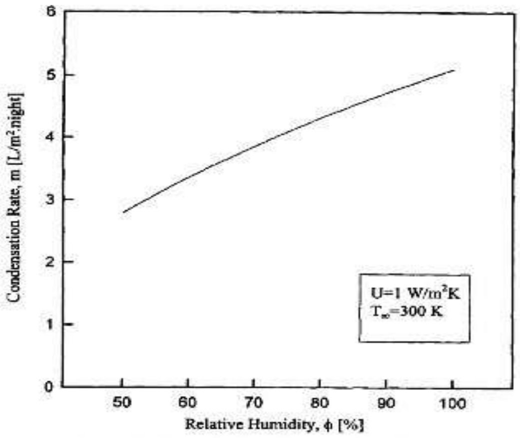
The liquid properties appear in equation (4), and $h_{fg}$are weak functions of temperature especially in the temperature range of interest. The saturation vapour temperature T_{sat} is the water vapour saturation temperature corresponding to the vapour pressure $P_v(= \emptyset P_g)$where ∅ is the relative humidity of the humid air and P_g is the saturation pressure of the water vapour at T_{atm}. A closed form expression which gives $P_g$in terms of $T_{atm}$and T_{atm} in terms of P_v is given as [17] :

$$lnP_r = 5.92714 - \frac{6.09648}{T_r} - 1.28862lnT_r + 0.16934T_r^6 + \omega\left(15.2518 - \frac{15.6875}{T_r} - 13.4721lnT_r + 0.43577T_r^6\right) \quad (14)$$

Where $T_r\left[=\frac{T}{T_c}\right]$ and $P_r[=\frac{P}{P_c}]$ are the reduce temperature and pressure, respectively. T_c and P_c are the critical temperature and pressure, respectively, and ω is the Pitzer acentric factor.

3. Results

Figures 3-5 are plots of equation (10) under different operating conditions. Figure 3 shows the effect of the relative humidity ∅ on the condensation rate. It is clear from this figure that the condensation rate (in litres/m^2-night) is a strong function of ∅. As an example, doubling the relative humidity approximately doubles the condensation rate.



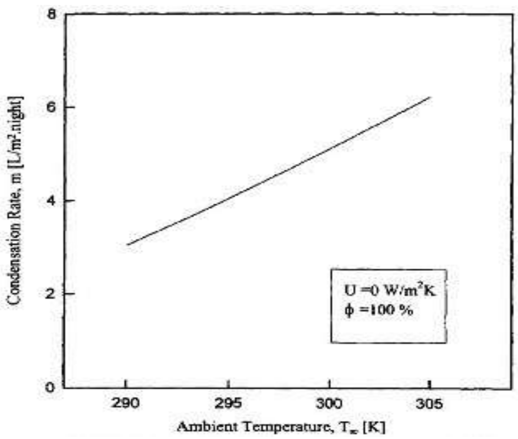


Fig. 3: Effect of relative humidit on condensation rate. Fig. 4: Effect of ambient temperature on condensation rate

Also, Fig. 4 shows that the condensation rate is a strong function of the ambient temperature T_{∞}. As an example, raising the ambient temperature by 15 °may double the condensation rate. On the other hand, and as seen from Fig. 5, it seems that the condensation rate is a weak function of theoverall heat transfer coefficient U.

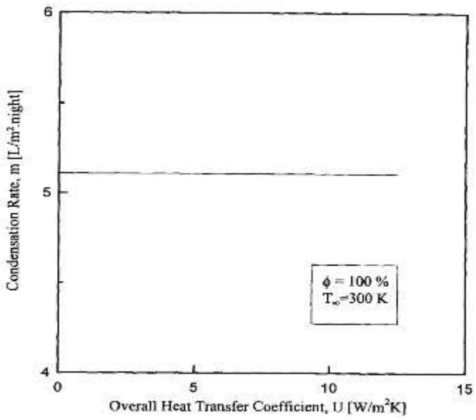


Fig. 5: Effect of heat transfer coefficient on condensation rate

4. Conclusion

A water condensation system by radiative cooling of humid air (as outlet of evaporative cooling systems) is designed to produce pure water. The results of the mathematical model, which is proposed to describe the thermal performance of the system, predict condensation rates within the range 3-7 litre/m^2-night. The effects of different design parameters on the distiller performance are investigated. It is found that the condensation rate is a strong function of the relative humidity and the temperature of the ambient. On the other hand, it seems that the condensation rate is a weak function of the overall heat transfer coefficient of the system and it does not depend on the same. This system can be operated efficiently in night, to operate radiative cooling systems in daytime, the diffusive or direct solar radiation should prevented from entering the cooling system. Thus the installation of such water condensation system with evaporative cooling applications may prove a quality improving strategy of evaporative cooling systems. The installation of such water recovery system with evaporative cooling systems provides pure distilled water as condensate output without affecting the performance of the cooling system in any regard.

At last it can be concluded that the combination of such water condensation system with evaporative cooling applications may prove a quality improving technique of evaporative cooling systems that will be beneficial for the human beings by two ways, first for thermal comfort and second getting potable water as by product of the cooling system, the water thus obtained is pure water which can be further used in various applications.

Table: 1 Nomenclature

Nomenclature			
A	emitter surface area(m^2)	Re_δ	Reynolds number
b	liquid film width(m)	t	time (s)
c	emitter specific heat capacity ($JKg^{-1}K^{-1}$)	T	temperature (K)
$c_{p,l}$	liquid film specific heat capacity($JKg^{-1}K^{-1}$)	T_c	critical temperature (K)
g	gravitational acceleration (ms^{-2})	T_r	reduced temperature (T/T_c)
h_L	average convective heat transfer coefficient ($Wm^{-2}K^{-1}$)	T_s	emitter surface temperature (K)
h_{fg}	latent heat of condensation (JKg^{-1})	T_{sat}	vapour saturation temperature corresponding to P_v(K)
h'_{fg}	augmented latent heat of condensation (JKg^{-1})	T_{sky}	effective sky temperature (K)
k_1	liquid film thermal conductivity (W $m^{-1}K^{-1}$)	T_∞	ambient temperature (K)
L	vertical length 0of emitter(m)	U	overall convection heat transfer Coefficient (W $m^{-2}K^{-1}$)
m	emitter mass(Kg)	Greek Symbols	
ṁ	condensation mass flow rate (Kg s^{-1})	δ	liquid film thickness (m)
P_c	critical pressure (Pa)	ε	emitter emissivity
P_g	saturation pressure at T_∞ P_a	μ_1	liquid film viscosity (Kg $m^{-1}s^{-1}$)
P_r	reduced pressure (P/P_c)	v_l	liquid film kinematic viscosity (m^2s^{-1})
P_v	vapour partial pressure (Pa)	ρ_l	liquid film density (Kg m^{-3})
P_∞	ambient pressure (Pa)	ρ_v	vapour density(Kg m^{-3})
Q	heat flux (W m^{-2})	σ	Stefan-Boltzmann constant (W $m^{-2}K^{-4}$)
		Ø	relative humidity

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