Performance model for small scale indirect evaporative Cooler.

Gert Boxem[1], Sebastiaan Boink[2] ,Wim Zeiler[1]

Technical University Eindhoven, The Netherlands
[1]fac. Building & Architecture, [2]fac. Mechanical Engineering

Corresponding email: g.boxem@bwk.tue.nl

SUMMARY

Indirect evaporative cooling an efficient and cheap way to reduce the heat load in a building. For an innovative small scale evaporative cooler (400 m^3/h) a model is made to predict the outlet temperature and cooling capacity for different environmental conditions in order to estimate the seasonal heat load reduction. The cooler is a compact counter flow heat exchanger with louver fin geometry on either side; the secondary side is coated with a water absorbing material. Due to the direct evaporation from the surface and the counter flow operation outlet temperatures below the wet bulb temperature can be achieved. The model uses Chilton & Colburn approximation for the evaporation, an approximation for the j-factor as found by Chang and Wang. The resulting model is based on estimated geometry properties and predicts the outlet temperature for given inlet conditions an airflow rates. The model predicts an over performance of the output temperature. For the inlet temperature below 24°C there a 20 % overshoot of the model, for higher inlet temperatures the overshoot is less than 10%.

INTRODUCTION

The demand for cooling increases due to higher comfort demand, compact way of building, higher internal heat load ect. For small scale building as primary schools, small utility buildings a demand for a simple way of cooling is desirable. Evaporative (adiabatic) cooling is an energy efficient way of cooling, though it has a drawback that with increasing outdoor humidity the cooling capacity decreases. In this paper a performance model for an indirect evaporative cooling is described. The purpose of the model is to use it in order to predict the effective cooling performance for use in building performance studies.

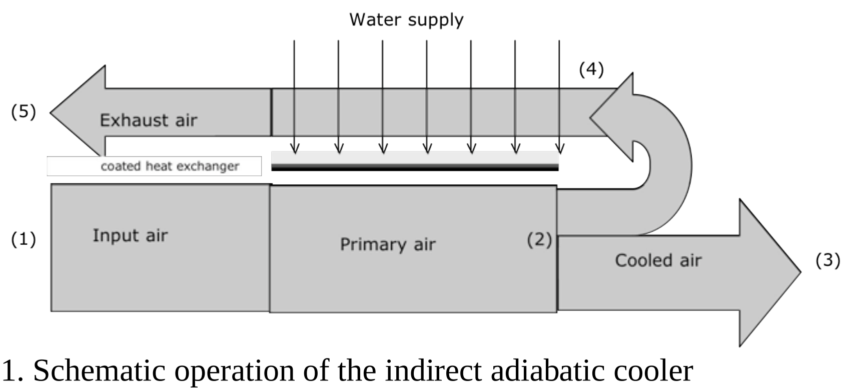


Figure 1. Schematic operation of the indirect adiabatic cooler

Operation

The main advantage of indirect adiabatic cooling is that the water content of the cooled air is not increased. The studied cooler is an innovative design where efficient use is made of counter flow geometry and of direct evaporation of water from the separating wall. The principle operation is given in fig.1.

On the primary side of the heat exchanger outside air (1) is cooled to outlet conditions. About 1/3rd of the air (2) is fed back to the secondary side (4). The secondary side of the heat exchanger has a hygroscopic coating, saturated with water. With regular intervals (typically 10min.) fresh water is sprayed onto the coating. Due to the direct evaporation from the wall a counter flow operation the outlet temperature (3) approximates the dew temperature of the inlet air. The secondary outlet air (5) is exhausted to the environment.

Construction

The heat exchanger consists of fin geometry according to Figure 3. The copper fins (primary and secondary) are glued back to back on a thin plastic foil. The separation plates between the fins achieve construction stability. This results in a very lightweight heat exchanger that after service life easily can be replaced.

Objective.

Measurements of the performance of the cooler have show that the outlet temperatures of the cooler are very near the dew point of the inlet air. This performance is due to the effective counter flow operation and the direct vaporization of water from the coated fins. In order to get insight in the performance of the cooler a model is made to relate the convective heat transfer on the primary side to the combined heat and mass transfer on the secondary side. The mathematical model can be used to enhance the performance of the cooler for different configurations and also can be used to predict the performance of the cooler

METHOD

The main principle of the cooler is the evaporation of water on the secondary side. The heat transfer consists of a convective part and a mass transfer part due to the evaporation. An enthalpy balance is set up for a surface element in the heat exchanger (Figure 2).

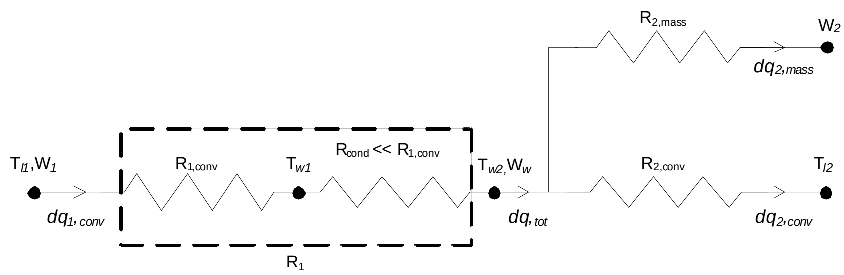


Figure 2. Model of the enthalpy exchange for a surface element of the heat exchanger.

In order to find a solution for (1) approximations have to be found for the convective heat and convective mass transfer coefficients h_c, h_m.

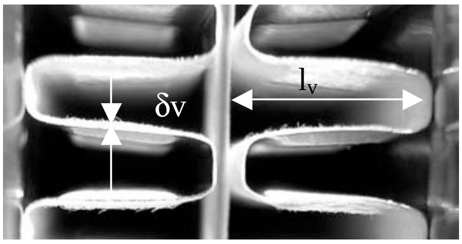
$$d(T_1,T_2,w_1,w_2)\ of\ dq_{tot} = f(\underbrace{\dot{V}_1,\dot{V}_2}_{\text{Operational Flows}}, \underbrace{A_c,A_m}_{\text{constrution}}, \underbrace{T_1,T_2,w_1,w_2}_{\text{Proces variables}}, \underbrace{\rho_l,c_{pl},c_{pw,g},c_{pw,f},h_{fg}}_{\text{material properties}}, \underbrace{h_c,h_m}_{\text{process parameters}}\ \underbrace{T_{w2},w_w}_{\text{other variables}}) \quad (1)$$

Chilton-Colburn correlation.
The heat exchanger is build up with louvered fins according to figure 3. For an estimate of the mass convection coefficient h_m the Chilton-Colburn analogy in a boundary layer is used and expressed in the dimensionless Colburn factor j_c. [1]

$$\frac{Nu}{\mathrm{Re}\,\mathrm{Pr}^{1/3}} = j_c = \frac{1}{2} f \qquad for \quad 0.5 < \mathrm{Pr} < 2 \quad (2)$$

The friction factor *f* and the Colburn factor j_c are supposed to be only a function of the Reynolds number.
The Reynolds number based on the hydraulic diameter of the canals gives *Re* = 570 and suggest a laminar flow. Though to the louvers on the fins flow through the fins leads to a turbulent flow, a so-called "low Reynolds-number turbulent flow" [2].
In the paper "A generalized friction correlation for louver fin geometry" by Chang e.o. [3] experimental correlations are given for a different arrangement of fins and louvers. For this study a general approximation from [3] is used to estimate j_c.

$$j_c = C_{jc}\,\mathrm{Re}^{-n} = 0.425\,\mathrm{Re}_{Lp}^{-0.496} \qquad (3)$$



l_v = 8mm
δ_v= 0.05 mm
k_v=401 W/mK

Figure 3 Fin geometry of the heat-exchanger

From (3) an expression fort the heat-transfer coefficient h_c can be derived:

$$h_c = \frac{C_{j_c}\left(\frac{\mu_l c_p}{k_l}\right)^{1/3}}{\left(L_p / k_l\right)} \left(\frac{L_p}{\mu A_k}\right)^{1-n} \left(\dot{m}\right)^{1-n} \qquad (4)$$

For the given configuration with A_k=0.0048m^2, L_p= 1mm, and the general properties of air this gives:

$$h_c = 344 \cdot \dot{m}^{0.505} \qquad (5)$$

According to Chilton-Colburn j_m=j_c and from (6) h_m can be expressed in h_c (7)

$$\frac{Sh}{ReSc^{1/3}} = \frac{Nu}{RePr^{1/3}} \Rightarrow h_m = h_c \frac{D_{AB}}{k_l} \underbrace{\left(\frac{k_l}{c_{pl}\rho_l D_{AB}}\right)^{1/3}}_{Lewis\ number} \qquad (6,7)$$

With ((5) follows an estimate for h_m:

$$h_m = 9.46 \cdot 10^{-4} \;\Rightarrow\; h_m = 0.325 \cdot \dot{m}^{0.505} \qquad (8)$$

Assumptions and simplifications.
With the found heat and mass a heat coefficients a heat and mass balance can be derived. The material properties of water and air (Table 2) are assumed to be constant over the length of the heat exchanger. The thermal resistance R_{cond} (figure 2) is assumed to be negligible to the convective heat resistances.
The effective conducting surface is based is based on the number of fins, the length and an estimated fin efficiency according to [4].

$$\eta_v = \frac{\left(tanh\left(ml_v\right)\right)}{ml_v} \quad \text{with} \quad m = \sqrt{\frac{2h_c}{k_v \delta_v}} \qquad (9)$$

With a length of the exchanger of 0.48 m, 108 (two sided) fins, height l_v=0.008 m A_v=1.66 m^2 and with a fin efficiency according to (9) the effective surface can be expressed as a function of h_c.

$$A_{lc} = \eta_{lv} A_v \approx -1.5 \cdot 10^{-3} h_c A_v = -2.49 h_c \qquad (10)$$

The fins are coated only on one side so $A_{lm} = \frac{1}{2} A_{lc}$.

The temperature drop due to the thermal resistance of the wetted coating is neglected. Assumed is that the humidity at the boundary of secondary side is saturated. The capillary resistance of the coating is also neglected.

Model.
The resulting model is a set of 3 coupled (11,12,13) witch can be numerically solved with boundary conditions (14,15,16).

$$\frac{dT_1}{dx} = -\underbrace{\frac{1}{\rho_l \dot{V}_1 (c_{pl} + w_1 c_{pw,g})}}_{Constant} \left(\frac{(T_1 - T_2)}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} + \frac{\frac{1}{A_{2c}h_{2c}}}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} \cdot \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \right) \qquad (11)$$

$$\frac{dT_2}{dx} = -\frac{1}{\rho_l \dot{V}_2 (c_{pl} + w_2 c_{pw,g})} \left(\frac{(T_1 - T_2)}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} - \frac{\frac{1}{A_{1c}h_{1c}}}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} \cdot \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \right) \qquad (12)$$

$$\frac{dw_2}{dx} = -\frac{1}{\rho_l \dot{V}_2 (h_{fg} + c_{pw,g} T_2)} \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \qquad (13)$$

$$T_1(0) = T_{1,in}\ ;\ T_2(1) = T_{1,uit} \qquad \text{and} \quad w_2(1) = w_{1,uit} = w_{1,in} = w_1(0) \qquad (14,15,16)$$

The equations are solved with Matlab® the used routines are collected in [9].

Results

Figure 4 shows the heat transfer along the length of the exchanger. The secondary air enters from the right. |At first the latent heat transfer (2) is also used to cool the secondary air, the sensible heat transfer (2) is negative. Due to the vapour transfer along the total length of the heat exchanger a good performance is achieved. The latent heat transfer is about 2 times larger than the sensible heat transfer.

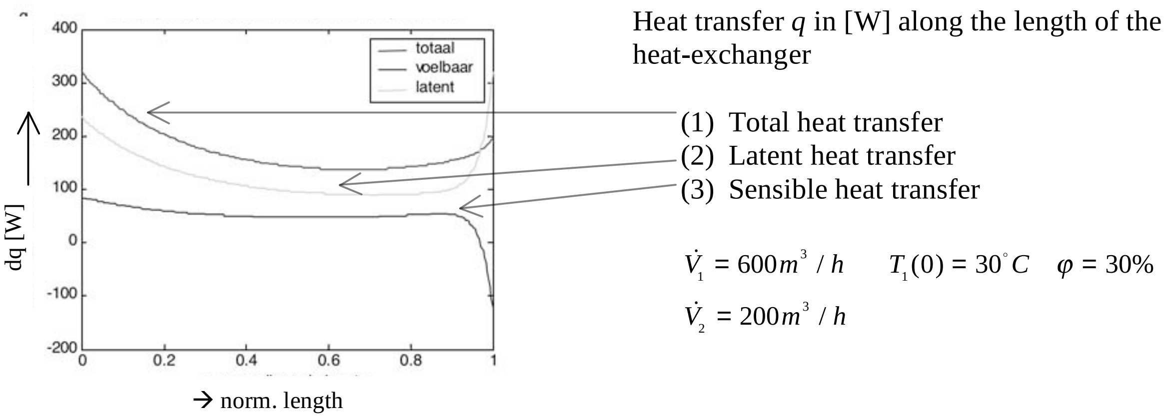


Figure 4 Latent en Sensible heat transfer along the length x/L of the heat exchanger.

Table 1. shows the calculated results of the model, the sensible cooling strongly depends on the humidity of the inlet air. Between rel. humidity 30 and 70% the temperature difference ΔT drops from 10-15K to 4-5 K. In Table 2 the simulated data is compared to measured data of the cooler [5]

Table 1 Predicted temperature difference between inlet en outlet air.

	Rel. humidity of inlet air.					
	30%		*50%*		*70%*	
T_{in} [°C]	T_{out} [°C]	ΔT [K]	T_{out} [°C]	ΔT [K]	T_{out} [°C]	ΔT [K]
22	10.1	10.0	13.8	8.2	17.2	4.8
23	10.6	10.5	14.5	8.5	18.1	4.9
24	11.1	11.2	15.2	8.8	18.9	5.1
25	11.7	12.0	15.9	9.1	19.8	5.2
26	12.2	12.5	16.6	9.4	20.7	5.3
27	12.7	13.2	17.3	9.7	21.6	5.4
28	13.3	14.0	18.1	9.9	22.4	5.6
29	13.8	14.5	18.9	10.1	23.3	5.7
30	14.4	15.0	19.6	10.4	24.2	5.8
31	14.9	15.7	20.4	10.6	25.1	5.9
32	15.5	16.5	21.2	10.8	26.1	5.9

Table 2. Comparison of model results and measured data [5]

	Rel. humidity of inlet air.								
	30%			*50%*			*70%*		
T_{in} [℃]	Model	Meas.	Δ	Model	Meas.	Δ	Model	Meas.	Δ
22	10.1	12.0	-19%	13.8	15.1	-16%	17.2	18.2	-21%
23	10.6	12.5	-18%	14.5	15.6	-13%	18.1	18.8	-14%
24	11.1	12.8	-15%	15.2	16.2	-11%	18.9	19.3	-8%
25	11.7	13.0	-11%	15.9	16.7	-9%	19.8	20.0	-4%
26	12.2	13.5	-10%	16.6	17.2	-6%	20.7	20.8	-2%
27	12.7	13.8	-8%	17.3	18.0	-7%	21.6	21.8	-4%
28	13.3	14.0	-5%	18.1	18.2	-1%	22.4	22.5	-2%
29	13.8	14.5	-5%	18.9	19.0	-1%	23.3	23.6	-5%
30	14.4	15.0	-4%	19.6	19.8	-2%	24.2	24.5	-5%
31	14.9	15.3	-3%	20.4	20.5	-1%	25.1	25.5	-7%
32	15.5	15.5	0%	21.2	21.2	0%	26.1	26.8	-12%

DISCUSSION

The model gives a good insight in the principal operation of the cooler, although based on experimental approximation for fin geometry; the model gives a reasonable approximation of the performance. The found model gives (Table 2) an over estimation of the temperature drop of max. 20%. The neglect of: heat conductance through the wall, the moisture transport through the coating all causes an overestimation of the cooling effect. For the lower inlet temperatures (<24 °C) the secondary air over the total length is staying close to the saturation line, when the air at the surface is not fully saturated, as was assumed, this will have a significant effect on the cooling performance. The neglected R_{cond} of the wall will also lead to an increase of error at lower temperature.
To improve the model for the lower temperature range the thermal conductance of the wall and the vapour transport through the coating should be incorporated in the model.

Another thing was not accounted for in the model, is or the long term the saline deposition could lead to an extra thermal barrier. The manufacturer claimed on basis of endurance test that the deposition does not affect the water transport in the coating.

Table 3 Used material properties for air, water and copper

Air	Water	Copper
$\rho_l = 1.185 \quad kg \cdot m^{-3}$	$\rho_{w,f} = 998.0 \quad kg \cdot m^{-3}$	$k_v = 401 \quad W \cdot m^{-1} \cdot K^{-1}$
$c_{pl} = 1006.9 \quad J \cdot kg^{-1} \cdot K^{-1}$	$h_{fg0} = 2502 \quad J \cdot kg^{-1} (273.15\,K)$	
$\mu_l = 18.21 \cdot 10^{-6} \quad N \cdot s \cdot m^{-2}$	$c_{pw,f} = 4181 \quad J \cdot kg^{-1} \cdot K^{-1}$	
$k_l = 25.9 \cdot 10^{-3} \quad W \cdot m^{-1} \cdot K^{-1}$	$c_{pw,g} = 1868 \quad J \cdot kg^{-1} \cdot K^{-1}$	
$D_{AB} = 0.26 \cdot 10^{-4} \quad m^2 \cdot s^{-1} (T=298\,K)$		Source: [3]

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Used Symbol

A_c	convective surface (m^2)	q_{conv}	Sensible heat transfer (W)
A_m	masstransfer surface (m^2)	q_{mass}	latent heat transfer (W)
ρ_l	density of air (kg/m^3)	q_{tot}	total heat transfer (W)
c_{pl}	spe(m^2K/W)	R_{conv}	conv. thermal resistance (m^2K/W)
$c_{pw,f}$	cific heat of air (J/kg.K)	R_{cond}	cond. thermal resistance (m^2K/W)
$c_{pw,g}$	specific heat water (J/kg.K)	R_{mass}	mass. conv. resistance (m^2K/W)
μ_l	specific heat of vapour (J/kg.K)	h_c	thermal convection coeff. (W/m^2K)
D_{AB}	dyn. viscosity of air (Ns/m^2)	h_m	mass convection coeff. (kg/m^2s)
p	diff. coeff. water in air (m^2/s)	k_l	heat conductance of air (W/mK)
p_{sat}	air-pressure (Pa)	k_w	heat conductance of wall (W/mK)
w_{sat}	sat. vapour pressure (Pa)	k_v	heat conductance of fin (W/mK)
w	sat. moisture content (kg/kg)	L_w	thickness of wall (m)
T	moisture content (kg/kg)	u_m	average airspeed (m/s)
$\dot{V}$	temperature (K)	D_h	hydraulic diameter (m)
$\dot{m}$	volumeflow (m^3/s)	l_v	length of fin (m)
h	massflow (kg/s)	δ_v	thickness of fin (m)
h_{fg}	specific enthalpy (J/kg)	η_v	fin efficiency (-)
	evap. enthalpy water (J/kg)		

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