Modeling and Design Analysis of a Regenerative Indirect Evaporative Heat Exchanger Using an Effectiveness Method

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ABSTRACT

Indirect evaporative cooling (IEC) is a water-based cooling technology that is attractive for space cooling in dry and hot climates due to its lower energy consumption (compared to vapor compression air conditioners) and lack of humidification (compared to direct evaporative cooling). The key component of advanced IEC or hybrid IEC/DX (Direct eXpansion vapor compression) systems(Elberling 2006) is the IEHX (indirect evaporative heat exchanger). A practical accurate model of an IEHX is needed to characterize the thermal behavior of these coolers and to support their implementation by HVAC designers. This paper presents a methodology for the thermal modeling of IEHXs that is analogous to the effectiveness-NTU method for sensible heat exchangers. The governing differential equations that describe IEHX behavior are transformed to a set of analytical equations that can be solved stably within short computation times. The model can be applied without any performance data for simple cases, or using measurements at two different air flow rates (often available in manufacturers' catalogues). The model's performance was investigated utilizing experimental data from the literature. As an implementation example, the method was used to analyzing the sizing of a plate regenerative-type IEHX under constraints of fixed HX volume and space cooling load. It produced an optimal channel height of 2-3 mm to best satisfy specified trade-offs between wet bulb effectiveness, coefficient of performance, and cooling produced per unit area.

INTRODUCTION

Indirect evaporative cooling (IEC) is a water-based cooling technology that is attractive for space cooling in dry and hot climates due to its lower energy consumption (compared to vapor compression air conditioners) and lack of humidification (compared to direct evaporative cooling)(Maheshwari, Al-Ragom et al. 2001). Among the components incorporated into advanced IEC or hybrid IEC/DX systems(Elberling 2006), the IEHX (heat exchanger) is the most critical one, as it is the core technology, or the heart of the system.

A typical IEHX consists of a number of thin parallel plates assembled to form a multi-layer sandwich of alternating dry and wet channels. The primary air stream goes through the dry channel, being cooled down by transferring heat across

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the wall to the wet channel, where water evaporates and the heat is discharged via the secondary air stream in the wet channel. The cooled primary air is then supplied to the conditioned space.

There are many possible configurations for IEHX designs and their performance is heavily dependent on the operating conditions and the climate in which they are applied (Erens and Dreyer 1993). Among the possible design configurations, the regenerative type IEHX is the most commonly used (Fig. 1-a). It siphons off a portion of the sensibly pre-cooled primary air stream, and injects that air into the secondary air (wet) channel to reduce the secondary-air wet-bulb temperature, resulting in much higher effectiveness as compared to the single-stage IEHX. The regenerative IEHX coolers have the potential to cool the primary air to its dew point temperature. Examples of research in this area include modeling work (Hsu, Lavan et al. 1989; Zhan, Zhao et al. 2011; Hasan 2012) and laboratory testing (Riangvilaikul and Kumar 2010), as well as field monitoring(Bruno 2011). (Hsu, Lavan et al. 1989) numerically investigated three types of configurations, namely parallel-flow, counter-flow and regenerative IEHXs, using a finite difference method, and the simulation results show the regenerative type IEHX holds the highest wet bulb effectiveness of 130%. (Zhan, Zhao et al. 2011) numerically studied an M-cycle (regenerative type) IEHX using a finite element method and their results indicate that a wet bulb effectiveness of 167% could be achieved, much higher than conventional cross-flow IEHXs(typically 60-80%). (Riangvilaikul and Kumar 2010) tested a plate regenerative (counter flow) IEHX and the tested wet bulb effectiveness ranged between 92 and 114% at a wide range of ambient air conditions. Field testing of a commercially available IEHX (Bruno 2011) also indicated a wet bulb effectiveness that ranged from 93% up to 106% for commercial applications, and from 118% up to 129% for residential applications. All the above investigations indicate that the regenerative type IEHX has much higher wet bulb effectiveness than conventional IEHXs.

Due to its high wet-bulb effectiveness, the regenerative type IEHX is attractive to HVAC engineers for energy efficient space cooling. However, a thorough literature review indicates that a practical accurate model that supports the implementation of IEHX by HVAC designers is not available. The existing numerical (finite difference/element) models often require detailed IEHX characterization data as inputs and some sophisticated numerical methods for solutions, resulting in complex solutions and long computational times. This type of simulation is typically not acceptable for hour-by-hour annual simulation of an IEC operated in a wide range of climates under multiple control strategies, nor is suitable for size optimization. Because such system-level simulation and optimization processes require thousands of computation solutions, short computational time is crucial. To be most useful, the model should be incorporated into building simulation packages, e.g.(EnergyPlus) such that it is simple and practical for HVAC designers to use.

This paper attempts to model and analyze a regenerative IEHX using a simplified IEHX model previously proposed by the authors(Liu, Modera et al. 2012). The model should meet the needs of HVAC designers, including:

1. Simplicity of data inputs
2. Ability to model different operating conditions and climates
3. Short computational time
4. Reasonable accuracy

THERMA-HYDRAULIC MODELLING

Figure 1 shows a regenerative IEHX with a counter-flow configuration, as well as the thermal resistances across the modeling element. Heat transfer in the dry channel of an IEHX is similar to that in a sensible heat exchanger (Kaka, Shah et al. 1987); while heat and mass transfer in the wet channel is analogous to that in an indirect-contact cooling tower (ElDessouky, AlHaddad et al. 1997), both of which have been studied extensively. Here the regenerative IEHX is modeled utilizing a methodology proposed by the authors that is analogous to the classic effectiveness-NTU method for sensible heat exchangers(Liu, Modera et al. 2012). The governing differential equations that describe IEHX behavior are transformed to a set of analytical equations that can be solved stably within short computation times. The model's performance was investigated and validated utilizing experimental data from the literature. Detailed model assumptions, derivation and validation can be found in (Liu, Modera et al. 2012)

The ε -NTU method for analysis of sensible HXs and IEHXs is illustrated in Table 1. A variable, $\overline{K}$, analogous to the air specific heat c_{pa} in the dry channel, is introduced for the thermal resistance calculation in the wet channel. Detailed procedures for determining $\overline{K}$ and UA can be found in (Liu, Modera et al. 2012).

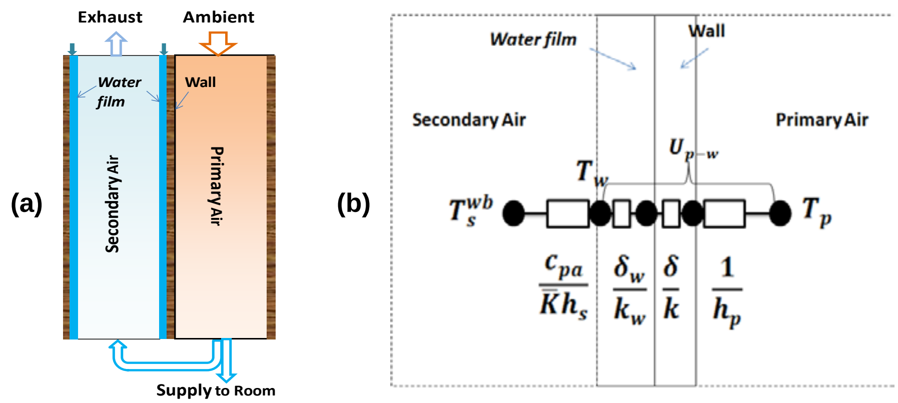


Figure 1 Regenerative IEHX with counter-flow configuration (a) sketch, and (b) thermal resistances across the element.

Table 1. Procedure of ε -NTU method for sensible HXs and IEHXs analysis(Liu, Modera et al. 2012)

ε -NTU method for Sensible HXs	Modified ε -NTU method for IEHXs
$C_c = \dot{m}_c c_{pc};\ C_h = \dot{m}_h c_{ph}$	$C_c = \dot{m}_s \overline{K};\ C_h = \dot{m}_p c_{pa}$
$C_{min} = \min(C_c, C_h)\,; C_{max} = \max(C_c, C_h)\ ;\ C_r = \frac{C_{min}}{C_{max}};\ NTU = \frac{UA}{C_{min}}$	
where, $\frac{1}{U} = \frac{1}{h_h} + \frac{\delta}{k} + \frac{1}{h_c}$	$\frac{1}{U} = \frac{1}{h_p} + \frac{\delta}{k} + \frac{\delta_w}{k_w} + \frac{c_{pa}}{\overline{K}h_s}$
$\varepsilon = f(NTU, C_r, flow\ arrangement)$ *where,* $\varepsilon = \frac{1-\exp(-NTU(1-C_r))}{1-C_r\exp(-NTU(1-C_r))}$ *(counter flow);* $\varepsilon = 1 - exp[(\frac{1}{C_r})(NTU)^{0.22}\{exp[-C_r(NTU)^{0.78}] - 1\}]$ *(cross flow (both fluids unmixed))*	
$q_{max} = C_{min}(T_{h,i} - T_{c,i})$	$q_{max} = C_{min}(T_{p,i} - T_{s,i}^{wb})$
$q = \varepsilon\, q_{max}$	
$T_{h,o} = T_{h,i} - \frac{q}{C_h};\quad T_{c,o} = T_{c,i} + \frac{q}{C_c}$	$T_{p,o} = T_{p,i} - \frac{q}{C_h};\quad T_{s,o}^{wb} = T_{s,i}^{wb} + \frac{q}{C_c}$

Given that the wet side entering air temperatures (equal to the supply air temperatures in Figure 1) are unknown a priori, iteration is required. Figure 2 is the computational flow chart that includes an iteration loop to find the outlet air temperature of primary air T_{po}. The $\overline{K}$ value is also iteratively obtained as detailed in (Liu, Modera et al. 2012).

Besides the thermal prediction, the fan power for operating an IEHX is estimated. The fan power is a combination of the product of air flow rate and the corresponding pressure drop in the dry and wet channels.

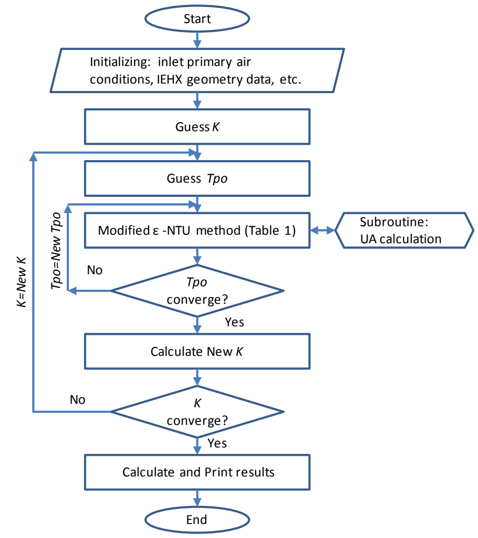


Figure 2: Computational flow chart for regenerative IEHX modeling

$$Fan\ power = \frac{\Delta P_p \dot{m}_p + \Delta P_s \dot{m}_s}{\eta \rho} \tag{1}$$

where, η is the fan efficiency.

The channel pressure drop ΔP is calculated with the Darcy formula,

$$\Delta P = f \frac{L}{D_h} \frac{\rho V^2}{2} \tag{2}$$

where, f is the average friction factor for flow in parallel plates, calculated from the correlation(Shah 1978) including entry effects as well as fully developed flow friction.

$$f = \frac{4}{Re}\left[\frac{3.44}{\sqrt{L^+}} + \frac{\frac{1.25}{4L^+} + \frac{96}{4} - \frac{3.44}{\sqrt{L^+}}}{1 + \frac{0.00021}{(L^+)^2}}\right] \tag{3}$$

where, L^+ is a Graetz-type variable $L^+ = (\frac{D_h}{L})Re$.

The room-based cooling capacity is determined by:

$$Q_{cap} = (1 - x)\dot{m}_p c_{pa}(T_{rm} - T_{p,o}) \tag{4}$$

where, x is the fraction of primary air re-entering the wet side.

The wet-bulb effectiveness of the IEHX based on Pescod's definition (Pescod 1979) is:

$$\varepsilon^{wb} = \frac{T_{p,i} - T_{p,o}}{T_{p,i} - T_{s,i}^{wb}} \tag{5}$$

The COP is defined as:

$$COP = \frac{Q_{cap}}{fan\ power} \tag{6}$$

The normalized cooling per unit area is defined as:

$$q = \frac{Q_{cap}}{A} \tag{7}$$

MODEL IMPLEMENTATION EXAMPLE

The model can be used to determine the performance of the IEHX over a wide range of ambient conditions, cooling loads, air flow rates, and geometries. Due to space constraints, for this example the following parameters are fixed: ambient conditions, cooling load, design room temperature, and IEHX width, length, and total height. The analysis focuses on the effect of primary air reuse fraction, channel height and channel air velocity. To account for the effect of the water film on the pressure drop through the wet channel, it is assumed that the pressure drop in the wet condition is 20% higher than that in the dry condition with same channel size and flow rates(Pescod 1979). The assumed constraints, design variables, and performance indicators are listed in table 2.

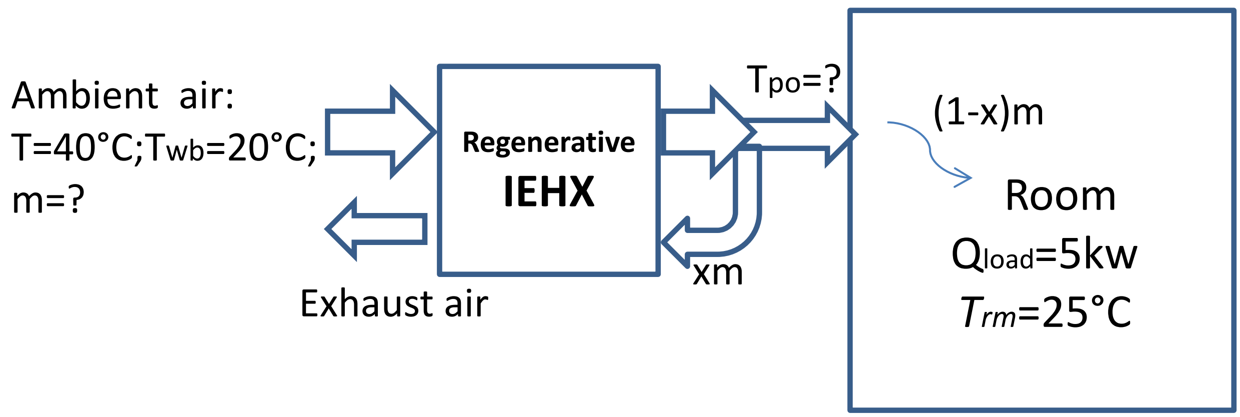


Figure 3. Sketch of the simulated regenerative IEHX linked to room with design constrains.

Table 2. Assumed constraints, Design Variables and Performance Indicators

Constraints	**Values**	**Design Variables**	**Symbols**
Cooler Volume (m^3/ ft^3)	0.4/14.1	Dry channel air velocity	V
Design room temperature(°C/°F)	25/77	Fraction primary air re-entering the wet channel	x
Room cooling load(kW/ton)	5/1.42	Channel height(dry/wet)	H
Ambient dry bulb temperature(°C/°F)	40/104	**Performance Indicators**	
Ambient wet bulb temperature(°C/°F)	20/68	Room based cooling capacity	Q_{cap}
Total HX height (m/ft)	1.2/3.94	Coefficient of performance	COP
HX plate length(equal to width) (m/ft)	0.6/1.97	Wet bulb effectiveness	ε_{wb}
		Area normalized cooling produced	q

The first variable to be considered will be the primary air re-entry fraction, which is perhaps the most critical parameter for designing a high performance regenerative IEHX (Hsu, Lavan et al. 1989; Erens and Dreyer 1993). A larger re-entry fraction will result in higher wet-bulb effectiveness (equivalent a lower supply air temperature), at the expense of a reduced air flow to the room, which reduces the cooling capacity of the unit. Therefore, it is desirable to examine the re-entry fraction first.

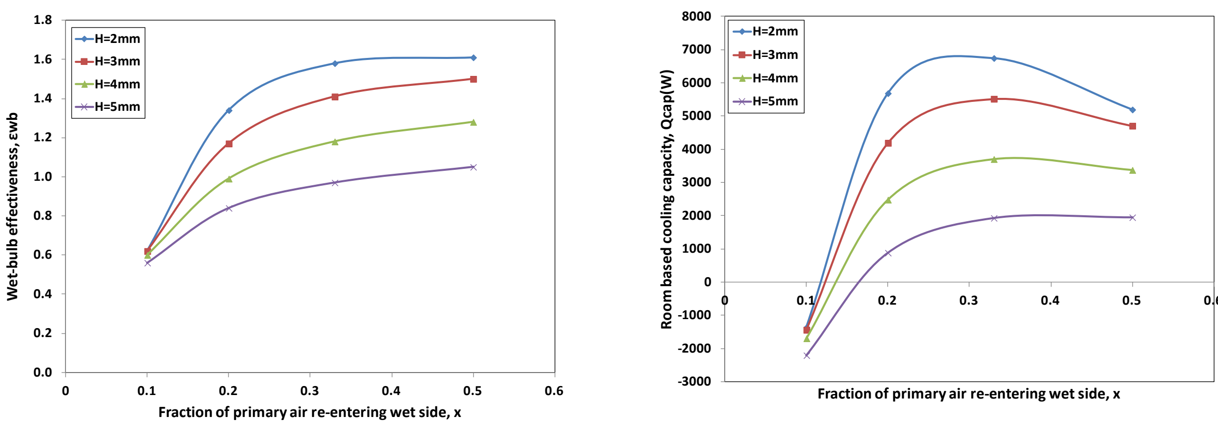
After finding the optimal re-entry fraction at certain fixed conditions, the effect of channel height can be considered. In the literature, several studies have explored design variables like channel height with their models (Hsu, Lavan et al. 1989; Guo and Zhao 1998; Zhan, Zhao et al. 2011). However, those explorations are all parametric analysis of the design variables, seldom performing the analysis under constraints associated with real world cooling applications. In fact, engineering design analysis can only be meaningful when given constraints. This example will perform such analysis under pre-set constraints.

RESULTS AND DISCUSSION

A number of simulations were conducted using the validated IEHX model(Liu, Modera et al. 2012). The channel air velocity, channel height, and fraction of primary air re-cycled were varied as shown in Table 3.

Table 3. Variation of the Design Variables					
Design Variables	Primary air velocity, V		Channel height, H		Fraction of primary air to wet side, x
Units	m/s	ft/s	mm	Inch	-
Range	1~6	3.3~19.7	1~6	0.04~0.24	0.1-0.5
Increment	1	3.3	1	0.04	-

The first step was to find the optimal value of x. Initially simulation, only the fraction x and the channel height H were varied, while the air velocity in the dry channel was fixed at 2m/s (6.6 fps). The ambient air condition and room temperature were chosen to be equal to those in Table 2. The simulation results are illustrated in Figure 4, which indicates that the optimal x varies only slightly from 0.3 to 0.4 between channel heights of 2-5mm (0.08-0.2 in.). Higher x value results in higher wet bulb effectiveness, but the most variation occurs at small fractions (Figure 4-a). The x value for maximum cooling capacity shifts slightly from 0.25 to 0.4 as the channel height increases from 2mm to 5mm (0.08 to 0.2 in.) (Figure 4-b). In addition, smaller channel height results in higher wet bulb effectiveness and cooling capacity, which agrees with previous work(Hsu, Lavan et al. 1989).



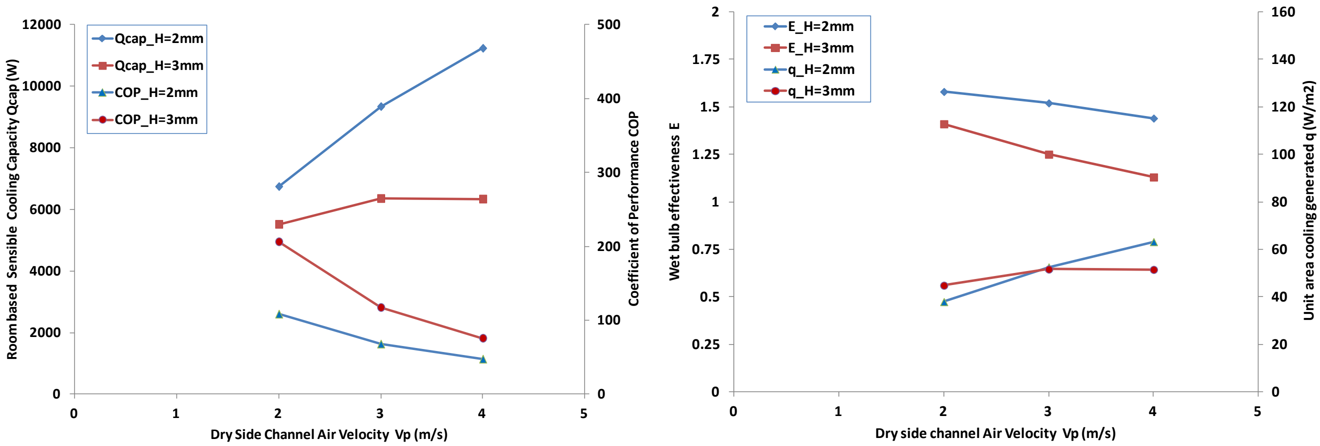
(a) **(b)**

Figure 4. Variations of wet-bulb effectiveness (a), and room based cooling capacity (b) at different fractions of primary air re-entering the wet side, x, and various channel heights *H* (channel air velocity is fixed at 2m/s(6.6 fps)., channel length is 0.6m(1.97ft)).

After the optimal value of x is found (here we choose x=0.33), another group of simulations was conducted with the design variables specified in Table 3. Table 4 summarizes the simulation results that deliver the required cooling while satisfying the constraints listed in Table 2. As shown in Table 4, a 2mm (0.08 in.) channel height and 4m/s (13.3 fps) primary air velocity result in a maximum room based cooling capacity of 11234W(38332 Btu/h), but also correspond to the lowest COP of 48. Table 4 also shows that the unit with 2 mm (0.08 in.) plate spacing has a higher cooling capacity and wet bulb effectiveness but a lower COP, as compared to a 3 mm (0.12 in.) plate spacing at each primary air velocity. It should be remembered that the simulated COP values will be higher than those of an IEC cooler operated in the real world, because the energy use simulated in this model is idealized as the IEHX channel friction loss only, and ignores all other losses. Nevertheless, the simulated values of wet bulb effectiveness and COP are much higher than those associated with conventional IEHXs, consistent with experimental studies (Riangvilaikul and Kumar 2010; Bruno 2011).

Table 4. Selected simulation results that satisfy the constraints listed in table 2	
Inputs (x=0.33)	**Simulation Results**

Primary air velocity		Channel height		Channel length		Total HX height		Supply air temp.		Cooling capacity	Wet bulb effectiveness	COP	Area normalized cooling capacity	
m/s	fps	mm	In.	m	ft	m	ft	°C	°C	W	%	/	W/m^2	BTU/h
2	6.6	2	.079	0.6	1.97	1.1	3.61	8.3	46.9	6744	158%	108	38	130
3	9.8	2	.079	0.6	1.97	1.1	3.61	9.6	49.3	9342	152%	68	53	181
4	13.1	2	.079	0.6	1.97	1.1	3.61	11.1	52.0	11234	144%	48	63	215
2	6.6	3	.118	0.6	1.97	1.1	3.61	11.9	53.4	5518	141%	206	45	154
3	9.8	3	.118	0.6	1.97	1.1	3.61	14.9	58.8	6356	125%	117	52	178
4	13.1	3	.118	0.6	1.97	1.1	3.61	17.5	63.5	6338	113%	76	51	174



(a) **(b)**

Figure 5: Variation of (a) room based cooling capacity Q_{cap} and COP, as well as (b) wet bulb effectiveness ε_{wb} and unit area generated cooling q of a regenerative IEHX at different dry-channel air velocities V and channel heights H (selected values satisfy all the constraints).

Figures 5-a and 5-b respectively show the variation of room based cooling capacity Qcap and COP, and the wet bulb effectiveness ε_{wb} and area-normalized cooling q of a regenerative IEHX as a function of dry-side channel air velocities V and channel heights H. Figure 5 shows that the wet bulb effectiveness and COP decrease, while the cooling capacity increases with increasing channel air velocity. These differing trends show that the best choice of values for design parameters such as channel height and air velocity will depend to an extent on the priorities of the designer.

CONCLUSIONS

This paper details the modeling and design analysis of a regenerative IEHX using a proposed modified NTU-effectiveness method for IEHXs. The trade-offs among wet bulb effectiveness, cooling capacity, and energy use for a regenerative IEHX under several design constrains were shown. It is found that the optimal fraction of primary air re-entering the wet channel is 0.3-0.4 for channel heights between 2-5mm (0.08-0.2 in.) and that this fraction is not very sensitive to the channel height. The optimal channel height that results in maximum performance and satisfies all the design constrains is found to be 2-3 mm. Not surprisingly, simulation results indicate that a regenerative type IEHX produces a higher effectiveness than conventional IEHXs if they are carefully designed. The modeling and design analysis of a regenerative IEHX in this paper also suggests that the proposed effectiveness-NTU model is an efficient and practical model for HVAC engineers.

NOMENCLATURE

A = heat transfer area

c_{pa} = specific heat of moist air

C = fluid capacity

h/h_m = convective heat/mass transfer coefficient

H = channel height

$\overline{K}$ = Ratio of enthalpy change versus wet bulb temperature change in the wet side of IEHX

k = thermal conductivity

m = mass flow rate of fluids

NTU = number of heat transfer units

T = dry bulb temperature

$Qcap$ = cooling capacity

U = overall heat transfer coefficient

ε = effectiveness

δ = thickness of the plate or water film

Subscripts

c = cold

h = hot

i/o = inlet/out

p/s = primary/secondary

w = wall or water film

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